IN TWO SECTIONS—SECTION TWO

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Indexes to A.S.M.E. Papers and Publications

THIS and the following pages will serve as a guide to the current publications of the A.S.M.E.

Regular Society Publications, 1945

Mechanical Engineering, monthly (see index on page RI-91)
A.S.M.E. Transactions, monthly including the *Journal of Applied Mechanics* (see index on page RI-101)
A.S.M.E. Mechanical Catalog and Directory, 1946 edition.

Publications Issued in 1945

Bibliography on Cutting of Metals
Riveted Joints—A review of the literature covering their development, with bibliography and abstracts of the most important articles.
Sources of Information on Instruments
1943 Automotive and Oil-Engine Power Cost Reports

American Standards

Gear Tolerances and Inspection
Preferred Standards for Large 3600-Rpm 3-Phase 60-Cycle Condensing Steam Turbine-Generators
Pipe Threads
Screw Threads for High-Strength Bolting

Boiler Construction Code

1945 Addenda to:
Locomotive Boiler Code
Low-Pressure Heating Boiler Code
Power Boiler Code
Unfired Pressure Vessel Code
Specifications for Materials
Welding Qualifications

Power Test Codes

General Instructions
Definitions and Values
Part 3, Chapter 4—Resistance Thermometers
Part 20—Smoke-Density Determinations

Safety Codes

Elevator Inspector's Manual
Description and Schematic Layouts of Various Types of Undercar Safeties and Governors
Handling and Socketing of Wire Rope

How to Find Papers Presented at 1945 A.S.M.E. Meetings

THE technical programs of the meetings of the Society and of its Professional Divisions have been published in *Mechanical Engineering* and may be located by consulting the index on pages RI-91–RI-100. A majority of these papers were published, or will be published, in *Mechanical Engineering* or the Transactions (including the *Journal of Applied Mechanics*) and may be located by reference to the indexes of these publications.

Publications Developed by the Technical Committees

THE Society's technical committees, the first of which was organized many years ago and all of which have been continuously at work on codes, standards, research, and other special reports, have developed a series of publications of permanent value to the membership. The following list is presented here for record and for ready reference. This list covers the entire group of publications of these committees completed to date which are now available.

To assist members in securing copies of these publications the sale price is also given. A discount of 20 per cent is allowed to A.S.M.E. members on all publications except where otherwise noted.

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Large Rivets (B18.4—1937), $0.65
Plow Bolts (B18f—1928), $0.35
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Slotted-Head Proportions: Machine Screws, Cap Screws, and Wood Screws (B18c—1930), $0.45
Small Rivets (B18a—1927), $0.30
Socket Set Screws and Socket-Head Cap Screws (B18.3—1938), $0.40
Supplement to Socket Set Screws (B18.30—1944), $0.10
Tinners', Coopers', and Belt Rivets (B18g—1929), with 1942 Addendum, $0.35
Track Bolts and Nuts (B18d—1930), $0.40
Wrench-Head Bolts and Nuts and Wrench Openings (B18.2—1941), $0.65

Piping and Pipe Fittings

Air Gaps and Backflow Preventers in Plumbing Systems (A40.4—1942 and A40.6—1943), $0.45
Brass Fittings for Flared Copper Tubes (A40.2—1936), $0.35
Cast-Iron Pipe Flanges and Flanged Fittings for 25 Lb Maximum Saturated Steam Pressure (B16b2—1931), $0.40
Cast-Iron Pipe Flanges and Flanged Fittings Class 125 (B16a—1939), $0.60
Cast-Iron Pipe Flanges and Flanged Fittings Class 250 (B16b—1944), $0.45
Cast-Iron Pipe Flanges and Flanged Fittings for 800 Lb Maximum Hydraulic Pressure (B16b1—1931), $0.35
Cast-Iron Soil Pipe and Fittings (A40.1—1935), $0.65
Cast-Iron Long Turn Sprinkler Fittings for 150 and 250 Lb Maximum Saturated Steam Pressure (B16g—1929) and Addendum (B16g1—1937), $0.50
Cast-Iron Screwed Fittings for 125 and 250 Lb Maximum Saturated Steam Pressure (B16d—1941), $0.40
Cast-Iron Screwed Drainage Fittings (B16.12—1942), $0.45
Code for Pressure Piping (B31.1—1942), $2.00
Face-to-Face Dimensions of Ferrous Flanged and Welding End Valves (B16.10—1939), $0.55
Ferrous Plugs, Bushings, Lock Nuts, and Caps (B16.14—1943), $0.40
Malleable-Iron Screwed Fittings for 150 Lb Maximum Saturated Steam Pressure (B16c—1939), $0.50
Pipe Plugs (B16e2—1936), $0.35
Pipe Threads (B2.1—1945), $1.25
Scheme for the Identification of Piping Systems (A13—1928), $0.50
Steel Pipe Flanges and Flanged Fittings for 150 to 2500 Lb Maximum Steam Service Pressure (B16e—1939), $1.25
Soldered-Joint Fittings (A40.3—1941), $0.45
Steel Butt-Welding Fittings (B16.9—1940), $0.40
Threaded Cast-Iron Pipe for Drainage, Vent, and Waste Services (A40.5—1943), $0.25
Wrought-Iron and Wrought-Steel Pipe (B36.10–1939), $0.50

Letter and Graphical Symbols and Charts

Abbreviations for Scientific and Engineering Terms (Z10.1—1941), $0.35
Drawings and Drafting-Room Practice (Z14.1—1935), $0.50
Engineering and Scientific Charts for Lantern Slides (Z15.1—1932), $0.50
Engineering and Scientific Graphs for Publications (Z15.3—1943), $0.75
Graphical Symbols for Use on Drawings in Mechanical Engineering (Z32.2—1941), $0.50
Letter Symbols for Gear Engineering (B6.5—1943), $0.25
Letter Symbols for Hydraulics (Z10.2—1942), $0.35
Letter Symbols for Mechanics of Solid Bodies (Z10.3—1942), $0.25
Letter Symbols for Heat and Thermodynamics (Z10c—1943), $0.35
Time Series Charts (Z15.2—1938), $1.25

American Standards—Miscellany

Fire-Hose Coupling Screw Threads (B26—1925), $0.25
Hose Coupling Screw Threads (B33.1—1935), $0.25
Indicating Pressure and Vacuum Gages (B40—1939), $0.40
Preferred Thickness for Uncoated Thin Flat Metals (B32.1—1941), $0.25
Rolled Threads for Screw Shells of Electric Sockets and Lamp Bases (C44—1931), $0.35
Gear Tolerances and Inspection (B6—1945), $0.65
Preferred Standards for Large 3600-Rpm 3-Phase 60-Cycle Condensing Steam Turbine-Generators (published 1945), $0.25
Shaft Couplings (B49—1932), $0.35
Spur Gear Tooth Form (B6.1—1932) $0.45

Small Tools and Machine Tool Elements

Machine Tapers (B5.10—1943), $0.60
Milling Cutters (B5c—1930), $0.75
Reamers (B5.14—1941), $0.75
Taps—Cut and Ground Threads (B5.4—1939), $1.25
Terminology and Definitions for Single-Point Cutting Tools (B5.13—1939), $0.40
Adjustable Adapters (B5.11—1937), $0.50
Chucks and Chuck Jaws (B5.8—1936), $0.45
Circular and Dovetailed Forming Tool Blanks and Holding Elements (B5.7—1943), $0.50
Shafting and Stock Keys (B17.1—1943), $0.45
Markings for Grinding Wheels (B5.17—1943), $0.25
Involute Splines, Side Bearings (B5.15—1939), $0.65
Jig Bushings (B5.6—1941), $0.35
Lathe Spindle Noses (B5.9—1936), $0.50
Spindle Noses and Arbors (B5.18—1943), $0.25
Rotating Air Cylinders and Adapters (B5.5—1932), $0.35
Tool Holder Shanks Tool Post Openings (B5.2—1943), $0.35
Tool Life Tests (B5—1945), $0.40
T-Slots, Their Bolts, Nuts, Tongues, and Cutters (B5a—1941), $0.35
Twist Drills (B5.12—1940), $0.55
Code for Design of Transmission Shafting (B17c—1927), $0.75
Shafting and Stock Keys (B17.1—1934), $0.45
Screw Threads for Bolts, Nuts, Machine Screws, and Threaded Parts (B1.1—1935), $0.60
Screw Threads for High-Strength Bolting (B1.4—1945), $0.25
Screw Thread Gages and Gaging (B1.2—1941), $0.60
Acme and Other Translating Threads (B1.3—1941), $0.45
Tolerances, Allowances, and Gages for Metal Fits (B4a—1925), $0.50
Woodruff Keys, Keyslots, and Cutters (B17f—1930), $0.35

BOILER CONSTRUCTION CODE

1943 Editions: with 1944–1945 Addenda
- Locomotive Boiler Code, $0.75
- Low-Pressure Heating Boiler Code, $0.75
- Miniature Boiler Code, $0.50
- Power Boiler Code, $2.25
- Specifications for Materials, $3.00
- Suggested Rules for Power Boilers, $1.00
- Unfired Pressure Vessel Code, $1.50
- Welding Qualifications, $0.65

Boiler Code Interpretation Service, $5.00 annually
API-ASME Code for Unfired Pressure Vessels
- 1943 Edition, with 1944 Supplement, $1.25

POWER TEST CODES AND AUXILIARY SECTIONS

Test Codes for

Atmospheric Water-Cooling Equipment (1930), $0.45
Coal Pulverizers (1944), $0.70
Displacement Compressors, Vacuum Pumps, and Blowers (1939), $0.75
Dust Separating Apparatus (1941), $0.90
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Feedwater Heaters (1927), $0.35
Gaseous Fuels (1944), $0.75
Gas Producers (1928), $0.55
Hydraulic Prime Movers (1938 with 1942 Addenda), $0.60
Internal-Combustion Engines (1930), $0.55
Reciprocating Steam-Driven Displacement Pumps (1927), $0.65
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Steam Condensing Apparatus (1938), $0.65
Steam Locomotives (1941), $0.55
Steam Turbines (1941), $2.50
Appendix to Test Code for Steam Turbines (1943), $1.50

Auxiliary Sections

General Instructions (1945), $0.45
Definitions and Values (1945), $0.80
Part 1 — General Considerations (1935), $0.35
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- Chapter 1, Barometers; Chapter 6, Tables, Multipliers, and Standards (1941), $0.60
- Chapter 4, Bourdon, Bellows, Diaphragm, and Deadweight Gages (1938), $0.65
- Chapter 5, Liquid Column Gages (1942), $0.75

Part 3 — Temperature Measurement
- Chapter 1, General; Chapter 5, Pyrometric Cones; Chapter 6, Liquid-in-Glass Thermometers; and Chapter 7, Bourdon-Tube Thermometers (1931), $0.75
- Chapter 2, Radiation Pyrometers (1936), $0.55
- Chapter 3, Thermocouple Thermometers or Pyrometers (1940), $0.65
- Chapter 4, Resistance Thermometers (1945), $0.80
- Chapter 8, Optical Pyrometers (1940), $0.35

Part 4 — Head Measuring Apparatus (1933), $0.35
Part 5 — Chapter 4, Flow Measurement by Means of Standardized Nozzles and Orifice Plates (1940), $2.75
Part 6—Electrical Measurements (1934), $1.25
Part 8—Measurement of Indicated Horsepower (1941), $0.75
Part 9—Heat of Combustion (1943), $0.40
Part 11—Determination of Quality of Steam (1940), $0.45
Part 12—Measurement of Time (1942), $0.40
Part 13—Speed Measurements (1939), $0.45
Part 14—Linear Measurements (1936), $0.55
Part 15—Measurement of Surface Areas (1937), $0.75
Part 16—Density Determinations (1931), $0.30
Part 17—Determination of the Viscosity of Liquids (1931), $0.45
Part 18—Humidity Determinations (1932), $0.50
Part 20—Smoke-Density Determinations (1945), $0.65
Part 21—Leakage Measurements (1942), $0.60

RESEARCH

Fluid Meters:
- Part 1—Theory and Application (1937), $3.00
- Part 3—Selection and Installation (1933), $1.50

Report of the A.G.A.-A.S.M.E. Committee on Orifice Coefficients (1935), $2.75
Tests on Electrical Equipment for Drilling Rotary Drilled Oil Wells (1933), $0.85
Tests on Steam Equipment for Drilling Rotary Drilled Oil Wells (1932), $0.85
Bibliography on Aircraft Plywood (1944), $1.00
Bibliography on Management Literature and Supplement (1903–1935), $2.75
Bibliography on Marketing Research (1935), $1.00
Bibliography on the Cutting of Metals (published 1945), $6.50
Riveted Joints—A Critical Review of the Literature Covering Their Development, with Bibliography and Abstracts of the Most Important Articles (Published 1945), $4.50
Sources of Information on Instruments (published 1945), $0.75

SAFETY CODES

Safety Code for Cranes, Derricks, and Hoists (B30.2—1943), $1.50
Safety Code for Elevators (A17.1—1937 with 1942 Supplement), $1.25
Elevator Inspectors' Manual (A17.2—1945), $1.25
Safety Code for Jacks (B30.1—1943), $0.30
Safety Code for Mechanical Power-Transmission Apparatus (B15—1927), $0.35
Compressed-Air Machinery and Equipment (B19—1938), $0.30
Description and Schematic Layouts of Various Types of Undercar Safeties and Governors (published 1945), $0.40
Handling and Socketing of Wire Rope (published 1945), $0.25

Biographies

BIOGRAPHIES issued under the sponsorship of the A.S.M.E. Biography Committee are as follows:

Autobiography of an Engineer, by W. LeR. Emmet (1940), $3.50
Autobiography of John Fritz (1940), $3.25
Biography of Fred J. Miller (1941), $1.00
Life of Henry Laurence Gantt, by L. P. Alford (1934), $5.00

Books on Special Subjects

Automotive—Diesel Engine Cost and Performance Data (1943), $0.25
Corrosion-Resistant Metals (1936), $1.25
Creative Engineering (1944), $0.50
Design Data on Strength of Materials, Book 1 (1944), $1.50
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Engineering's Part in the Development of Civilization (1939), $1.50
Flow of Water in Pipes and Pipe Fittings (1941), $8.00
Forging of Steel Shells (1944), $0.75
Furnace-Performance Factors (1944), $1.00
General Discussion on Lubrication (1938) (no discount), $6.50
Graphitization of Steel Piping (1943–1944), $1.50
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Manual of Consulting Practice (1939), $0.40
Oil Engine Power Cost Report (1943), $1.25
Reflections on the Motive Power of Heat (1943), $2.75
Sixty Year Index to A.S.M.E. Technical Papers (1941), $3.75
Theoretical Steam Rate Tables (1937), $1.25

Regular Society Publications

*Mechanical Engineering**—Annual Subscription rate in United States $6.00; to Canada $6.75; elsewhere, $7.50
A.S.M.E. Transactions,* including *Journal of Applied Mechanics*, Annual subscription rate in United States, $12.00, elsewhere, $12.75
1946 A.S.M.E. Mechanical Catalog and Directory, $3.00 (sent gratis to members upon request)

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*Subscription price included in A.S.M.E. membership dues.

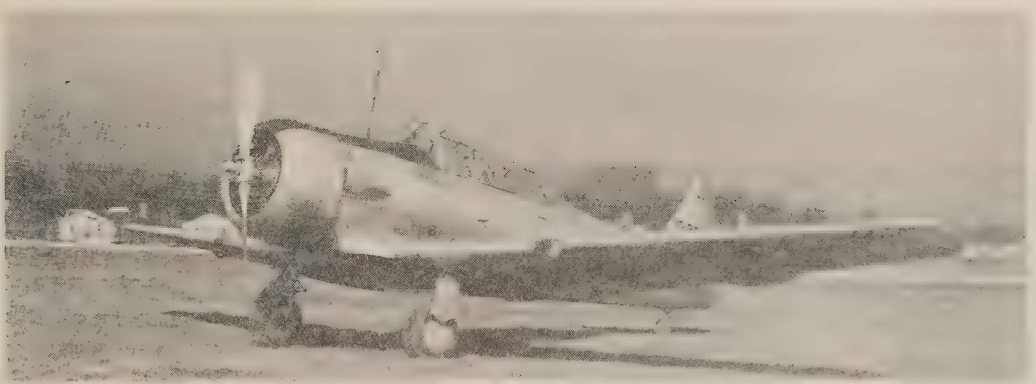

Fig. 1 North American AT-6C Airplane

The Practicability of Vibration Control in Light Aircraft

By DWIGHT C. KENNARD, JR.,[1] DAYTON, OHIO

The approach of the Air Technical Service Command to the problem of developing and evaluating a new engine-mount design is outlined, including a description of instrumentation, methods of vibration analysis, and presentation of natural-frequency formulas for elastically mounted power plants. The results of flight-vibration tests are presented for the standard engine mount, and an improved experimental engine mount installed in the North American AT-6C airplane. The results are discussed in detail showing the relative merits of the two engine installations. The test results show that the experimental engine mount, by providing natural frequencies of the mounted power plant which fall within the frequency range specified by the Air Technical Service Command, reduces appreciably the vibration of the engine and airplane structure.

Introduction

The development of successful antivibration mountings for aircraft radial engines of 1800 or more cu in. displacement has not been paralleled in the field of smaller aircraft engines. This development apparently has lagged owing to the understandable reluctance on the part of airplane manufacturers to accept what was considered to be a penalty in weight and cost in light airplane designs incorporating antivibration engine mountings. Pilots came to regard many of the light airplanes as uncomfortably "rough," and maintenance problems due to excessive vibration were looked upon as an inherent evil in such airplanes. However, the benefits of vibration control in airplanes equipped with larger engines was proving so successful that the Air Technical Service Command undertook a program of research and development directed toward attaining like benefits in light airplanes. Efforts were directed toward realizing an effective yet practical antivibration engine-mounting system for such light engine installations as are used in training, cargo, liaison, and other types of Army aircraft.

As a first step in this program, the Air Technical Service Command issued a design specification,[2] defining requirements for the antivibration provisions to be incorporated in an experimental engine mount for the Pratt and Whitney R-985 engine. Such a mount was subsequently procured[3] and thoroughly tested with an R-985 engine installed on a test stand. Upon satisfactory completion of these tests, the second step was to procure a similar engine mount for the Pratt and Whitney R-1340 engine to be installed in an AT-6C airplane.

This paper concerns the vibration tests which were conducted by the Air Technical Service Command on the AT-6C standard and experimental engine-mount installations in flight (1).[4]

Description of the AT-6C Airplane

The North American AT-6C airplane which was chosen for the test program is a two-place all-metal low-wing monoplane with monocoque fuselage and full cantilever wings, Fig. 1. The landing gear is the conventional type and is retractable. The tactical mission of the airplane is to provide a means for the advanced training of student pilots and their transition flying training to combat-type airplanes. The airplane is equipped

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[1] Aeronautical Engineer, Dynamics Branch, Aircraft Laboratory, Engineering Division, Air Technical Service Command, Wright Field. Jun. A.S.M.E.

Contributed by the Aviation Division of The American Society of Mechanical Engineers.

[2] This specification embodied the standard AAF requirements which restrict the natural frequencies of the mounted power plant to a lower limit of 400 cycles per min and an upper limit corresponding to 45 per cent normal rated engine rpm, or 75 per cent normal rated propeller rpm, whichever has the lower value.

[3] This engine mount was designed and manufactured by the United Aircraft Corporation, Pratt and Whitney Engine Division, and the Goodyear Tire and Rubber Corporation.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

with a Pratt and Whitney nine-cylinder single-row air-cooled radial R-1340-AN-1 engine, which drives directly a Hamilton Standard two-blade controllable-pitch constant-speed aluminum-alloy propeller, 9 ft diam. The engine has a power rating of 550 bhp at 2200 rpm and produces a maximum power for take-off amounting to 600 bhp at 2250 rpm.

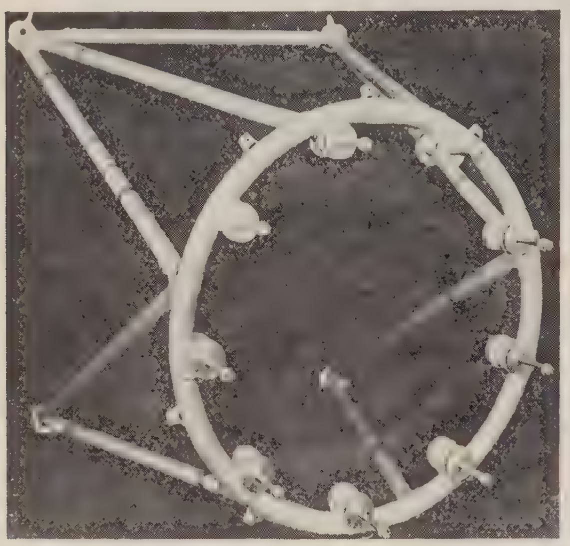


Fig. 2 Standard Engine-Mount Assembly for AT-6C Airplane

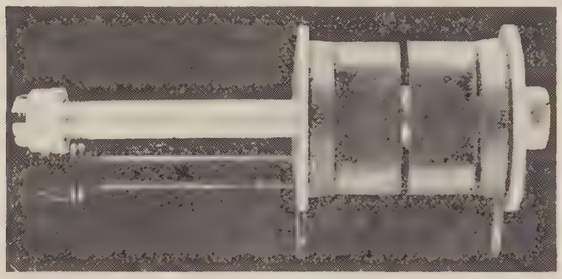


Fig. 3 Standard Engine Mounting Unit for AT-6C Airplane

Engine Mounts Tested

The engine mounting structure is the conventional welded-steel tubular-truss construction, embodying a circular ring, around the periphery of which are welded the engine attaching bosses. The nine equally spaced bosses of the standard mount are provided with elliptically shaped recesses which accommodate compression-type rubber blocks interposed between the mounting structure and the crankcase attaching bolts, Figs. 2 and 3. Experience has shown that this type of elastic mounting is not generally effective because the rubber affords insufficient elasticity when forced to act in compression.

The experimental mount tested utilizes six engine attaching points (attaching points behind cylinders Nos. 1, 4, and 7 are omitted). This arrangement provides symmetrical elastic restraint by virtue of the fact that three pairs of mounting units are equidistantly spaced about the periphery of the mounting ring, Fig. 4. The engine-mount bosses are set at an angle with circular shear type sandwich rubber bushings located on each side of the bosses. The bushings are much stiffer in a compressionwise direction than in the shear direction so that they act as directional springs tending to deflect mainly in the shear plane which is parallel to the bonded steel facing plates, Fig. 5. The bushings are held in place on each side of the mounting-ring bosses by forged angle bolts so that no metal-to-metal contact is allowed between the bolts and the bosses. The directional stiffness of the bushings is utilized by arranging the faces of the mounting-ring bosses to bear an angular relationship to the thrust line in such a way that the elastic center of the mount coincides with the center of gravity of the power-plant mass. This arrangement is termed "virtual center of gravity suspension" of the power plant. The outer bonded steel facings of the rubber bushings are eccentric with respect to each other so that when they are deflected by full engine torque, they become approximately concentric. The inner steel faces of the rubber bushings are locked to the mounting-ring bosses by mating dowel pins. The flanges of the inner faces also mate with the mounting-ring boss so as to effect a journal bearing. Shoulders on the forged angle bolt space the outer steel facings when the nut is tightened. Torque arms join adjacent angle bolts so that they cannot twist. Table 1 presents a weight breakdown for the two engine-mount assemblies tested, which shows that the experimental assembly entails an increased weight over the standard assembly amounting to only 4.5 lb.

Fig. 4 Experimental Engine-Mount Assembly for AT-6C Airplane

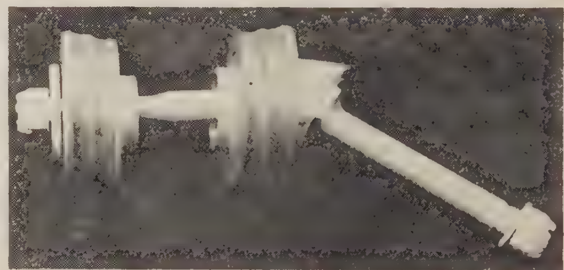


Fig. 5 Experimental Engine Mounting Unit for AT-6C Airplane

TABLE 1 WEIGHT BREAKDOWN FOR STANDARD AND EXPERIMENTAL MOUNTING ASSEMBLIES

	Weight, lb
Experimental mounting assembly, shown in Fig. 4	43.75
Standard mounting assembly, shown in Fig. 2	39.25
Experimental mounting units, total	9.48
Rubber sandwiches, 12 required	4.02
Angle bolts, 6 required	3.96
Nuts and washers	0.50
Torque arms, 3 required	1.00
Standard mounting units, total	5.09
Rubber bushings, 18 required	0.92
Engine mount bolts, 9 required	2.52
Nuts and washers	1.10
Bushings, 9 required	0.55

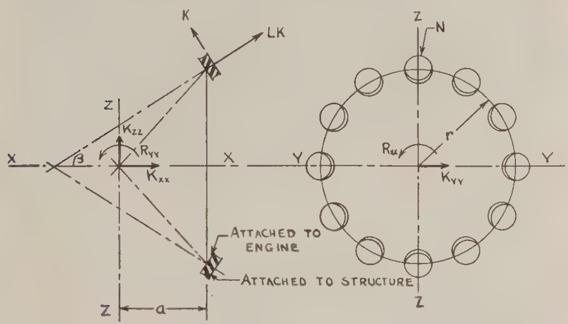


FIG. 6 SCHEMATIC ARRANGEMENT OF VIRTUAL-CENTER-OF-GRAVITY POWER-PLANT SUSPENSION

ESTIMATION OF POWER-PLANT NATURAL FREQUENCIES

It has been shown that the stiffnesses of a virtual-center-of-gravity type power-plant suspension may be expressed as follows, referring to Fig. 6 (2):

Roll about the crankshaft axis

$$R_{zz} = Nkr^2 \text{ in-lb per radian}$$

Pitching or yawing about the lateral or vertical axis

$$R_{yy} = R_{zz} = \frac{Nkr^2}{2}\left[\frac{a^2}{r^2} + L\left(\frac{a}{r}\sin\beta - \cos\beta\right)^2 + \left(\frac{a}{r}\cos\beta + \sin\beta\right)^2\right] \text{ in-lb per radian}$$

Thrustwise along the crankshaft axis

$$K_{zz} = Nk(\sin^2\beta + L\cos^2\beta) \text{ lb per in.}$$

Along the lateral or vertical axis

$$K_{yy} = K_{zz} = \frac{Nk}{2}\left[1 + (\cos^2\beta + L\sin^2\beta)\right] \text{ lb per in.}$$

where N = number of equally spaced mounting units
k = shear rate of each mounting unit, lb per in.
L = ratio of compressionwise rate to shear rate of each unit
r = radius of mounting units, in.
a = distance between plane of mounting units and virtual point of suspension, in.
β = angle made by the compression axis of mounting units with respect to thrust line, deg

It can be shown that the point of virtual suspension may be expressed as

$$a = \frac{1}{2}\,\frac{r\sin 2\beta}{\dfrac{2}{L-1} + \sin^2\beta} \text{ in.}$$

Neglecting the effect of gyroscopic forces and structural coupling, the natural frequencies may be determined by the well-known relationships

$$F = \frac{60}{2\pi}\sqrt{\frac{\text{Angular stiffness}}{\text{Moment of inertia}}} = \frac{60}{2\pi}\sqrt{\frac{R}{I}} \text{ cycles per min.}$$

or

$$F = \frac{60}{2\pi}\sqrt{\frac{\text{Translational stiffness}}{\text{Mass}}} = \frac{60}{2\pi}\sqrt{\frac{K}{M}} \text{ cycles per min.}$$

Applying these equations to the experimental mount under consideration

$N = 3 \times 2 = 6$
$k = 1250$ lb per in.
$L = 12.7$
$r = 12.2$ in.
$a = 10.75$ in.
$\beta = 35$ deg

Power-plant moment of inertia about the thrust axis (minus propeller inertia) = 240 lb-in. sec²
Power-plant moment of inertia about the vertical or lateral axis (including maximum propeller inertia) = 355 lb-in. sec²
Weight of power plant (including propeller) = 996 lb

Roll about the crankshaft axis
Stiffness = $6 \times 1250 \times 12.2^2$ = 1,120,000 in-lb per radian

$$\text{Natural frequency} = \frac{60}{2\pi}\sqrt{\frac{1{,}120{,}000}{240}} = 645 \text{ cycles per min}$$

Pitching or yawing about lateral or vertical axis

$$\text{Stiffness} = \frac{6 \times 12.2^2 \times 1250}{2}\left[\frac{10.75^2}{12.2^2} + 12.7\left(\frac{10.75}{12.2}\sin 35^\circ - \cos 35^\circ\right)^2 + \left(\frac{10.75}{12.2}\cos 35^\circ + \sin 35^\circ\right)^2\right]$$
$$= 1{,}170{,}000 \text{ in-lb per radian}$$

$$\text{Natural frequency} = \frac{60}{2\pi}\sqrt{\frac{1{,}170{,}000}{355}} = 547 \text{ cycles per min}$$

Thrustwise along the crankshaft axis
Stiffness = $6 \times 1250\,(\sin^2 35^\circ + 12.7\cos^2 35^\circ)$ = 66,500 lb per in.

$$\text{Natural frequency} = \frac{60}{2\pi}\sqrt{\frac{66{,}500}{996} \times 386} = 1520 \text{ cycles per min}$$

Laterally or vertically

$$\text{Stiffness} = \frac{6 \times 1250}{2}\left[1 + (\cos^2 35^\circ + 12.7\sin^2 35^\circ)\right]$$
$$= 21{,}750 \text{ lb per in.}$$

$$\text{Natural frequency} = \frac{60}{2\pi}\sqrt{\frac{21{,}750}{996} \times 386} = 876 \text{ cycles per min}$$

Distance of elastic center from plane of mounts

$$a = \frac{1}{2}\,\frac{12.2\sin 70^\circ}{\dfrac{2}{12.7-1} + \sin^2 35^\circ} = 11.5 \text{ in.}$$

Distance of elastic center ahead of center of gravity = 11.5 − 10.75 = 0.75 in.

INSTRUMENTATION FOR FLIGHT TESTS

The Air Technical Service Command has standardized on the use of Sperry-M.I.T. instrumentation for flight measurements of engine vibration and associated vibration of the aircraft structure. This equipment comprises velocity-type vibration pickups

which are fed into integrating amplifiers. The amplifier outputs actuate a four-element recording oscillograph, the traces of which indicate amplitudes which are proportional to vibratory displacement through a useful frequency range of 7 to 1500 cycles per sec.

The bridge network shown in Fig. 7 was used on the amplifier outputs so as to produce simultaneous traces on the four strings of the oscillograph which would correspond to addition, subtraction, and individual voltage outputs of a given pickup pair. The addition and subtraction traces indicated the translational and rotation components of power-plant vibratory motion.

Six pickups were suitably located to measure vibratory motion of the power plant in translation along and rotation about its three principal central axes as shown in Fig. 8. The pickup pair which measured power-plant vertical and pitching motion could not be located symmetrically with respect to the center of gravity. Hence a correction factor was applied to the vertical motion at the power-plant center of gravity. The pickup pair which measured lateral and rolling vibration was located in a plane 8 in. ahead of the center of gravity so that the lateral motion indicated contained components due to yawing vibration of the power plant about its center of gravity. A pickup was located in each cockpit to measure lateral vibration of the fuselage structure, Fig. 9. Selection of pickups was accomplished by means of a hand-operated rotary switch which fed the outputs of two pickups at a time into the amplifiers. Pickups were calibrated before and after tests by means of a mechanically driven calibrator which produces sinusoidal vibratory motion at a predetermined amplitude.

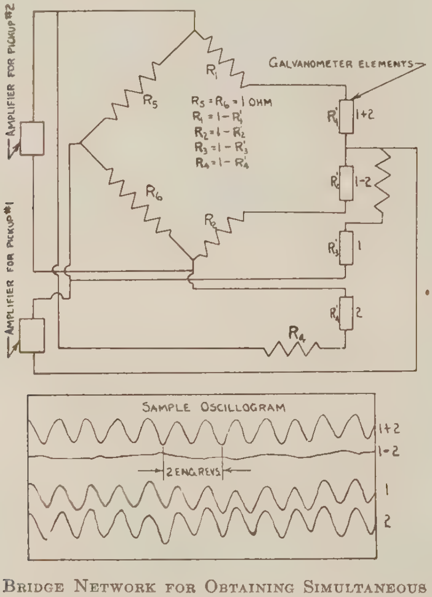
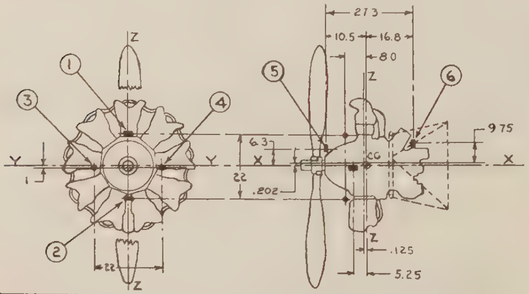


Fig. 7 Bridge Network for Obtaining Simultaneous Indication of Addition, Subtraction, and Individual Voltage Outputs of Paired Pickups

(Sample oscillogram first-order translational vibratory motion.)



Mode	Roll about X-X	Pitch about Y-Y	Yaw about Z-Z	Thrustwise along X-X	Lateral along Y-Y	Vertical along Z-Z
Pickups	1 − 2	5 − 6	3 − 4	3 + 4	1 + 2	5 + 6

Fig. 8 Pickup Locations and Modes of Power-Plant Vibration Measured

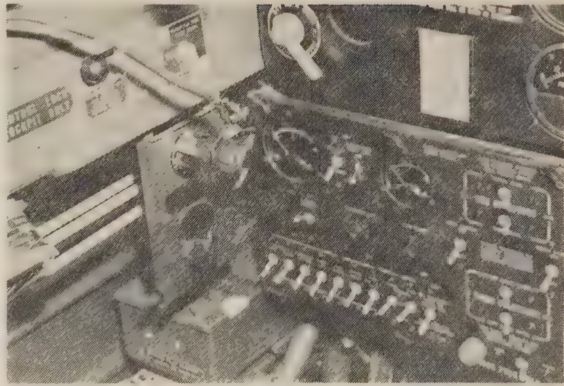


Fig. 9 Vibration Pickup Installation in Front Cockpit of AT-6C Airplane

In order to correlate the vibration frequencies with engine speeds, a pickup coil was installed on one of the spark-plug cables. The voltage induced in this coil was fed into the second stage of one of the amplifiers. This produced a mark superimposed on the vibration traces each instant the corresponding cylinder fired.

The amplifying and recording equipment was located in the rear cockpit where it could be controlled by the operator, Fig. 10. The amplifiers and oscillograph were powered by the 12-v electrical system of the airplane. The voltage input to the equipment was maintained at 12 v by means of a rheostat in series.

"Scratch plates" were installed on the three unused crankcase mounting bosses in the experimental engine-mount installation for the purpose of measuring over-all deflections of the mounted power plant. Stylii were held against the plates by means of springs in such a way that relative motion between the engine mount and crankcase resulted in scratches on the plates.

Test Procedure. The following tests were conducted in consecutive order:

1 Vibration records for the standard AT-6C engine-mount installation were taken in level flight at 8000 ft altitude throughout the operating range of engine speeds. Upon completion of this test, the standard engine mount was removed from the airplane and the experimental mount installed.

2 The test just outlined was repeated with the experimental engine-mount installation.

3 Vibration records for the experimental engine-mount installation were taken at various engine speeds while the airplane was in a 20-deg bank both to the right and to the left.

4 A series of ground tests were conducted for the purpose of checking the natural modes and frequencies of the power plant elastically suspended on the experimental mount. The natural mode in roll about the crankshaft axis was excited by running the engine on one magneto with one spark plug disconnected. This caused an excitation about the crankshaft axis at a frequency corresponding to one-half engine speed. The engine speed was

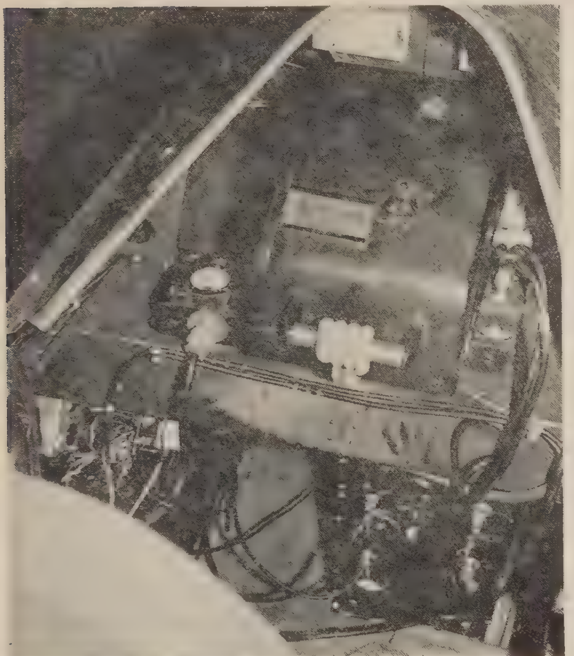


Fig. 10 Installation of Vibration-Recording Equipment in AT-6C Airplane

increased to 2200 rpm on both magnetos, at which speed the engine was switched over to the magneto having one spark plug disconnected. A vibration record of the rolling mode was then taken as the engine was throttled back to idling speed in a period of approximately 25 seconds. The spark plug was reconnected and sufficient masking tape was applied to a propeller blade to cause an unbalance of approximately 1.05 in-lb rotating at crankshaft speed. The engine speed was again raised to 2200 rpm and lowered to idling speed while vibration records of the pitching, vertical, yawing, and lateral modes were being taken.

5 A series of tests were conducted to determine the maximum movement of the engine relative to the supporting structure under various conditions by means of stylii bearing on scratch plates. The first condition investigated was that of cold idling of the engine. Stylii were allowed to bear on the scratch plates for approximately 1 min while the engine was idling in a cold condition. For the second condition, the stylii were allowed to bear on the plates while the engine was started, held at idling speed for a short time, and then shut down. The third condition consisted of an entire flight including starting, "warming up" the engine, taking off, climbing to an altitude of 8500 ft, making $4^1/_2$ turns of a right-hand spin, landing, and shutting off the engine. The stylii were allowed to bear against the plates for the entire flight.

6 In order to determine the durability of the experimental mount in service, the airplane was assigned to Headquarters, Air Technical Service Command, for postadministrative and flying duties.

Analysis of Vibration Records

Vibration records were prepared for harmonic analysis by enlarging to a 40-cm base line a section of each vibration trace included between two spark indications which mark an interval corresponding to four engine revolutions. The enlarged vibration traces were then analyzed for the five largest harmonics by means of a Coradi-type five-element rolling-sphere harmonic analyzer, Fig. 11.

The principle underlying the operation of the Coradi-type harmonic analyzer can be understood by considering the mathematical properties of the Fourier series (3). It can be shown that any periodic curve $f(t)$ can be expressed as a Fourier series, that is

$$f(t) = a_0 + a_1 \sin \omega t + a_2 \sin 2\omega t + \ldots + a_n \sin n\omega t \ldots$$
$$+ b_1 \cos \omega t + b_2 \cos 2\omega t + \ldots + b_n \cos n\omega t \ldots$$

where the function repeats itself in a period $T = \frac{2\pi}{\omega}$. The frequency of the fundamental or first harmonic is represented by ω and the frequency of the nth harmonic by $n\omega$. It can be shown, furthermore, that the amplitudes may be expressed as

$$a_n = \frac{\omega}{\pi} \int_0^{\frac{2\pi}{\omega}} f(t) \sin n\omega t dt$$

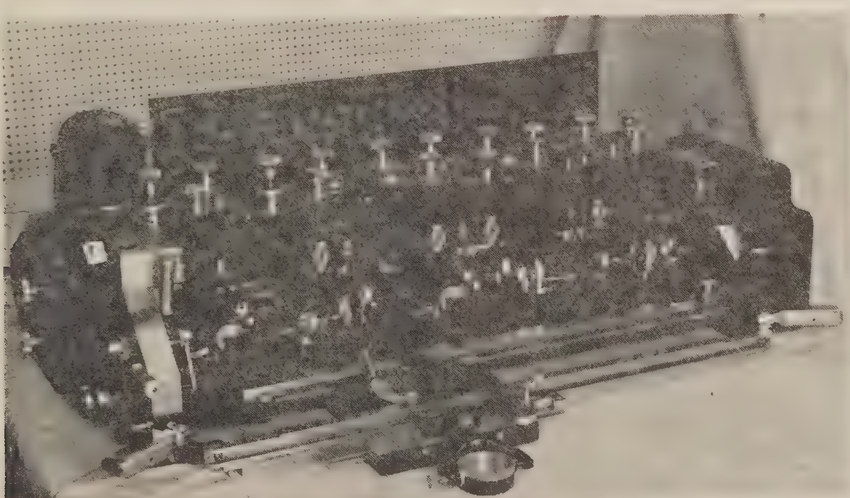


Fig. 11 Air Technical Service Command Five-Element Rollng Sphere Harmonic Analyzer

$$b_n = \frac{\omega}{\pi} \int_0^{\frac{2\pi}{\omega}} f(t) \cos n\omega t dt$$

These definite integrals can be evaluated for a particular harmonic by determining the area included between the harmonic curve and the neutral axis. The Coradi-type harmonic analyzer measures these areas by applying the principles utilized in the ordinary planimeter. Each integrator head is geared to revolve about a vertical axis n times as the curve follower moves on a lead screw across a span of 40 cm which represents the period of the first harmonic $\frac{2\pi}{\omega}$. On each integrator head are two planimeter wheels in vertical planes at right angles to each other, Fig. 12.

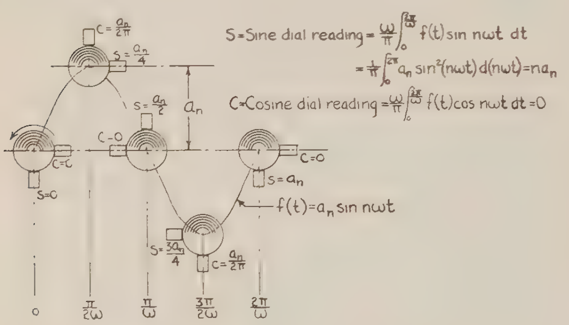


FIG. 12 ILLUSTRATION OF THE WAY IN WHICH THE CORADI-TYPE HARMONIC ANALYZER OPERATES UPON THE PERIODIC CURVE $f(t) = a_n \sin n\omega t$

(The sphere makes one complete revolution about a vertical axis in the period included between 0 and $\frac{2\pi}{\omega}$. At the end of the revolution, the sine dial registers a_n and the cosine dial registers 0, hence the amplitude $A_n = \sqrt{a_n^2 + b_n^2} = a_n$. In n periods the sine dial will read na_n.)

The planimeter wheels bear against a ground-glass sphere. The sphere in turn rides on a wheel which revolves in accordance with back-and-forth motion of the entire machine. The machine is constrained to back-and-forth motion only, and this motion is controlled by the operator in keeping the curve follower on the vibration trace as the lead screw advances the follower across the carriage. The speed of the lead screw can be controlled by a foot pedal which varies the speed of the driving motor. The planimeter wheels are calibrated to read Σa_n and Σb_n. In order to determine a_n and b_n, the planimeter readings must be divided by the harmonic being extracted, that is

$$a_n = \frac{\Sigma a_n}{n} \quad \text{and} \quad b_n = \frac{\Sigma b_n}{n}$$

The resultant amplitude is

$$A_n = \sqrt{a_n^2 + b_n^2}$$

and the phase angle may be found as

$$\tan \phi_n = \frac{b_n}{a_n}$$

The property of the analyzer which enables it to exclude all harmonics other than the nth harmonic being extracted can be understood by considering a harmonic amplitude h_0 having a frequency $m\omega$. Then

$$f(t) = h_0 \sin (m\omega t + \phi)$$

and

$$a_n = \frac{\omega}{\pi} \int_0^{\frac{2\pi}{\omega}} h_0 \sin (m\omega t + \phi) \sin n\omega t dt$$

but

$$\sin (m\omega t + \phi) = \sin m\omega t \cos \phi + \cos m\omega t \sin \phi$$

$$a_n = \frac{\omega h_0 \cos \phi}{\pi} \int_0^{\frac{2\pi}{\omega}} \sin m\omega t \sin n\omega t dt + \frac{\omega h_0 \sin \phi}{\pi} \int_0^{\frac{2\pi}{\omega}} \cos m\omega t \sin n\omega t dt = 0$$

Similarly $b_n = 0$, showing that the planimeter dials produce a zero reading over the complete cycle except for the nth harmonic.

DISCUSSION OF TEST RESULTS

The test results presented in Table 2 indicate that the experimental engine mount was very effective in attenuating that vibratory response of the power plant due to first-order forcing functions. These forcing functions, which correspond to the rotational speed of the engine crankshaft, arise from the inherent inertia forces of the articulated connecting-rod system, gas-pressure forces in the cylinders, and static, dynamic, and aerodynamic unbalances in the directly driven propeller.

On the other hand, the natural frequencies of the power plant on the standard mount were in a range where they could be excited by the first-order forcing functions, that is, the inertia and gas-pressure forces which produce alternating first-order couples about the crankshaft axis, induced resonance of the power plant on the standard mount at 1440 rpm. Inertia forces and propeller unbalances excited angular and translational modes about and along the vertical and lateral axes of the power plant. Resonances occurred in pitch about the lateral axis at 1460 and 2020 engine rpm. These peaks were reflected slightly in the measured vertical vibration and thus indicate appreciable cou-

TABLE 2 FIRST-ORDER POWER-PLANT VIBRATION FOR STANDARD AND EXPERIMENTAL MOUNTINGS

Nominal engine speed settings, rpm	Roll about X-X		Pitch about Y-Y		Yaw about Z-Z		Thrustwise along X-X		Lateral along Y-Y		Vertical along Z-Z	
	Std	Exp	Std	Exp	Std	Exp	Std	Exp	Std	Exp	Std	Exp
1100	0.0035	....	0.0045	0.0080	0.0055	....	0	....	0.0025	....	0.0020	....
1200	0.0115	0.0070	0.0070	0.0095	0.0055	0.0095	0.0005	0.0010	0.0045	0.0025	0.0010	0.0010
1300	0.0315	0.0065	0.0090	0.0090	0.0100	0.0135	0.0005	0.0010	0.0070	0.0035	0.0010	0.0010
1400	0.0375	0.0060	0.0300	0.0110	0.0245	0.0055	0.0025	0.0005	0.0040	0.0025	0.0030	0.0005
1500	0.0320	0.0090	0.0165	0.0125	0.0445	0.0045	0.0035	0.0005	0.0030	0.0025	0.0035	0.0015
1600	0.0280	....	0.0160	0.0120	0.0380	0.0085	0.0025	0.0005	0.0040	....	0.0035	0.0010
1700	0.0240	0.0085	0.0080	0.0050	0.0240	0.0115	0.0015	0.0010	0.0040	0.0030	0.0020	0.0010
1800	0.0150	0.0085	0.0120	0.0065	0.0150	0.0150	0.0015	0.0020	0.0040	0.0030	0.0020	0.0010
1900	0.0125	0.0100	0.0110	0.0055	0.0250	0.0140	0.0020	0.0010	0.0040	0.0030	0	0.0010
2000	0.0165	0.0100	0.0270	0.0055	0.0220	0.0155	0.0010	0.0015	0.0045	0.0030	0.0025	0.0010
2100	0.0110	0.0090	0.0135	0.0035	0.0235	0.0155	0.0010	0.0010	0.0040	0.0030	0.0005	0.0010
2200	0.0115	0.0090	0.0165	0.0080	0.0175	0.0180	0.0005	0.0005	0.0050	0.0035	0.0010	0.0010
2250	0.0145	0.0100	0.0155	0.0080	0.0110	0.0105	....	0.0010	0.0040	0.0030	0.0015	0.0010

NOTE: Double amplitudes in degrees for rotation, in inches for translation.

pling between the two modes. A resonance also occurred in yaw about the vertical axis at 1540 engine rpm which exhibited slight coupling with thrustwise vibration. A first-order lateral resonance of the power plant was measured at 1320 engine rpm.

The experimental mount eliminated all first-order resonances experienced with the standard mount. Reductions of power-plant response were attained amounting to as much as 76 per cent in roll about the crankshaft axis; 58 per cent in pitch about the lateral axis; and 84 per cent in yaw about the vertical axis. First-order lateral motion in the front cockpit also was reduced 40 to 75 per cent throughout the operating range of engine speeds. Lateral motion in the rear cockpit was not reduced but was very small for both engine-mount installations (0.0025 in. or less).

Since the natural frequencies of the power plant on the standard mount were well above the range where they can be excited by one-half order forcing functions (arising from irregularities in ignition and air-fuel induction) the corresponding power-plant responses indicated were largely due to excitation of a combined bending-torsion mode of the fuselage. This was confirmed by a lateral response in the rear cockpit which corresponded to the power-plant yawing response about the vertical axis at 2040 rpm (1020 cycles per min). A variation of this mode apparently was responsible for the magnified vertical power-plant response, coupled with roll about the crankshaft axis at 1780 to 1830 rpm (890 to 915 cycles per min).

The experimental mount did not alter appreciably the magnitude of one-half order power-plant response. Lateral response of the rear cockpit followed very closely that measured for the standard mount.

First-order crankshaft unbalance forces are largely compensated by the counterweight in a radial engine but this compensation introduces transverse second-order forces of appreciable magnitude because the center of mass of the rotating and articulating system travels in an oval rather than in a circular path, as does the center of mass of the counterweight. The frequency of the second-order forces is above the natural frequencies of the mounted power plant by a sufficient amount to result in approximately the same power-plant and cockpit response for either the standard or the experimental mount. Since the second-order unbalance forces act in the transverse plane, a comparatively low magnitude of vibration was measured in all modes of power-plant motion other than lateral or vertical. The second-order response of the cockpits probably was largely due to propeller-blade-passage impulses in the slip stream which impinged upon the canopy.

Because the moment of inertia of the power plant about a given transverse axis varies from a maximum to a minimum twice during each revolution of a two-blade propeller, second-order gyroscopic couples result during a bank and turn due to rotation of the airplane about its vertical axis. In order to check this effect for the experimental engine-mount installation, vibration records were taken at various engine speeds during a 20-deg bank and turn both to the right and to the left. Examination of the records revealed no important increase of second-order response of the power plant or the cockpit structure.

Two $3^1/_2$-order peak responses were indicated in the angular modes about the vertical and lateral axes at 1430 rpm (5000 cycles per min) and 1830 rpm (6410 cycles per min) which were the result of so called "conical whirls." In this particular case the whirls were largely independent of the dynamics of both engine mounts because they were generated by $4^1/_2$-order (frequency of cylinder firing) torsional vibration of the crankshaft combined with crankshaft bending. This mode of crankshaft vibration energized a propeller-bending mode which was symmetrical in the plane of rotation but asymmetrical perpendicular to the plane of rotation, that is, the propeller tips were moving out of phase in a thrustwise direction but in phase in the transverse plane. In such a condition, transverse reactions are imposed upon the thrust bearing due to the symmetrical vibration of the propeller blades in the plane of rotation, and moments about transverse axes are imposed upon the propeller shaft due to asymmetrical propeller-blade vibration perpendicular to the plane of rotation. These forces and moments cause a conical whirl of the engine at a frequency of $\left(n - \frac{P}{N}\right)$, where n is the harmonic order of propeller vibration, N is the engine speed, and P is the propeller speed (4).

In the case under consideration, the whirling frequency was $3^1/_2$-order ($4^1/_2$—1) although $4^1/_2$-order vibration of the power plant was negligible. This mode was also apparent in the indication of the lateral pickups which were located in a plane ahead of the power-plant center of gravity so that they responded to a yawing component of motion. Since there was no $3^1/_2$-order response vertically, it is apparent that the power plant underwent a conical whirl with a node in the vicinity of the power-plant center of gravity.

The engine run-down tests with one cylinder not firing, as conducted on the experimental engine mount, indicated that the power-plant natural frequency in roll about the crankshaft axis was 680 cycles per min (645 cycles per min calculated). The natural frequencies excited by unbalancing the propeller 1.05 in. lb were 850 cycles per min for the vertical-pitching mode and 865 cycles per min for the lateral-yawing mode. The calculated natural frequency for the vertical and lateral modes was 875 cycles per min and for the pitching and yawing modes, 547 cycles per min. The engine could not be run slow enough to produce a satisfactory excitation of the pitching-yawing natural modes, however. The foregoing natural frequencies are sufficiently low to provide adequate vibration isolation for the important forcing functions. Resonance with one-half order couples about the crankshaft axis is possible at 1360 rpm (680 cycles per min), but under the conditions of the flight tests, the response indicated was small.

The results of scratch-plate records indicating relative motion between the crankcase and mounting ring for the experimental mount are given in Table 3. The mounting elements are manufactured 0.140 in. eccentric in order to offset engine torque. The scratch-plate records show a deflection of 0.156 in. at take-off power. This means that full engine torque deflects the mounting elements 0.016 in. beyond the concentric configuration which is not considered to be excessive. No metal-to-metal contact in the mounts is likely to occur.

TABLE 3 SCRATCH-PLATE RECORDS OF RELATIVE MOTION BETWEEN CRANKCASE AND MOUNTING RING

Condition	Indicated deflection, in. Right	Left	Top	Remarks
Engine idling prior to warm-up	$^1/_8$	$^1/_8$	—	Deflection on clockwise side of neutral position due to torque reaction
Engine started, idled at 550 rpm, shut off	$^9/_{64}$	$^{13}/_{64}$	—	
Engine started, airplane taken off and climbed to 8500 ft, $4^1/_2$ turns of a spin made, airplane landed and engine shut off	$^5/_{32}$	$^5/_{32}$	$^5/_{32}$	Deflection due to torque reaction at take-off power
	$^1/_{16}$	$^1/_{16}$	$^1/_{16}$	Diameter of circular trace resulting from spin

Conclusion

The principles embodied in the design of the experimental engine mount herein described are applicable for the control of vibration in many types of aircraft where lightweight and simplicity of design are important factors. Vibration control need be limited no longer to the larger aircraft-engine installations.

On the contrary, development of this type of engine mount suggests new opportunities for attaining greater flying comfort and added serviceability in light airplanes without attendant design penalties. It will be an increased pleasure to fly in the light airplanes of tomorrow!

Acknowledgment

The writer is indebted to Ensign Paul V. Roberts for his valuable assistance in carrying out the flight-test program and the reduction of data which form the basis of this paper.

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(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Gasoline Explosion Pressures

By M. S. PLESSET[1] and F. R. GILMORE,[2] SANTA MONICA, CALIF.

A numerical method is outlined which permits the calculation of the temperatures and pressures obtained when mixtures of air and aviation-gasoline vapor are exploded in a closed chamber. The calculations use basic thermodynamic data to determine the final equilibrium state of the air and gasoline mixtures after combustion. Similar calculations have been carried out previously by other investigators, but only for a limited range of mixture ratios. Experimental measurements of the explosion pressures were made using an explosion bomb with a strain-gage pressure pickup. The measured pressures were of the same order of magnitude as the calculated pressures, but in general were somewhat above the theoretical values. Some possible explanations for this trend are discussed in the text.

Introduction

The investigation covered by this paper was initiated so that some conception could be formed of the temperatures and pressures developed in aviation combustion heaters under the assumption that an explosion takes place. No general solution of the problem applicable to all mixture ratios was found so that an analysis based on fundamental physico-chemical principles was necessary.

An estimate was also desired of the ignition limits of gasoline-air mixtures at various temperatures and pressures. These limits have an important bearing on the problem of combustion-heater ignition.

Theoretical Considerations

The pressures which are produced when a mixture of air and fuel vapor explodes in a closed chamber can be calculated from basic chemical principles, provided the assumption is made that all chemical reactions reach equilibrium at the high explosion temperature, and provided that sufficient thermodynamic data on the reaction products are available.

If aviation gasoline is assumed to have the average molecular composition, C_8H_{16}, the combustion of 1 mole of a gasoline-air mixture can be represented by the following total reaction:

$$X\, C_8H_{16} + 0.2095\,(1-X)\, O_2 + 0.7809\,(1-X)\, N_2 + 0.0093\,(1-X)\, A + 0.0003\,(1-X)\, CO_2 \rightarrow a\, CO_2 + b\, CO + c\, H_2O + d\, H_2 + e\, O_2 + f\, N_2 + g\, NO + h\, NO_2 + i\, CH_4 + j\, OH + k\, H + m\, O + n\, O_3 + q\, C\,(\text{graphite}) + r\, C_2\,(\text{gas}) + s\, C(\text{gas}) + t\, NH_3 + 0.0093\,(1-X)\, A$$

In this X is the mole fraction of gasoline vapor present before the explosion. The composition of dry air, according to Humphreys (1)[3] is used. Although argon does not take part in the chemical reaction, it has been included in the equation because its thermal capacity enters into the heat-balance equation. The seventeen letters a to t represent unknown coefficients which are to be determined. Some possible products of the reaction, such as CH and CH_2, have been omitted because thermodynamic data on these compounds are not available; it is believed that this omission should not appreciably affect the results. Other possible products, such as C_2H_6, C_2N_2, etc., can be shown to be entirely negligible.

The requirement that the foregoing reaction equation be balanced yields four equations for the unknown coefficients

$$a + b + i + q + 2r + s = 8X + 0.0003\,(1-X) \quad [1]$$

$$2c + 2d + 4i + j + k + 3t = 16X \quad [2]$$

$$2a + b + c + 2e + g + 2h + j + m + 3n = 0.4196\,(1-X) \quad [3]$$

$$2f + g + h + t = 1.5618\,(1-X) \quad [4]$$

The remaining thirteen equations necessary for the evaluation of the seventeen constants are derived from a consideration of the various reactions which are assumed to be in equilibrium at the explosion temperature; for example, the carbon monoxide can react with the oxygen to form carbon dioxide

$$CO + {}^1/_2O_2 \rightleftharpoons CO_2$$

For this reaction, there is a certain equilibrium constant

$$K_1 = \frac{p_{CO}\sqrt{p_{O_2}}}{p_{CO_2}}$$

where p_{CO}, p_{O_2}, and p_{CO_2} are the partial pressures of the carbon monoxide, oxygen, and carbon dioxide, respectively. If the explosion has taken place at constant volume, then b moles of carbon monoxide, e moles of oxygen, and a moles of carbon dioxide are occupying the same volume at the explosion temperature T_m that one mole of the mixed gases occupied at the initial temperature T_0 and pressure P_0. From the perfect-gas law, it follows that

$$p_{CO} = b\,\frac{T_m}{T_0}\,P_0$$

$$p_{O_2} = e\,\frac{T_m}{T_0}\,P_0$$

$$p_{CO_2} = a\,\frac{T_m}{T_0}\,P_0$$

The expression for the equilibrium constant then becomes

$$K_1 = \frac{p_{CO}\sqrt{p_{O_2}}}{p_{CO_2}} = \frac{b}{a}\sqrt{e\left[\frac{T_m}{T_0}\,P_0\right]} \quad [5]$$

In a similar manner, twelve other equations can be derived

$$K_2 = \frac{p_{H_2}\sqrt{p_{O_2}}}{p_{H_2O}} = \frac{d}{c}\sqrt{e\left[\frac{T_m}{T_0}\,P_0\right]} \quad [6]$$

$$K_3 = \frac{p_{NO}}{\sqrt{p_{N_2}p_{O_2}}} = \frac{g}{\sqrt{fe}} \quad [7]$$

[1] Analytical Group Engineer, Douglas Research Laboratories, Douglas Aircraft Company.

[2] Analytical Group, Douglas Research Laboratories, Douglas Aircraft Company.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

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$$K_4 = \frac{p_{NO}\sqrt{p_{O_2}}}{p_{NO_2}} = \frac{g}{h}\sqrt{e\left[\frac{T_m}{T_0}P_0\right]} \quad [8]$$

$$K_5 = \frac{\sqrt{p_{H_2}{}^3 p_{N_2}}}{p_{NH_3}} = \frac{\sqrt{d^3 f}}{t}\left[\frac{T_m}{T_0}P_0\right] \quad [9]$$

$$K_6 = \frac{p_{OH}\sqrt{p_{H_2}}}{p_{H_2O}} = \frac{j}{c}\sqrt{d\left[\frac{T_m}{T_0}P_0\right]} \quad [10]$$

$$K_7 = \frac{p_H{}^2}{p_{H_2}} = \frac{k^2}{d}\left[\frac{T_m}{T_0}P_0\right] \quad [11]$$

$$K_8 = \frac{p_O{}^2}{p_{O_2}} = \frac{m^2}{e}\left[\frac{T_m}{T_0}P_0\right] \quad [12]$$

$$K_9 = \frac{p_{O_3}}{\sqrt{p_{O_2}{}^3}} = \frac{n}{\sqrt{e^3}}\sqrt{\frac{T_0}{T_m P_0}} \quad [13]$$

$$K_{10} = \frac{\sqrt{p_{O_2}}}{p_{CO}} = \frac{1}{b}\sqrt{e\left[\frac{T_0}{T_m P_0}\right]},\ (q > 0) \quad [14]$$

$$K_{11} = \sqrt{p_{C_2\,(gas)}} = \sqrt{r\left[\frac{T_m}{T_0}P_0\right]},\ (q > 0) \quad [15]$$

$$K_{12} = \frac{p_{H_2}{}^2}{p_{CH_4}} = \frac{d^2}{i}\left[\frac{T_m}{T_0}P_0\right],\ (q > 0) \quad [16]$$

$$K_{13} = p_{C(gas)} = s\left[\frac{T_m}{T_0}P_0\right],\ (q > 0) \quad [17]$$

The values of these equilibrium constants at various temperatures are given in Appendix 1 (Table 3). A sufficient number of algebraic relations have now been obtained to permit the calculation of the amount of each gas present after combustion, provided the temperature can be determined.

The explosion temperature T_m is calculated by equating the energy of combustion of the fuel at constant volume to the energy required to heat the products of combustion to the temperature T_m plus the energy required to cause the dissociation which occurs at this temperature. The energy of complete combustion of 100-octane gasoline vapor can be taken to be 18,700 Btu per lb (for the liquid) plus 140 Btu per lb (energy of vaporization; see Appendix 1). The internal energies of various gases at different temperatures, using as base values (zero internal energy) the energy of CO_2, H_2O, O_2, and N_2, and A, are given in Table 2.

The solution of these relations to obtain the explosion temperature and pressure involves successive approximation. For a particular mole-fraction of fuel vapor X, and particular initial temperature and pressure T_0 and P_0, a value for the explosion temperature T_m is guessed, and the values for the equilibrium constants, $K_1, \ldots, K_{13}$, at the temperature T_m are interpolated from the table in Appendix 1. The values for T_0, T_m, P_0, $K_1, \ldots, K_{13}$ can now be substituted in the equilibrium equations. These equations are then solved for the unknown coefficients, $a, \ldots, t$, by the method of successive approximation. Thus, the chemical composition of the mixture after combustion has been calculated. The energy balance is then checked by comparing the energy evolved in the complete combustion of the fuel with the energy change associated with the heating of the reaction products from T_0 to T_m accompanied by the change in chemical composition from CO_2, H_2O, N_2, and O_2 to the composition just calculated. If the energy does not balance, a new value of T_m must be estimated and the whole process repeated.

After the temperature and composition of the mixture after explosion are found in this manner, the explosion pressure is readily calculated from the formula

$$P_m = P_0 N_m \frac{T_m}{T_0} \quad [18]$$

where N_m is the number of final moles of gas per unit initial mole and is therefore equal to $a + b + c + \ldots + t$.

An example of the calculation of explosion temperatures and pressures by this method is given in Appendix 2.

For five particular mixture ratios, corresponding to five values of X, thermodynamic charts were available which greatly simplified the problem of calculating the explosion temperatures and pressures. These charts are not reproduced here because they are as yet restricted in distribution.

Apparatus and Tests

In order to check the calculations of explosion pressures experimentally, a spherical explosion bomb was constructed by modifying a 9-in. aluminum pressure accumulator. A pair of strain gages was cemented to the outside surface of the bomb. The pressure pickup system is shown in Fig. 1, and a block diagram of the electrical circuit is given in Fig. 2.

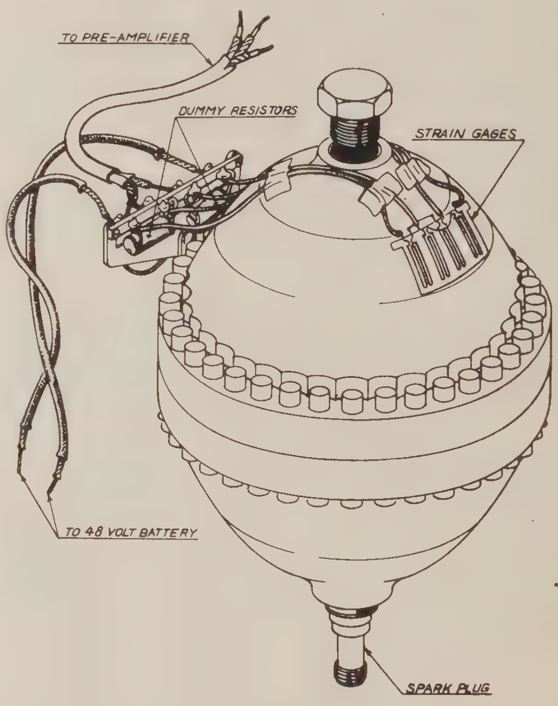

Fig. 1 Explosion Bomb and Pressure Pickup

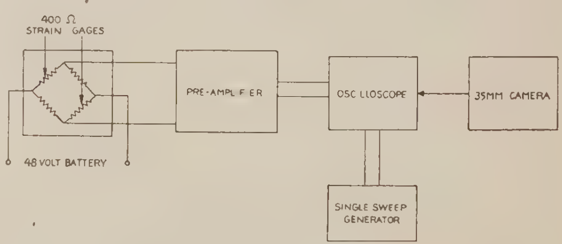

Fig. 2 Electrical Circuit of the Pressure Pickup

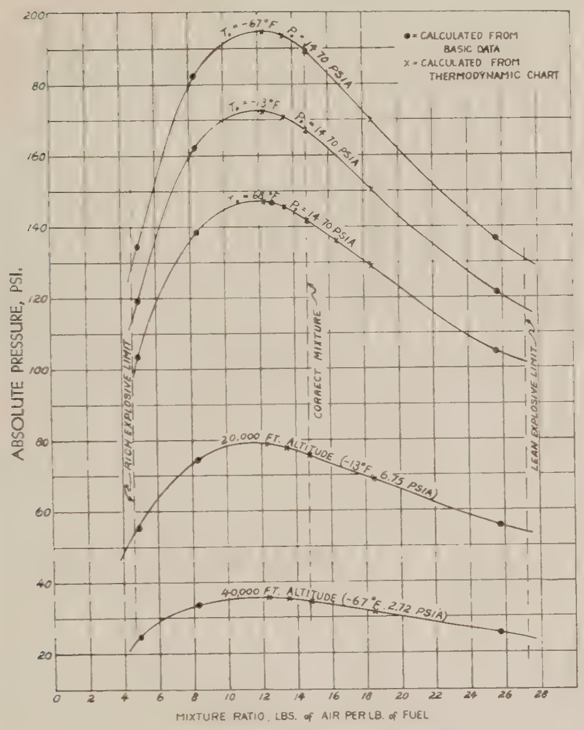

Fig. 3 Theoretical Explosion Pressures

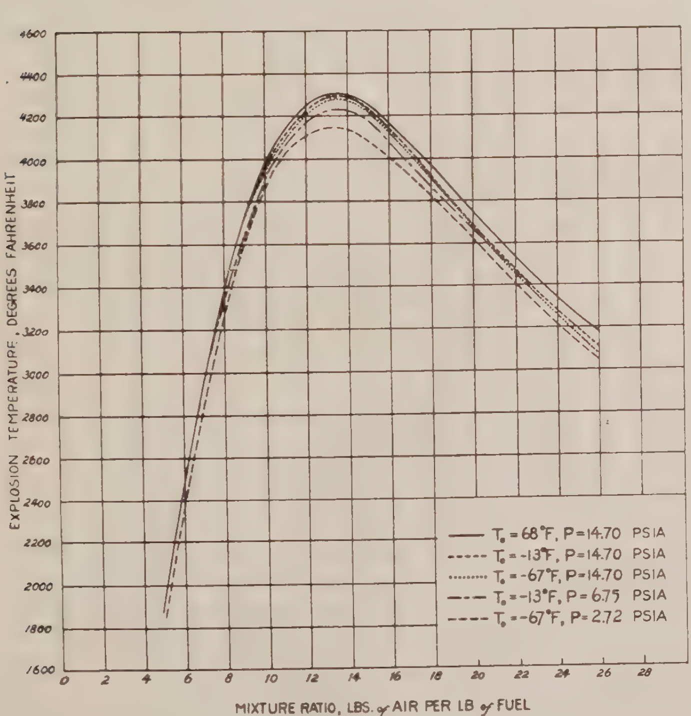

Fig. 4 Theoretical Explosion Temperatures

The two strain gages used to measure the pressure were made opposite legs of a balanced Wheatstone bridge. Pressure inside the bomb caused the surface of the bomb, and hence the gages, to be strained, which increased the resistance of the gages and produced a voltage difference across the vertical plates of an oscilloscope. A single-sweep voltage pulse was simultaneously applied to the horizontal plates.

In the tests, the bomb was charged by injecting measured amounts of 100-octane gasoline into it. The bomb was agitated manually, then allowed to stand for 15 or 20 min to permit complete vaporization. The pressure of the gasoline-air mixture immediately before the explosion was therefore 1 atm plus the partial pressure of the gasoline vapor. The mixture was exploded by a spark, and the pressure rise was shown visually by the oscilloscope. The oscilloscope trace was simultaneously photographed. To determine the actual explosion pressure, the oscilloscope trace deflection shown on the photographic film was measured and compared with the deflections produced by charging the bomb with compressed air under known pressures.

Results Obtained

Theoretical. The temperatures and pressures that should result when various mixtures of air and 100-octane gasoline are exploded in a closed chamber, at various initial temperatures and pressures, have been calculated by the method described in the section discussing theory. Sample calculations are given in Appendix 2. The results are presented graphically in Figs. 3 and 4.

The results have been related to the conventional mixture ratio M_R, which is defined as the ratio of the weight of air to the weight of gasoline vapor. It should be noted that

$$M_R = \frac{\text{Vol per cent of air}}{\text{Vol per cent of gasoline vapor}} \times \frac{28.98}{112.2} = 0.2583\,(1 - X)/X$$

Experimental. The results of the explosion pressure measurements made with the spherical bomb are given in Table 1. These results are also compared graphically with the theoretical curve in Fig. 5.

Discussion of Results

Theoretical. It can be seen from Figs. 3 and 4, that very good agreement is obtained between the algebraic method of calculation developed in this paper and the method based on thermodynamic charts.

Except for very rich mixtures, these theoretical calculations should be quite accurate. The gasoline has been rather arbitrarily assigned the average composition C_8H_{16}, which is equivalent to specifying a hydrogen-carbon ratio of 2, and an average molecular weight of 112. The hydrogen-carbon ratios for aviation gasolines are known to be very close to 2; moreover, a considerable variation of the ratio affects the calculated pressures only slightly (for example, use of the charts of reference (2) calculated for a hydrogen-carbon ratio of 2.25, gives for M_R = 14.76 a pressure 0.7 per cent lower). The molecular weight enters into the problem only in the determination of the amount of air displaced by the fuel vapor. A 10 per cent error in the estimated molecular weight would change the final results by only 0.5 per cent for X = 0.05.

TABLE 1 EXPERIMENTAL MEASUREMENTS OF EXPLOSION PRESSURES

M_R	Time Allowed for Vaporiz. min.	T_o °F	Time, Spark to Max. Press. sec *	P_o (calc.) psia**	ΔP psi	P_m psia
4.6	16	70	0.15	15.5	158	174
4.6	15	72	0.22	"	130	146
5.2	15	91	0.40	15.4	112	127
5.2	15	91	0.65	"	102	117
5.2	15	87	0.35	"	107	112
5.2	975	41	0.15	"	165	180
6	17	66	0.12	15.3	168	183
6	18	70	0.12	"	162	177
6	17	49	0.22	"	146	161
6	26	56	0.12	"	141	156
6	16	61	0.12	"	146	161
7	16	70	0.09	15.2	170	185
7	16	70	0.09	"	170	185
7	16		0.09	"	165	180
8	15	70	0.11	"	130	145
8	15		0.13	"	187	202 (1†)
8	25		0.12	"	182	197 (2)
11	16	80	0.09	15.1	168	183
11	17	84	0.09	"	168	183
11	16	87	0.09	"	162	177
15	20	90	0.12	14.9	161	176 (3)
15	15	92	0.12	"	153	168 (4)
15	15	92	0.17	"	147	162 (5)
18	15	86	0.21	"	120	135
18	17	80	0.23	"	112	127
18	17	80	0.26	"	112	127
18	15	75	0.23	"	112	127
18	15	75	0.35	"	95	110
22	27	84	0.40	"	103	118
22	16	84	0.45	"	95	110

* Tabulated times may all be short by about the same amount.
** Initial pressure equals one atmosphere plus the partial pressure of the gasoline vapor.
† Numbers indicate first five explosions in the bomb, showing decrease of pressure readings as inside of bomb becomes coated with carbon and other explosion products.

The error due to inaccuracies in the values for the internal energy of the gases (Appendix 1) should be less than 0.5 per cent. Inaccuracies in the values of the equilibrium constants would probably affect the results by less than 1 per cent. The possible errors caused by nonequilibrium conditions, radiation losses, and other minor effects[4] are presumably small, since calculations by Lewis and von Elbe[5] for explosion pressures of lighter gases (H_2, CO, C_2H_2) agree with the experimentally measured pressures within 3 per cent. (These lighter gases contain the same elements as gasoline, and the composition of the hot burned mixture depends only on the elements present and the temperature, not on the original compounds present.)

When very rich mixtures are exploded, radicals such as CH and CH_2 can be expected to take part in the reaction. No thermodynamic data on such radicals are available, so their presence has been perforce neglected in the calculations for this paper. The error thus introduced is probably not large, since the calculations show that the related molecule CH_4 is present only in a minor amount. However, at the lower explosion temperatures which occur with rich mixtures, the assumption that all chemical reactions reach equilibrium may be in appreciable error. Moreover, the presence of finely divided free carbon may greatly increase the radiation losses. For these reasons, that part of the theoretical curves to the left of $M_R = 8$ should be considered as approximate only.

Experimental. A graphical comparison of the explosion-pressure calculations with the experimental measurements is presented in Fig. 5. As was explained in the "Apparatus and Tests"

[4] Such as the nonuniform final temperature distribution (reference (3), pp. 167, 175–176, 293–299), and the "excitation lag" due to the greater rapidity with which energy flows into the translational and rotational rather than the vibrational degrees of freedom (reference (3), p. 306).

[5] Reference (3), pp. 301–308.

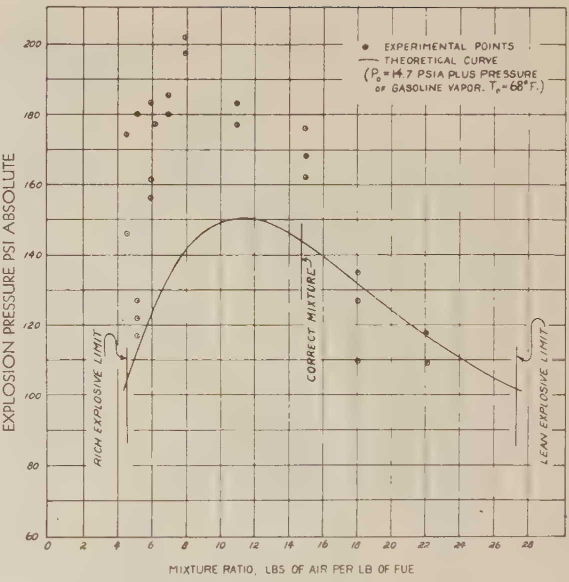

FIG. 5 COMPARISON OF THEORETICAL AND EXPERIMENTAL EXPLOSION PRESSURES

section, the initial pressures for the bomb explosions were slightly higher than 1 atm, and the initial temperature assumed in the calculation of the theoretical curve was 68 F.

The experimental points fall generally above the theoretical curve and also show much scatter among themselves. A number of factors probably contribute to these discrepancies. Perhaps the most important factor is the incomplete vaporization and mixing of the fuel. The theoretical calculations assume that the gasoline is completely vaporized and the vapor uniformly mixed with the air. Under the conditions of the tests, it is very possible that the heavier fractions of the gasoline remained unvaporized. Since the lighter hydrocarbons in the gasoline have higher heats of combustion, the actual explosion pressures would be above the calculated values. This agrees with the experimental measurements. Furthermore, for mixtures richer than the maximum-pressure mixture, the experimentally measured pressures would appear to be too high because the mixtures are actually leaner than would be supposed from the amount of gasoline injected. The large variations in pressures for repeated explosions of mixtures having supposedly the same mixture ratio can be attributed to variations in the degree of vaporization of the fuel and in the homogeneity of the explosive mixture.

A possible cause of error in the strain-gage pressure measurements is the thermal strain produced by the contact of the hot gases with the inner surface of the bomb wall. If the wall were uniformly heated, the temperature of the gages would rise simultaneously, and the error introduced would be small, since the gages are practically temperature-compensated for aluminum. However, only the inside surface of the bomb is heated by the gases. If the temperature at the outer surface remains constant, while the temperature increases toward the inner surface such that the average temperature increase throughout the shell is 1 C, it can be shown that the thermal strain causes a rise in the apparent pressure reading of about 85 psi, while the heat loss from the gas mixture reduces the actual pressure about 30 psi, leaving a net apparent increase in pressure of 55 psi. It is difficult to determine how high a temperature rise in the shell of the bomb should be expected during the short time between sparking

and attainment of maximum pressure, but a 1-deg C average rise might not be unreasonable. It might be expected that the layer of carbon and other materials deposited on the inner surface of the bomb after several charges have been exploded would reduce the amount of heat conducted to the walls and therefore reduce the apparent pressure readings. The experimental data, which designate the first five charges fired, appear to confirm this hypothesis (see Table 1).

An estimate has been made of the gasoline temperature as a function of air pressure which gives the lean-explosive-limit mixture with gasoline vapor saturation. This estimate is shown in Fig. 6, where air pressure has been converted into altitude by the standard atmospheric table. The gasoline vapor pressure as a function of temperature was taken from work of the Cooperative Fuel Research Committee.[6] As long as the rich explosive limit is not exceeded, the region above this curve is the region of true ignitibility, assuming that the lean-limit-mixture ratio is a constant independent of temperature and pressure. The latter assumption implies that the kinetics of the chain reactions involved in ignition do not change appreciably within the temperature and pressure range. Such an assumption can only represent an approximation and most likely is optimistic in that the air-gasoline lean-limit mixture will tend to decrease with temperature. The true ignition-limit curve would thus most likely be above the curve in Fig. 6.

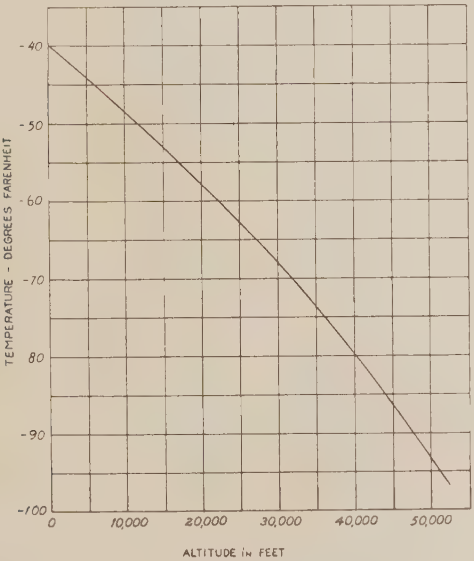

Fig. 6 Estimated Lean-Explosive-Limit Temperatures for Saturated Gasoline-Air Mixtures
(Standard atmospheric conditions assumed.)

Conclusions

A method is developed which permits the calculation of the temperatures and pressures attained when mixtures of air and aviation-gasoline vapor are exploded in a closed chamber. For a few mixture ratios, calculations may be made by the use of thermodynamic charts. Calculations by the two methods show close agreement. The results for various initial temperatures and pressures are presented graphically.

Experimental measurements of the explosion pressures were made using an explosion bomb with a strain-gage pressure pickup. The measured pressures were of the same order of magnitude as the calculated pressures, but in general were somewhat higher than the theoretical values. Possible explanations for this trend are incomplete vaporization of the fuel, incomplete attainment of chemical equilibrium at the explosion temperature, and errors in the pressure measurements caused by temperature gradients.

An estimate of the lower ignition-temperature limit as a function of altitude, for standard atmospheric conditions, is also given.

[6] Reference (4), p. 44.

Appendix 1

Thermodynamic Data

The internal energy of combustion (i.e., heat of combustion at constant volume) of gasoline vapor is equal to the energy of combustion of liquid gasoline plus the energy of vaporization of gasoline. A minimum net calorific value of 18,700 Btu per lb is specified[7] for 100-octane gasoline (liquid). No data have been found for the energy of vaporization of 100-octane gasoline, but Holcomb and Brown (5) give the heat of vaporization of normal octane at room temperature as 140 Btu per lb. Since this figure does not affect the total greatly, the approximation that the energy of vaporization of aviation gasoline equals 140 Btu per lb will be used. The energy of combustion of the gasoline vapor is therefore 18,700 + 140 = 18,840 Btu per lb. If the gasoline has an average molecular weight of 112.21, this value is equivalent to 1,174,400 cal per g-mole.

Values for the thirteen pertinent equilibrium constants at various temperatures are given in Table 2 and plotted in Fig 7. The values are taken from Lewis and von Elbe (3), except the values for log K_5, which are derived from an expression for the free energy of NH_3 given by Thacker, Folkins, and Miller (6).

Table 3 shows the internal energy per gram-mole of the various gases present after combustion, as a function of temperature. For the gases usually present in large amounts, exact energy values at various temperatures, taken from (2), are given. For gases present only in small amounts, approximate equations for the energy as a function of the absolute temperature are given. These expressions have been derived[8] from values of C_p; and from values of the energies of reaction.[9] The quantities are expressed in terms of gram-moles and degrees Kelvin (deg C + 273.2) rather than in the usual engineering units. In all cases, the zero energy level is taken as the internal energy of CO_2, H_2O, O_2, N_2, and A at 300 K, but corrections are given for changing to a base level of 20 C (68 F), —25 C (—13 F), or —55 C (—67 F)

Appendix 2

Example of Algebraic Calculation of Explosion Pressures

As an example of the method of calculating explosion pressures and temperatures as described in the section, "Theoretical Considerations," consider the problem with the particular initial conditions $P_0 = 1$ atm, $T_0 = 68$ F (20 C), and $X = 0.0300$.

Substitution of this value of X in Equations [1] to [4], inclusive, yields

$$a + b + i + q + 2r + s = 0.2403 \quad [19]$$

$$2c + 2d + 4i + j + k + 3t = 0.4800 \quad [20]$$

$$2a + b + c + 2e + g + 2h + j + m + 3n = 0.4070 \quad [21]$$

$$2f + g + h + t = 1.5150 \quad [22]$$

[7] Reference (4), p. 12.
[8] Reference (6), p. 586; and (7), p. 75.
[9] Reference (3), pp. 378–385.

TABLE 2 COMMON LOGARITHMS OF THE EQUILIBRIUM CONSTANTS AT VARIOUS TEMPERATURES
(Pressure units: atmospheres)

Temp °K	log K_1	log K_2	log K_3	log K_4	log K_5	log K_6	log K_7	log K_8	log K_9	log K_{10}	log K_{11}	log K_{12}	log K_{13}
300	-44.72	-39.77	-15.04	-6.42	. .	-43.3	-70.23	-80.2	-28.29	-24.08	. .	-8.935	. .
400	-32.43	-29.26	-11.13	-3.94	. .	-31.7	-51-35	-58.6	-22.15	-19.23	. .	-5.580	. .
600	-20.07	-18.64	-7.194	-1.44	. .	-20.0	-32.41	-36.9	-15.98	-14.41	. .	-2.047	. .
800	-13.89	-13.28	-5.231	-0.18	. .	-14.07	-22.88	-26.1	-12.89	-11.98	. .	-0.180	. .
1000	-10.20	-10.05	-4.052	+0.57	+3.26	-10.53	-17.13	-19.48	-11.03	-10.52	. .	+0.985	. .
1200	-7.755	-7.90	-3.267	+1.08	+3.72	-8.17	-13.28	-15.10	-9.79	-9.530	. .	+1.779	. .
1400	-5.999	-6.34	-2.706	+1.43	+4.08	-6.47	-10.51	-11.97	-8.89	-8.817	. .	+2.354	. .
1600	-4.715	-5.20	-2.285	+1.69	+4.36	-5.20	-8.429	-9.61	-8.22	-8.277	. .	+2.790	. .
1800	-3.690	-4.27	-1.959	+1.89	+4.56	-4.19	-6.803	-7.772	-7.70	-7.850	. .	+3.129	. .
2000	-2.862	-3.52	-1.695	+2.07	+4.72	-3.40	-5.496	-6.298	-7.29	-7.504	-1.695	+3.397	-5.455
2200	-2.193	-2.91	-1.479	+2.18	+4.84	-2.74	-4.424	-5.091	-6.95	-7.221	-1.10*	+3.616	-4.212
2400	-1.648	-2.41	-1.300	+2.29	+4.93	-2.19	-3.529	-4.078	-6.66	-6.980	-0.60*	+3.801	-3.187
2600	-1.206	-2.00	-1.150	+2.39	+5.01	-1.74	-2.769	-3.228	-6.42	-6.777	. .	+3.960	-2.315
2800	-0.811	-1.63	-1.019	+2.48	+5.07	-1.34	-2.115	-2.495	-6.21	-6.595	. .	+4.098	-1.573
3000	-0.470	-1.31	-0.907	+2.56	+5.12	-0.999	-1.548	-1.858	-6.03	-6.440	+0.474	+4.218	-0.926

* Interpolated values.
NOTE: These equilibrium constants are defined in the Theory section. All pressures are in standard atmospheres.
These values are derived from Ref. (2), Table 2, p. 382, except the values for log K_5, which are derived from the values of free energy for NH_3 given in Ref. (6), p. 587.
Some of these values are plotted in Fig. 7, for convenience in interpolation.

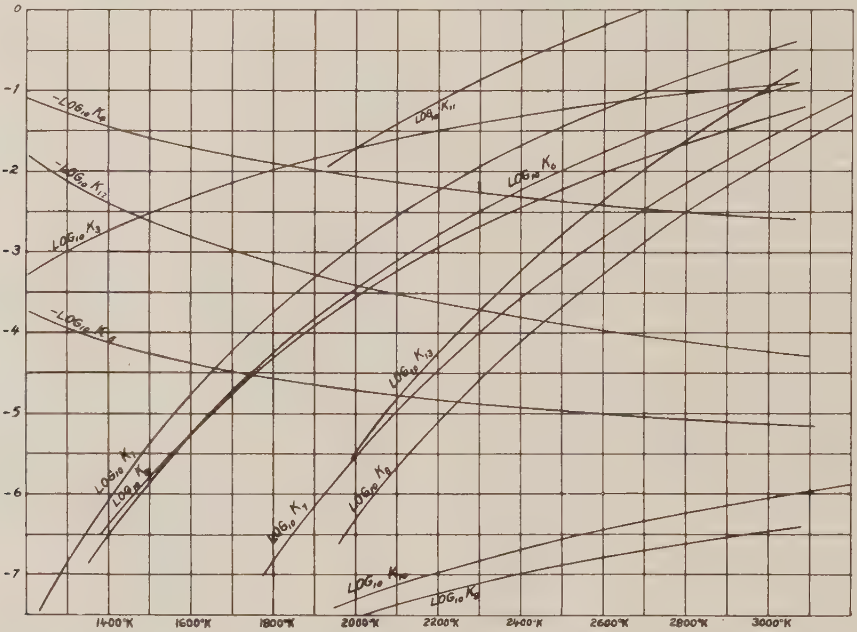

FIG. 7 LOGARITHMS OF EQUILIBRIUM CONSTANTS AT VARIOUS TEMPERATURES
(From Table 2.)

The next step in the calculation is to guess a value for the explosion temperature T_m, which value will be checked at the conclusion of the calculations. Suppose the value $T_m = 2200$ deg K is chosen. The values of the pertinent equilibrium constants $K_1, K_2, \ldots, K_{13}$, are found in Table 2. When these values and the values of T_0 and T_m are substituted in Equations [5] to [17], inclusive, the following equations result

$$\frac{b}{a}\sqrt{e} = 2.34 \times 10^{-3} \quad [23]$$

$$\frac{d}{c}\sqrt{e} = 4.49 \times 10^{-4} \quad [24]$$

$$\frac{g}{\sqrt{fe}} = 0.0332 \quad [25]$$

$$\frac{g}{h}\sqrt{e} = 55.5 \quad [26]$$

$$\frac{\sqrt{d^3 f}}{t} = 9{,}220 \quad [27]$$

$$\frac{j}{c}\sqrt{d} = 6.64 \times 10^{-4} \quad [28]$$

$$\frac{k^2}{d} = 5.02 \times 10^{-6} \quad [29]$$

TABLE 3 INTERNAL ENERGY OF VARIOUS GASES AT T DEG

(Base values at 300 deg equal internal energy of formation from CO_2, H_2O, O_2, N_2, A; units, calories per gram-mole)

	Temperature, Degrees Kelvin														Correction to be Added for Change of Base Temperature to:		
Gas	300	600	800	1000	1200	1400	1600	1800	2000	2200	2400	2600	2800	3000	20°C	-25°C	-55°C
CO_2	0	2,475	4,447	6,587	8,843	11,184	13,586	16,038	18,527	21,043	23,588	26,159	28,746	31,352	46	353	557
H_2O	0	1,896	3,282	4,786	6,409	8,129	9,949	11,864	13,859	15,909	18,009	20,154	22,334	24,539	41	312	492
N_2	0	1,517	2,589	3,727	4,921	6,157	7,423	8,718	10,039	11,368	12,711	14,061	15,425	16,798	34	257	405
O_2	0	1,602	2,779	4,025	5,316	6,637	7,990	9,366	10,762	12,181	13,624	15,084	16,563	18,058	34	261	413
CO	67,329	68,657	69,950	71,110	72,325	73,581	74,865	76,174	77,505	78,851	80,205	81,565	82,936	84,316	29	223	351
H_2	57,503	58,999	60,010	61,041	62,106	63,214	64,356	65,540	66,763	68,017	69,297	70,608	71,944	73,293	24	182	286
OH	34,700	36,225	37,246	38,295	39,377	40,517	41,702	42,917	44,162	45,432	46,742	48,067	49,412	50,784	29	221	349
H	80,326	81,220	81,815	82,411	83,007	83,603	84,199	84,795	85,390	85,986	86,582	87,178	87,773	88,368	12	91	143
O	58,801	59,695	60,290	60,886	61,482	62,078	62,674	63,270	63,865	64,461	65,057	65,653	66,248	66,843	17	130	206
NO	21,528	23,108	24,244	25,446	26,698	27,986	29,301	30,636	31,987	33,351	34,726	36,109	37,500	38,897	34	259	409
A	$-894 + 2.979\ T$														20	154	244
NO_2	$6190 + 5.0\ T + 3.5 \times 10^{-3}\ T^2 - 6.2 \times 10^{-7}\ T^3$														41	389	615
NH_3	$73,910 + 6.05\ T + 3.5 \times 10^{-4}\ T^2 + 1.7 \times 10^{-6}\ T^3$														53	401	631
O_3	$32,440 + 5.0\ T + 3.5 \times 10^{-3}\ T^2 - 6.2 \times 10^{-7}\ T^3$														41	391	619
CH_4	$189,500 + 4.74\ T + 5.1 \times 10^{-3}\ T^2 + 1.12 \times 10^5\ T^{-1}$														60	456	716
C (graphite)	$93,071 + 2.673\ T + 1.308 \times 10^{-3}\ T^2 + 1.17 \times 10^5\ T^{-1}$														12	92	144
C (gas)	$217,000 + 3.0\ T$														12	92	144
C_2(gas)	$309,000 + 4.5\ T + 5 \times 10^{-4}\ T^2$														12	92	144

NOTE: Precise energy values (in upper part of Table) taken from Ref. (2), Table 5A, p. 423. Approximate expressions in lower part of table (for gases present in small amounts) derived from values of C_p given in Ref. (6), p. 586, and Ref. (7), p. 75, and values of thermal energies and energies of reaction given in Ref. (3), pp. 378–385. T = temperature in degrees Kelvin (°C + 273.2).

$$\frac{m^2}{e} = 1.08 \times 10^{-6} \quad [30]$$

$$\frac{n}{e^{3/2}} = 3.07 \times 10^{-7} \quad [31]$$

$$\frac{1}{b}\sqrt{e} = 1.65 \times 10^{-7}, (q > 0) \quad [32]$$

$$\sqrt{r} = 2.9 \times 10^{-2}, (q > 0) \quad [33]$$

$$\frac{d^2}{i} = 550, (q > 0) \quad [34]$$

$$s = 8.18 \times 10^{-6}, (q > 0) \quad [35]$$

It should be noted that the last four equalities hold only when q is greater than zero, that is, when solid carbon is present. Although, with this particular gasoline-air mixture, not enough oxygen is present for complete combustion of the fuel to CO_2 and H_2O, more than sufficient oxygen is present to burn the fuel to CO and H_2O. Therefore, it might be supposed that the carbon atoms in the mixture are so completely oxidized that the free carbon remaining is so small in amount as to be completely vaporized, or, in other words, $q = 0$. In working the problem, this assumption will be made and then checked before the calculation is completed.

If $q = 0$, Equations [32] to [35], inclusive, must be replaced by the corresponding inequalities

$$\sqrt{e} > 1.65 \times 10^{-7}\, b, \quad (q = 0) \quad [36]$$

$$\sqrt{r} < 2.9 \times 10^{-2}, \quad (q = 0) \quad [37]$$

$$i < d^2/550, \quad (q = 0) \quad [38]$$

$$s < 8.18 \times 10^{-6}, \quad (q = 0) \quad [39]$$

These inequalities permit only upper limits to be set on the values of r, i, and s. However, in many instances, these upper limits are sufficiently small to show that r, i, and s are negligible. For mixture ratios which give moderately large upper limits, the exact values of r, i, and s can be calculated by transforming Equations [32], [33], [34], and [35] for equilibriums involving solid carbon into similar equations involving gaseous carbon, by the fundamental chemical principle that, in equilibriums involving a solid, the solid may be replaced by the corresponding vapor, at a pressure equal to the vapor pressure of the solid, without disturbing the equilibrium.

If it is assumed that $q = 0$ and that r, i, and s can be neglected, the problem is reduced to the solution of thirteen nonlinear simultaneous equations, Equations [19] through [31], in thirteen unknowns. These simultaneous equations cannot be solved explicitly, so the method of successive approximation must be used. Perhaps the simplest way to begin is to guess a value of a and follow through the evaluation of the other unknowns until all the equations check. Suppose the value $a = 0.0416$ is tried. Since i, q, r, and s are assumed negligible, Equation [19], becomes

$$a + b = 0.2403$$
$$b = 0.2403 - a = 0.1987 \quad [40]$$

From Equation [23]

$$e = (2.34 \times 10^{-3}\, a/b)^2$$
$$= 2.40 \times 10^{-7} \quad [41]$$

From Equation [24]

$$c = \frac{d\sqrt{e}}{4.49 \times 10^{-4}}$$
$$= 1.092\, d \quad [42]$$

From Equation [28]

$$j = 6.64 \times 10^{-4} \frac{c}{\sqrt{d}}$$
$$= 7.25 \times 10^{-4}\sqrt{d} \quad [43]$$

From Equation [29]

$$k = 2.24 \times 10^{-3}\sqrt{d} \quad [44]$$

From Equation [27]

$$t = 1.085 \times 10^{-4}\sqrt{d^3 f} \quad [45]$$

If the assumption is made that g, h, and t are fairly small, it follows from Equation [22] that $f = 0.757$. Therefore, from Equation [45]

$$t = 9.44 \times 10^{-5}\sqrt{d^3} \quad [46]$$

If the values of c, j, k and t from Equations [42], [43], [44], and [46] are substituted in Equation [20] that equation becomes

$$4.184d + 2.97 \times 10^{-3}\sqrt{d} + 2.83 \times 10^{-4}\sqrt{d^3} = 0.4800$$

whence

$$d = 0.1144 \quad [47]$$

It follows, from Equations [42], [43], and [44], that

$$c = 0.1249$$
$$j = 2.45 \times 10^{-4}$$
$$k = 7.58 \times 10^{-4}$$

Equation [25] can be rewritten

$$g = 0.0322\sqrt{fe}$$
$$= 1.63 \times 10^{-5}\sqrt{f} \quad [48]$$

If Equation [25] is divided by Equation [26], the result is

$$\frac{h}{e\sqrt{f}} = 5.98 \times 10^{-4}$$
$$h = 5.98 \times 10^{-4}\, e\sqrt{f}$$
$$= 1.44 \times 10^{-10}\sqrt{f} \quad [49]$$

The value of d from Equation [47] can be substituted in Equation [45] to give

$$t = 4.20 \times 10^{-6}\sqrt{f} \quad [50]$$

An equation for f results when the values for g, h, and t from Equations [48], [49], and [50] are substituted in Equation [22]

$$2f + 2.05 \times 10^{-5}\sqrt{f} = 1.5150$$
$$f = 0.7575$$

whence

$$g = 1.42 \times 10^{-5}$$
$$h = 1.25 \times 10^{-12}$$
$$t = 3.66 \times 10^{-6}$$

From Equations [30] and [31], using the value of e from Equation [41]

$$m = \sqrt{1.08 \times 10^{-6}\, e}$$
$$= 5.11 \times 10^{-10}$$

$$
\begin{aligned}
n &= 3.07 \times 10^{-7}\, e^{3/2} \\
&= 3.66 \times 10^{-17}
\end{aligned}
$$

All of the unknowns have now been evaluated using an estimated value of a, and twelve of the thirteen equations. To check the estimated value of a, use is made of the remaining equation, Equation [21]. Substitution of the values of the unknowns in the left side of Equation [21] yields the near equality

$$0.4071 = 0.4070$$

It remains to confirm the initial assumption that q, r, i, and s are negligible. The values of b and e calculated previously are such that

$$\sqrt{e} = 2.48 \times 10^{-3}\, b$$

Comparison of this with inequality [36]

$$\sqrt{e} > 1.65 \times 10^{-7}\, b, \ (q = 0)$$

shows that q does equal zero. The ratio of the pressure of the carbon vapor actually present to the vapor pressure of carbon at the explosion temperature is $1.65 \times 10^{-7}/2.48 \times 10^{-3} = 6.65 \times 10^{-5}$. Inequalities [37], [38], and [39] can then be converted into the following equations

$$\sqrt{r} = (6.65 \times 10^{-5})\,(2.9 \times 10^{-2}) = 1.9 \times 10^{-6}$$

$$i = \frac{6.65 \times 10^{-5}\, d^2}{550} = 1.21 \times 10^{-7}\, d^2$$

$$s = (6.65 \times 10^{-5})\,(8.18 \times 10^{-6}) = 5.44 \times 10^{-10}$$

Evaluation of r and i yields

$$
\begin{aligned}
r &= 3.6 \times 10^{-12} \\
i &= 1.58 \times 10^{-9}
\end{aligned}
$$

Thus the assumption that q, r, i, and s can be neglected is verified.

The calculations thus far have shown the equilibrium composition of the gasoline-air mixture at 2200 K. It must still be demonstrated that the heat evolved by the reaction is just sufficient to raise the temperature of the gas mixture to the assumed value of 2200 K.

The heat energy evolved at room temperature by the complete constant-volume combustion of 0.030 mole of gasoline vapor to H_2O (vapor) and CO_2 is (1,174,400) (0.030) = 35,232 cal (see Appendix 1).

The energy required to raise the temperature from 293 K to 2200 K and cause the dissociation which occurs at the latter temperature is calculated by using the coefficients $a, b, \ldots, t$ already calculated, and the internal energy per mole of the various gases at 2200 K above the energy of O_2, N_2, H_2O, and CO_2 at 293 K, values of which are given in Table 2.

The calculation may be conveniently presented as in Table 4.

The last two columns of Table 4 show the rate of change of energy with temperature and will be used to estimate how much the actual explosion temperature differs from 2200 K. The

TABLE 4

Gas	No. moles	ΔE per mole[a]	ΔE per react.	$\frac{d(\Delta E \text{ per mole})}{dT}$	$\frac{d(\Delta E \text{ per react.})}{dT}$
a CO_2	0.0416	21089	878	12.2	0.51
b CO	0.1987	78880	15670	6.8	1.35
c H_2O	0.1249	15950	1991	10.5	1.31
d H_2	0.1144	68041	7783	6.4	0.73
f N_2	0.7575	11402	8630	6.71	5.14
j OH	0.00025	45461	11	....	....
k H	0.00076	85998	65	....	....
A	0.0090	5680	51	....	....
Total	1.2471		35079		9 04

[a] Refer to Table 3.

difference between the energy evolved and the energy at 2200 K is 35,232 — 35,079 = 153 cal. The actual temperature is, therefore, approximately 153/9.04 = 16.9 deg above 2200 K. This is not different enough from 2200 K to change the equilibrium constants significantly, so the value T_m = 2217 K (1944 C) may be taken as accurate. The explosion pressure P_m can be calculated from Equation [18]

$$
\begin{aligned}
P_m &= P_0 N_m T_m / T_0 \\
&= (1.000)\,(1.2471)\,(2217)/293.2 \\
&= 9.42 \text{ atm} = 138.5 \text{ psia}
\end{aligned}
$$

It should be noted that if the energy-balance calculation shows that the temperature is considerably different from the estimated temperature which was used to calculate the unknown coefficients, the entire calculations must be repeated using the equilibrium constants at the newly estimated temperature. However, if the difference between the two temperatures is not too great, the recalculation may be greatly simplified by neglecting the coefficients which the previous calculation has shown are negligible. In fact, for all mixture ratios, more than one half of the seventeen coefficients can be neglected; however, the coefficients which may be neglected are different for different mixture ratios. In the calculations presented, the precise value of every coefficient has been given so that there will be no question as to which coefficients are negligible.

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Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Heat Transfer in Annuli

By C. Y. CHEN,[1] G. A. HAWKINS,[2] and H. L. SOLBERG[3]

A survey of the literature reveals that few data are available relative to heat transfer in annular spaces for the case of laminar flow. This paper presents data acquired in an experimental study[4] of the heat-transfer coefficients at the inner wall of four different annuli for the case of laminar flow of water. Both heating and cooling tests were conducted. The experiments were performed over a range of Reynolds numbers from 200 to 2000. The accompanying dimensionless equation was

$$\frac{hD}{k} = 1.02\left(\frac{VD\rho}{\mu_a}\right)^{0.45}\left(\frac{c_p\mu_a}{k}\right)^{1/3}\left(\frac{\mu_a}{\mu_w}\right)^{0.14}\left(\frac{D}{L}\right)^{0.4}\left(\frac{D_1}{D_2}\right)^{0.8}\cdot\left(\frac{D^3\rho^2\beta g\Delta t}{\mu_a^2}\right)^{0.05}$$

found to correlate both the heating and the cooling data for all four annuli with an average deviation of ±6.6 per cent and a maximum deviation of ±14.1 per cent. This equation may be used for computing the heat-transfer coefficient on the inner wall of an annulus for laminar flow of water for Reynolds numbers in the range of 200 to 2000.

Nomenclature

The following nomenclature is used in the paper:

a = constant
A = area, sq ft
b = constant
c_p = heat capacity at constant pressure, Btu/(lb$_{mass}$)(deg F)
d = constant
D_1 = outer diameter of annulus, ft
D_2 = inner diameter of annulus, ft
D = equivalent diameter of annulus, ft
D_e = characteristic dimension of annulus, ft
e = constant
f = constant
g = gravitational acceleration, ft/hr^2
h = surface coefficient of heat transfer, Btu/(hr)(sq ft)(deg F)
k = thermal conductivity, Btu/(hr)(ft)(deg F)
L = characteristic linear dimension, ft
L = heating or cooling length of an annular section, ft
m = constant
m = hydraulic radius, ft
q_r = heat transferred by radiation, Btu/hr
t = temperature, deg F
T = absolute temperature, deg R
V = velocity, fph
α = constant
ϵ = emissivity of surface for radiation
β = coefficient of cubical expansion, (deg F)$^{-1}$
ρ = density, (lb$_{mass}$)/cu ft
μ = dynamic (absolute) viscosity, (lb$_{mass}$)/(hr)(ft)
μ_w = dynamic (absolute) viscosity at the surface temperature, (lb$_{mass}$)/(hr)(ft)
μ_a = dynamic (absolute) viscosity evaluated at bulk-fluid temperature, (lb$_{mass}$)/(hr)(ft)
Δ = difference

Introduction

The double-pipe heat exchanger is frequently used in engineering work to transmit heat from one fluid to another. The design of this type of heat exchanger requires a knowledge of the heat-transfer coefficient in the annular space. Designing engineers have long felt the need for a single heat-transfer correlation which could be used for determining the coefficient for the case of an annular space for either a heating or a cooling process. In an annular space, heat transfer may take place either at the inner surface of the annulus or at the outer surface or both.

A survey of the literature reveals that in the last decade a number of successful investigations have been made on the heat transfer in annular spaces for the case of turbulent flow. However, very little has been done for the case of laminar flow in annular spaces. Jakob and Rees (1)[5] presented a mathematical theory of heat transfer between the walls of an annulus and a fluid passing through it in laminar flow for the cases of uniform heating or cooling from outside, inside, or from both sides at the same time.

In a recent paper, Davis (2) suggested a tentative dimensionless equation for determining heat-transfer coefficients in annuli for the case of laminar flow. Since there were no experimental data available, the constants of the equation were left as unknowns.

The criterion of flow is the Reynolds number $\frac{VD\rho}{\mu}$. In case of flow in an annular space, usually an equivalent diameter is used for D. There are two methods of calculating the equivalent diameter of an annulus. In the first of these methods, a value of 4 times the hydraulic radius m is used for the equivalent diameter. The hydraulic radius may be expressed as follows

$$m = \frac{\frac{\pi}{4}(D_1^2 - D_2^2)}{\pi(D_1 + D_2)} = \frac{D_1 - D_2}{4}$$

The equivalent diameter in this case is therefore $(D_1 - D_2)$.

In the second method, the heated perimeter is used. Where heat transfer takes place at the inner wall the equivalent diameter is

$$D = \frac{4 \times \frac{\pi}{4}(D_1^2 - D_2^2)}{\pi D_2} = \frac{D_1^2 - D_2^2}{D_2}$$

This was the first suggested by Jordan (3) and later by Nusselt (4).

[1] Research Foundation Fellow, Purdue University, Lafayette, Ind. Jun. A.S.M.E.
[2] Westinghouse Research Professor of Heat Transfer, Purdue University. Mem. A.S.M.E.
[3] Head, School of Mechanical and Aeronautical Engineering, Purdue University. Mem. A.S.M.E.
[4] Based on a doctoral thesis, Purdue University, 1943, by one of the authors.[1]
Contributed by the Heat Transfer Division of The American Society of Mechanical Engineers.

[5] Numbers in parentheses refer to the Bibliography at the end of the paper.

Although both of these equivalent diameters have been used for correlating data dealing with heat transfer in annular spaces, the one based on 4 times the hydraulic radius was used in this investigation to compute the Reynolds number.

For flow in a round pipe, it has been established that the lower critical Reynolds number occurs at about 2300. The lower critical Reynolds number for flow in an annular space has not been completely ascertained. Using an annular cylinder having a ratio of radii of 0.818 to 1, Page (5) found a value of 2400 as the lower critical Reynolds number. Researchers (6, 7) on fluid friction in annular spaces have shown that there is no abrupt change in friction factor for Reynolds number below 2000. It is therefore believed that the lower critical Reynolds number for flow in annular spaces must be about 2000.

In the present experiments, the range of Reynolds number investigated was from 200 to 2000. A few cases of abrupt increase in the heat-transfer coefficient occurred near a Reynolds number of 2000. This was taken as an indication that the critical Reynolds number had been reached and these data were discarded. None of the tests were conducted at a Reynolds number less than 200, owing to the difficulty of regulating the flow with the available equipment.

Theoretical Considerations

The problem of forced convection has long defied mathematical solution. Only for the simplest cases and corresponding simplified assumptions has the mathematical solution been successful. In the present cases, because of geometrical complications, variable temperature gradients, and heat exchange, dimensional analysis was used to correlate the test data. By using general dimensionless groups which have been used in other correlations, the following equation was selected as a first approximation

$$\frac{hD_c}{k} = \alpha\left(\frac{VD_c\rho}{\mu_a}\right)^a\left(\frac{c_p\mu_a}{k}\right)^b\left(\frac{L}{D_c}\right)^d\left(\frac{D_1}{D_c}\right)^e\left(\frac{\mu_w}{\mu_a}\right)^f \ldots\ldots [1]$$

where D_c is some characteristic dimension of the annulus. Since the criterion of flow is the Reynolds number based on the equivalent diameter D, it is logical to use this equivalent diameter for D_c throughout the correlation.

The factor $(D_1/D_c)^e$ expresses the effect of geometric proportion of different annuli. Inasmuch as the equivalent diameter D is a function of D_1 and D_2, it will simplify computations to use the factor $(D_1/D_2)^e$ directly rather than to use an equivalent or some other form.

For the case of laminar flow of fluids of low viscosities such as water, natural convection sometimes plays a prominent part. When dimensional analysis is applied to problems dealing with natural convection, the Grashof number, $\frac{L^3\rho^2\Delta t\beta g}{\mu^2}$ appears. Introducing the Grashof number, Colburn (8) has correlated the data for water, air, and petroleum oil for the case of laminar flow in round pipes by use of the following multiplying factor:

$$(1 + 0.015\mathbf{G}^{1/3})$$

It has been pointed out by Martinelli and his associates (9) that for fluids of low viscosities the Grashof number appears in the general equation as a constant multiplier. Thus the effect of free convection which is represented by the Grashof number does not decrease as the Reynolds number becomes larger, but remains a constant multiplier.

In this work attempts were made to express the free convection effect as a function of the Reynolds number, the product of the Grashof and Prandtl numbers, and other dimensionless groupings. However, for the data obtained in these experiments the most satisfactory correlation was found using the factor $K(\mathbf{G}/\mathbf{R})^r$, where K is a constant. By use of this relation in Equation [1], it is possible to combine the equation constant and K and the various Reynolds numbers. Hence in the final equation the Grashof number appears as a multiplying factor.

It is recognized that this procedure is open to question. Other correlations may be possible, which would make the equation suitable for use beyond the range of the experimental data reported.

Since D, the equivalent diameter, is to be used in Equation [1], in the Reynolds number, it is desirable to use this same D as the characteristic dimension L in the Grashof number. There seems to be no universally accepted method of evaluating viscosity in the Grashof number. Colburn (8, 10) used the viscosity at the film temperature but there is some uncertainty as to how this film temperature should be computed. The viscosity at the bulk fluid temperature μ_a therefore was used in the computation of the Grashof number in this correlation.

Hence Equation [1] becomes

$$\frac{hD}{k} = \alpha'\left(\frac{VD\rho}{\mu_a}\right)^{a'}\left(\frac{c_p\,\mu_a}{k}\right)^b\left(\frac{L}{D}\right)^d\left(\frac{D_1}{D_2}\right)^e\left(\frac{\mu_w}{\mu_a}\right)^f\left(\frac{D^3\rho^2\,\Delta t\beta g}{\mu_a^{\,2}}\right)^m . [2]$$

For both the laminar and the turbulent flow in round pipes, Sieder and Tate (11) and Colburn (8, 10) used an exponent for the Prandtl number equal to $^1/_3$. For heat transfer at the inner wall of the annuli with turbulent flow, Monrad and Pelton (12), and Davis (2) correlated the data with a dimensionless equation, in which the exponent of $^1/_3$ was used for the Prandtl number. Based on these results, a value of $b = ^1/_3$ was used in correlating the experimental data.

The use of $(\mu_w/\mu_a)^f$ in the heat-transfer correlation was first introduced by Sieder and Tate (11). For flow inside round pipes they used —0.14 for f. This factor accounts for the effect of the increased or decreased velocity gradients at a heated or cooled wall and may hold even if another wall is placed opposite to it as in an annulus. The 0.14 power is valid in crossflow according to Sieder. It seems probable that the presence of another wall would not affect appreciably the velocity gradient at the heated or cooled wall. Davis (2) used this power to correlate the data for heat transfer at the inner wall of the annuli with turbulent flow, and $f = -0.14$ was also used to correlate the data in this investigation. All of these assumptions can be justified only in the light of additional experimental data.

Description of Apparatus

Fig. 1 is a schematic diagram of the complete setup of the equipment. Letter A designates a large water tank equipped with a suitable overflow. The heating device consisted of copper coils connected to a steam pipe. A mechanical stirrer was used to maintain a uniform temperature.

From tank A water was pumped to the test section. A valve and a by-pass were inserted to control the flow. The inlet temperature to the annular section was measured by a thermometer inserted through a cork plug at one end of a tee. The temperature was measured at a point where the leading pipe made a turn which was located at a considerable distance from the entrance to the test section. The location of the thermometer was selected in order to eliminate unnecessary turbulence near the entrance. The leading pipe was suitably insulated.

Water flowed through the annular section into the mixing chamber where the outlet temperature was measured by means of thermometers. From the mixing chamber, the water flowed either to the weighing tank or to the drain. Water flowing in the inner pipe followed a somewhat similar path, as shown in Fig. 1.

The test section consisted of two brass pipes, one inside the other. Fig. 2 shows a part of the test section. The inner pipe

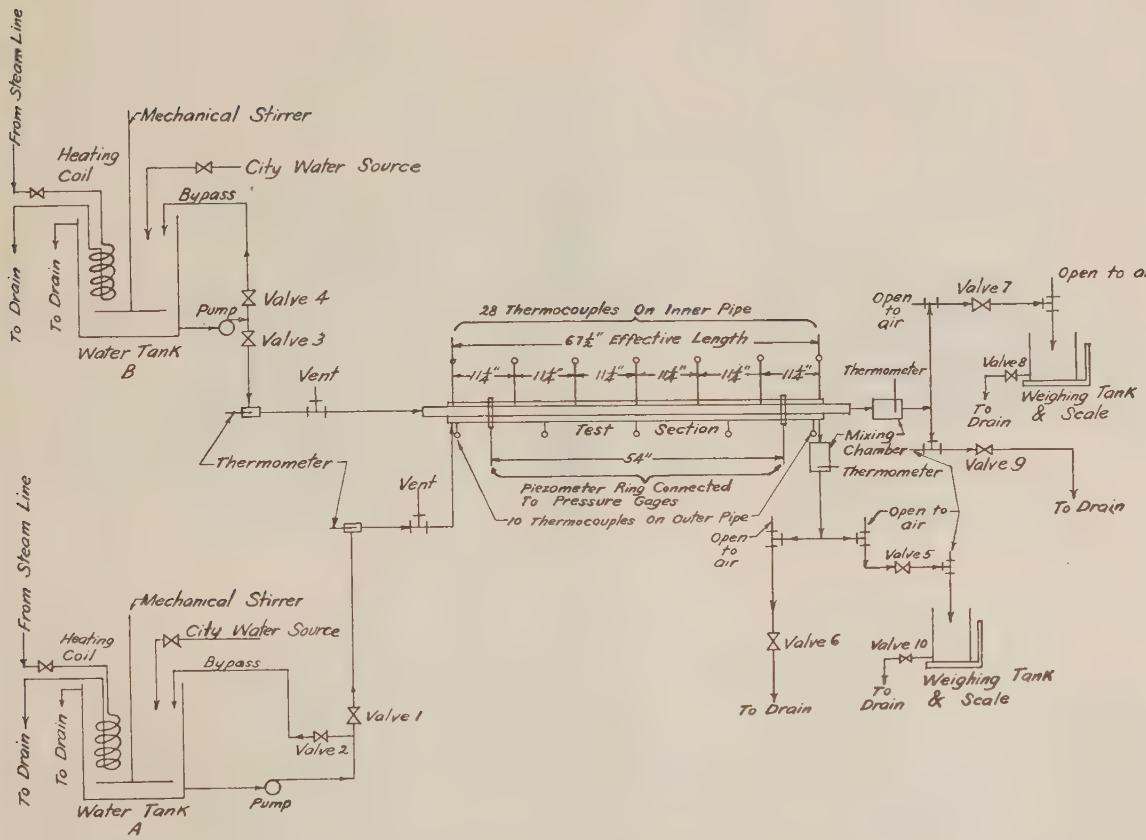

Fig. 1 Schematic Diagram of Complete Setup

was 7 ft in length. It was connected to the inlet pipe by a bronze sleeve, threaded on one end and soldered to the inner pipe at the other end. The outer pipe was $66^1/_2$ in. long. Both of its ends were slipped into the ends of two cast-iron pieces and the joints soldered. These two cast-iron pieces served as the entrance and exit pieces and were designed to have the flow area in the section leading to the inlet and outlet of the annular section approximately equal to the cross-sectional area of the annular space, thus eliminating rapid changes in the flow areas. Wooden plugs screwed into these cast-iron pieces were designed to permit smooth entrance to and from the annulus, and to serve as heat insulation. Steel packing glands were used to prevent leakage.

A total of $67^1/_2$ in. effective internal heating or cooling length was used in all tests. No calming section was used owing to construction difficulties. Water was led into and out of the test section in a horizontal direction.

The following brass pipes were used:

Pipe	Outside diameter, in.	Inside diameter, in.
A	$2^1/_4$	2
B	$1^1/_2$	$1^3/_8$
C	1	$^7/_8$
D	$^3/_4$	$^1/_2$

Four combinations were tested. These were *A-C*, *A-D*, *B-C*, and *B-D*. The inner pipe was centered inside the outer pipe by the wooden plugs and steel packing glands, and was strong enough to prevent sagging.

The temperatures of the outside surface of the inner pipe were measured along the pipe by calibrated 28-gage copper-constantan thermocouples. Seven sets of couples located $11^1/_4$ in. apart were spaced over the effective length. Each set of couples consisted of 4 pairs of couples placed around the section at 90-deg intervals so as to give an average temperature of the section. The first and last sets were placed near the ends of the wooden plugs.

The thermocouples were attached to the pipe wall by a modified method suggested by Colburn and Hougen (13). Longitudinal slots 3 in. long, $^1/_{16}$ in. wide, and $^1/_{16}$ in. deep were cut on the outside surface of the inner pipe. The cotton-covered insulation on the wires was first impregnated with a water-resistant varnish. After welding the junction, the insulated wires were twisted together and pressed into the slot. A drop of solder was placed over the welded ends of wires to form a rigid connection between the thermocouple and the pipe. The soldered joint was then polished with fine emery cloth in order to get a surface flush with the pipe surface. Waterproof varnish was applied to the wires in the slots in order to make the surface of the slot flush with the pipe surface.

The thermocouples were led out vertically from the inner pipe surface through $^1/_8$-in. holes drilled in the outer pipe and through a small packing gland to the outside. These small packing glands were screwed into seats which were soldered to the outside surface of the outer pipe. All the thermocouple wires were led downstream except the last group which was brought upstream because of space limitations. In this case a fine copper wire was

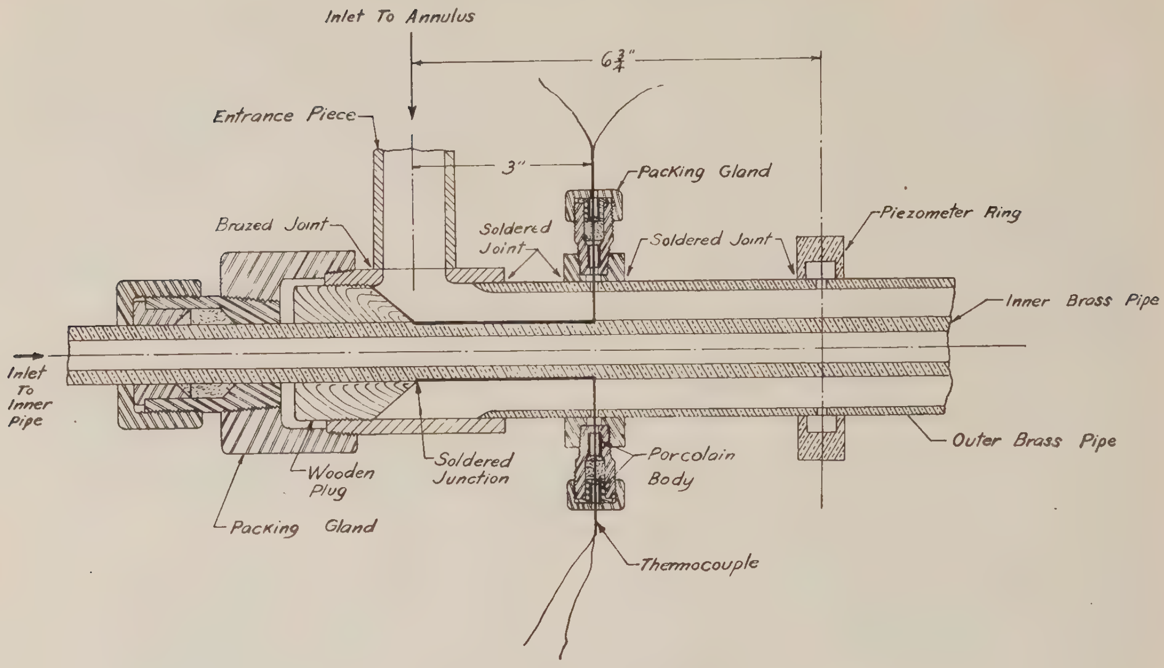

FIG. 2 ENTRANCE SIDE OF TEST SECTION

wound around the ends to prevent them from becoming loose. This prevented the couples from being moved out of the slot by the flowing fluid. After the tests the couples were all inspected and were found to be practically in the same condition as they were when first placed in the slots.

The outer pipe was cut into three sections to simplify the operation of installing the thermocouple wires. After all the couples were installed and checked, the outer pipe sections were joined by two sleeves and soldered. The surfaces at the sections were faced smoothly so that when the sections were joined together in the sleeve, they would closely match each other and thus prevent turbulence.

The short lengths of thermocouple wire leading from the inner pipe surface through the holes in the outer pipe surface and through the glands, were impregnated with waterproof varnish. Since 28-gage thermocouple wires were used, it is believed that the wires caused very little disturbance to the flow. The accuracy obtained from averaging the temperatures around the section should more than offset the small additional turbulence caused by the thermocouples.

Five sets of thermocouples were soldered on the outside surface of the outer pipe. Each set consisted of two pairs of couples, one at the top and the other at the bottom. These gave the outer-wall temperature gradient along the pipe.

Piezometer holes $^1/_8$ in. diam were drilled at two sections on the outer pipe 54 in. apart, spaced equally from the two ends. At each section the four openings were connected to a brass piezometer ring. The pressure readings from a Wahlen gage and an inclined draft gage served as a check as to whether laminar flow existed in the annular space.

The mixing chambers, Fig. 3, were made of 4-in. standard galvanized pipe and were fitted with half-moon baffles on the inside to promote mixing. In order to obtain the outlet temperature, two thermometers were inserted through the rubber stoppers at the top of the mixing chamber with their bulbs widely spaced at two different heights in the chamber. The exact check of the two thermometer readings indicated the efficiency of the mixing chamber. A short glass tube inserted through one of the rubber stoppers and connected to a glass stopcock served as a vent for the air to escape from the chamber when the fluid started to flow. The results of temperature measurements made on the mixing chamber during preliminary calibration tests did not show any evidence of stratification.

The outside surface of the outer pipe and the mixing chambers were heavily insulated.

The test section was mounted horizontally with a slight inclination to enable air to escape from the vent at the exit end.

From the mixing chamber, water was led to the weighing tank and the drain through separate pipe lines and valves. As shown in Fig. 1, two tees were placed in the flow paths before the valves. The tees were open to the atmosphere at one end to stabilize the flow.

After the tests had been completed on each combination of pipes, the inner pipe was taken out and examined. The outer wall of the inner pipe was found to be slightly fouled. When the inner pipe was used again, in a different combination, its outer surface was carefully polished.

TEST PROCEDURE

All thermometers were calibrated. The thermocouples were checked against each other by running water at the same temperature through both the inner pipe and the annular section. Only parallel flow was employed and both heating and cooling runs were made. Water was used in both the annulus and the inner pipe.

To start a test, the sources of water were turned on and the mechanical stirrer set into operation. Cooling tests were conducted first. Valves nos. 1 and 2, Fig. 1, were adjusted to give the flow corresponding to the Reynolds number. Valves nos. 3 and 4 on the inner pipe side were adjusted to give a flow such that

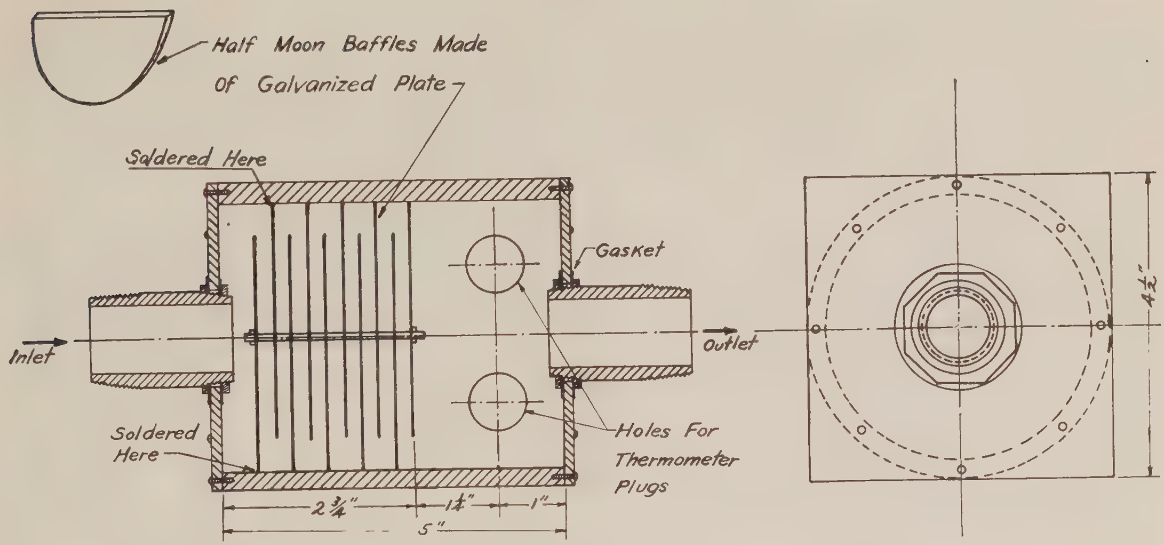

FIG. 3 MIXING CHAMBER

the drop in temperature of the fluid in the annular section would be as small as practical but large enough to give a suitable temperature between water flowing out from the annulus and water flowing out from the inner pipe. Valves nos. 5 and 7 were kept open while valves nos. 6 and 9 were closed. Water flowed into the two weighing tanks during the test except for a brief instant when the initial tank readings were taken.

The various temperature and pressure readings were taken at regular intervals. After these readings had remained constant for about 1 hr, thermal equilibrium was reached and a set of readings was taken, which included the inlet and outlet temperatures of water in both the annulus and the inner pipe, the pipe-wall temperatures, and the weight rate of flow in both the annulus and the inner pipe.

Check tests were conducted on different days to ascertain whether or not the surfaces were becoming contaminated by dirt or other substances. In every case these test results agreed.

Experimental Results and Conclusions

The original experimental data together with computations are presented in Tables 1, 2, 3, and 4. In computing the data the average fluid temperature in the annulus was taken as the arithmetic average of the inlet and outlet temperatures. The heat transferred was based on readings on the inner pipe. The heat-transfer coefficient was based on the arithmetic-mean temperature difference. The inner pipe-wall temperature was the average of the 28 thermocouple readings. Values of μ, c_p, and k were based on those given by Dorsey (14). The coefficients of cubical expansion of water at different temperatures were based on Kopp's data on the relative volumes of water at different

TABLE 1

D_1 = 2 in. D_2 = 1 in. $\frac{D_1}{D_2} = 2$ D = 1 in.

Heating or cooling length, L = 67 1/2 in. $\frac{L}{D} = 67.5$

Run No.	Water Temp. °F, Annulus Out	Water Temp. °F, Annulus In	Water Temp. °F, Inner Pipe Out	Water Temp. °F, Inner Pipe In	Aver. Water Temp. In Annulus, F	Aver. Inner Pipe Wall Temp. F	Temp. Diff. t F	Water Rate in Annulus lb hr^{-1}	Water Rate in Inner Pipe lb hr^{-1}	Heat Transferred (Based on readings on Inner Pipe B hr^{-1}	Heat Transfer Coefficient h Bhr^{-1}ft^{-2}F	Nu $(\frac{hD}{k})$	Pr $(\frac{c_p \mu_a}{k})$	Gr $(\frac{D^3 \rho^2 g \beta \Delta t}{\mu_a^2})$	$\frac{\mu_a}{\mu_w}$	Re $(\frac{uD\rho}{\mu_a})$
Cooling Tests: Water in the Annulus Being Cooled																
1	106.3	126.1	98.3	62.7	116.2	103.9	12.3	67.5	28.5	1013	56.0	12.60	3.79	1,425,000	0.885	245
2	112.2	126.0	108.9	61.7	119.1	106.3	12.8	95.7	26.7	1260	66.6	14.97	3.68	1,624,000	0.870	358
3	108.9	120.7	93.0	61.9	114.8	98.9	15.9	170.3	61.3	1906	81.5	18.4	3.85	1,770,000	0.850	610
4	106.8	121.0	83.9	60.5	113.9	94.1	19.8	202	113.6	2660	91.2	20.6	3.88	2,150,000	0.811	719
5	114.3	121.8	102.1	64.0	118.1	105.9	12.2	238	42.5	1620	90.1	20.3	3.72	1,595,000	0.887	881
6	121.8	121.4	92.4	60.7	116.8	101.2	15.6	316	83.1	2640	115.0	25.9	3.76	1,848,000	0.832	1156
7	113.0	122.7	85.6	62.0	117.9	98.9	19.0	367	143.1	3380	120.9	27.2	3.72	2,310,000	0.825	1356
8	112.8	121.3	80.6	61.0	117.1	97.5	19.6	480	202	3960	137.2	30.9	3.75	2,330,000	0.817	1760
9	114.3	120.2	90.9	60.9	117.3	100.7	16.6	539	102.3	3070	125.7	28.3	3.74	1,985,000	0.847	1980
Heating Tests: Water in the Annulus Being Heated																
10	71.0	64.8	73.1	113.6	67.9	75.2	7.5	113.0	16.9	683	64.0	15.43	7.05	128,500	1.100	236
11	71.3	64.7	76.6	114.1	68.0	77.4	9.4	166.6	27.6	1037	74.7	18.00	7.04	167,000	1.133	348
12	70.3	63.2	79.1	118.1	66.8	78.5	11.7	205	36.0	1403	81.5	19.65	7.19	194,200	1.168	421
13	69.2	64.0	81.2	117.1	66.6	78.6	12.0	312	47.2	1693	95.9	23.2	7.20	196,100	1.173	638
14	69.6	63.1	96.1	119.0	66.4	82.9	16.5	428	124.5	2850	117.0	28.1	7.22	270,000	1.239	876
15	70.8	63.9	101.4	115.8	67.4	86.1	18.7	483	232	3340	121.5	29.3	7.12	320,000	1.271	1000
16	68.8	64.2	99.3	117.1	66.5	83.1	16.6	666	165.3	2940	120.5	29.1	7.21	272,000	1.198	1360
17	67.6	62.9	100.3	114.9	65.3	85.2	19.9	885	288	4200	143.1	34.7	7.35	304,000	1.293	1780
18	67.8	63.3	100.7	115.4	65.6	84.8	19.2	964	290	4260	150.5	36.4	7.32	300,000	1.282	1945

TABLE 2

D_1 = 1.375 in. D_2 = 0.750 in. $\frac{L}{D}$ = 108.0 $\frac{D_1}{D_2}$ = 1.833 D = 0.625 in.

Heating of cooling length, L = 67 1/2 in.

Cooling Tests: Water in the Annulus being Cooled

Run No.	Water Temp. F, Annulus Out	Annulus In	Inner Pipe Out	Inner Pipe In	Aver. Water Temp. in Annulus, F	Aver. Inner Pipe Wall Temp. F	Temp. Diff. t F	Water Rate in Annulus lb hr^{-1}	Water Rate in Inner Pipe lb hr^{-1}	Heat Transferred (Based on readings on Inner Pipe) B hr^{-1}	Heat Transfer Coefficient h B$hr^{-1}ft^{-2}$F	Nu $(\frac{hD}{k})$	Pr $(\frac{c_p\mu_a}{k})$	Gr $(\frac{D^3\rho^2 g\beta\Delta t}{\mu_a^2})$	$\frac{\mu_a}{\mu_w}$	Re $(\frac{\mu D\rho}{\mu_a})$
19	106.4	113.2	100.4	63.2	109.8	104.8	5.0	69.3	10.8	414	74.5	10.58	4.06	117,800	0.952	334
20	107.8	118.2	102.4	61.1	113.0	103.5	9.5	81.8	19.5	805	76.4	10.80	4.05	244,000	0.907	408
21	109.4	126.1	99.8	63.0	117.8	103.5	14.3	85.5	35.5	1305	82.8	11.63	3.72	425,000	0.866	446
22	111.0	121.9	100.5	61.1	116.5	103.5	13.0	135.9	34.5	1359	94.9	13.36	3.77	370,000	0.880	699
23	109.2	118.8	97.4	62.3	114.0	99.7	14.3	187.3	48.2	1690	107.0	15.10	3.88	381,000	0.865	942
24	107.7	118.1	92.9	61.3	112.9	97.4	15.5	227	68.8	2170	126.4	17.82	3.92	399,000	0.848	1180
25	111.0	120.9	92.3	62.7	116.0	98.6	17.4	255	80.8	2390	124.2	17.50	3.80	491,000	0.837	1310
26	116.8	121.9	107.1	61.9	119.4	109.5	9.9	310	33.5	1511	137.7	19.32	3.66	307,000	0.910	1640
27	112.8	121.6	95.2	61.5	117.2	99.0	18.2	352	86.8	2920	145.0	20.3	3.75	531,000	0.831	1823
28	114.2	123.9	95.1	61.8	119.1	99.5	19.6	383	101.6	3380	156.3	21.9	3.66	604,000	0.820	2020
										Heating Tests:	Water in the Annulus Being Heated					
29	74.2	64.9	79.1	112.8	69.6	80.0	10.4	91.8	27.3	925	80.1	12.00	6.88	49,400	1.147	276
30	71.5	64.8	75.8	110.4	68.2	73.4	9.2	126.3	24.1	834	81.6	12.27	7.02	39,800	1.128	374
31	69.7	64.8	73.5	110.8	67.3	74.9	7.6	158.8	19.5	728	86.5	13.21	7.12	31,800	1.104	482
32	73.9	66.2	85.9	111.3	70.1	82.1	12.0	178.8	50.3	1278	96.1	14.41	6.83	58,300	1.168	542
33	71.7	65.9	85.1	113.3	68.8	82.3	13.5	252	52.7	1486	99.6	14.97	6.95	61,300	1.190	751
34	71.4	65.8	89.7	111.0	68.6	82.8	14.2	303	82.5	1760	112.0	16.83	6.97	64,000	1.200	902
35	71.3	66.1	94.0	112.0	68.7	83.8	15.1	388	112.9	2030	122.0	18.33	6.97	68,400	1.212	1156
36	70.4	64.5	95.4	113.4	67.5	85.3	17.8	440	146.0	2630	133.5	20.2	7.10	75,000	1.260	1288
37	70.0	63.5	96.1	110.2	66.8	87.4	20.6	543	246	3460	152.0	22.9	7.18	83,500	1.302	1575
38	69.6	63.7	95.6	109.7	66.7	87.2	20.5	650	265	3730	164.5	24.8	7.20	82,800	1.302	1881

TABLE 3

D_1 = 1.375 in. D_2 = 1 in. $\frac{L}{D}$ = 179.8 $\frac{D_1}{D_2}$ = 1.375 D = 0.375 in.

Heating or cooling length, L = 67 1/2 in.

Cooling Tests: Water in the Annulus Being Cooled

Run No.	Water Temp. F, Annulus Out	Annulus In	Inner Pipe Out	Inner Pipe In	Aver. Water Temp. in Annulus, F	Aver. Inner Pipe Wall Temp. F	Temp. Diff. t F	Water Rate in Annulus lb hr^{-1}	Water Rate in Inner Pipe lb hr^{-1}	Heat Transferred (Based on Readings on Inner Pipe) B hr^{-1}	Heat Transfer Coefficient h B$hr^{-1}ft^{-2}$F	Nu $(\frac{hD}{k})$	Pr $(\frac{c_p\mu_a}{k})$	Gr $(\frac{D^3\rho^2 g\beta\Delta t}{\mu_a^2})$	$\frac{\mu_a}{\mu_w}$	Re $(\frac{\mu D\rho}{\mu_a})$
39	95.9	117.5	84.2	60.0	106.7	91.8	14.9	68.1	56.1	1360	62.0	5.30	4.20	68,500	0.848	285
40	99.3	120.8	85.7	59.0	110.5	92.3	18.2	86.7	69.1	1844	68.8	5.86	4.05	93,100	0.825	375
41	105.3	123.7	90.8	59.7	114.5	97.6	16.9	110.0	59.3	1843	74.1	6.27	3.85	98,200	0.840	497
42	98.8	115.4	76.9	58.9	107.1	90.4	16.7	158.9	137.4	2440	99.0	8.47	4.18	77,000	0.830	669
43	99.5	115.5	73.6	58.9	107.5	87.7	19.8	282	289	4250	145.3	12.41	4.16	93,100	0.801	1191
44	105.7	116.6	82.4	58.8	111.2	95.6	15.6	340	151.3	3570	156.0	13.06	3.93	80,000	0.845	1490
45	110.0	121.1	84.9	59.0	115.6	99.1	16.5	362	157.5	4080	168.5	14.30	3.82	98,500	0.848	1650
46	113.2	122.2	82.2	60.5	117.7	104.3	13.4	400	155.1	3370	171.0	14.41	3.73	85,600	0.876	1863
47	109.9	120.1	80.0	59.5	115.0	98.4	16.6	443	211	4320	176.2	14.92	3.84	98,200	0.843	2010
										Heating Tests:	Water in the Annulus Being Heated					
48	72.2	61.1	75.5	112.1	66.7	75.7	9.0	78.4	22.1	810	60.8	5.51	7.18	7,860	1.126	202
49	75.6	64.9	91.2	111.2	70.3	82.6	12.3	111.2	85.6	1265	69.8	6.29	6.80	13,200	1.171	302
50	70.8	63.2	85.5	112.4	67.0	78.4	11.4	185	56.3	1520	90.7	8.21	7.16	10,070	1.161	482
51	71.2	63.5	96.1	112.1	67.4	80.6	13.2	235	120.6	1930	99.0	8.96	7.12	11,980	1.190	615
52	69.6	62.6	94.1	109.5	66.1	79.7	13.6	269	132.0	2030	101.7	9.22	7.23	11,490	1.195	693
53	68.3	62.6	93.9	110.0	65.4	77.6	12.2	339	127.3	2050	114.3	10.40	7.35	9,890	1.175	864
54	68.8	61.7	97.4	112.5	65.3	79.6	14.3	486	223	3350	159.0	14.45	7.36	11,500	1.210	1235
55	67.2	61.7	96.2	112.0	64.5	77.5	13.0	705	233	3680	192.0	17.48	7.48	11,890	1.190	1780
56	65.6	61.2	96.1	113.4	63.4	75.8	12.4	840	208	3600	197.1	18.00	7.57	8,970	1.180	2080

TABLE 4

D_1 = 2 in. D_2 = 0.750 in. $\frac{L}{D}$ = 54.0 $\frac{D_1}{D_2}$ = 2.66 D = 1.25 in.

Heating or cooling length, L = 67 1/2 in.

Cooling Tests: Water in the Annulus Being Cooled

Run No.	Water Temp. F, Annulus Out	Annulus In	Inner Pipe Out	Inner Pipe In	Aver. Water Temp. in Annulus, F	Aver. Inner Pipe Wall Temp. F	Temp. Diff. t F	Water Rate in Annulus lb hr^{-1}	Water Rate in Inner Pipe lb hr^{-1}	Heat Transferred (Based on readings on Inner Pipe) B hr^{-1}	Heat Transfer Coefficient h B$hr^{-1}ft^{-2}$F	Nu $(\frac{hD}{k})$	Pr $(\frac{c_p\mu_a}{k})$	Gr $(\frac{D^3\rho^2 g\beta\Delta t}{\mu_a^2})$	$\frac{\mu_a}{\mu_w}$	Re $(\frac{\mu D\rho}{\mu_a})$
57	103.3	118.0	96.9	62.7	110.6	97.1	13.5	57.3	22.9	785	52.6	14.92	4.02	2,610,000	0.865	216
58	107.6	114.6	99.5	64.0	111.1	103.5	7.6	81.2	15.0	532	63.0	17.88	4.00	1,492,000	0.925	307
59	107.0	115.3	95.5	63.1	111.1	100.4	10.7	129.0	31.1	1006	85.1	24.1	4.00	2,100,000	0.897	488
60	107.5	117.9	91.7	61.8	112.7	97.6	15.1	140.3	46.1	1380	82.8	23.5	3.93	3,100,000	0.852	540
61	109.9	119.2	93.7	61.7	114.5	100.7	13.8	191.0	51.0	1630	107.0	30.2	3.85	2,980,000	0.871	747
62	113.1	119.8	97.5	62.6	116.4	103.7	12.7	258	45.0	1570	111.8	31.4	3.78	2,910,000	0.881	1027
63	113.3	120.2	92.9	62.5	116.7	101.4	15.3	326	69.5	2110	124.7	35.1	3.76	3,550,000	0.857	1303
64	114.1	120.2	93.3	63.0	117.2	102.1	15.1	416	77.5	2350	140.9	39.6	3.75	3,520,000	0.860	1668
65	113.8	120.5	89.9	62.8	117.2	98.8	18.4	460	107.6	2920	144.1	40.6	3.75	4,270,000	0.830	1842
										Heating Tests:	Water in the Annulus Being Heated					
66	74.9	64.1	85.6	109.6	69.5	82.4	12.9	108.0	44.1	1060	74.6	22.4	6.89	484,000	1.182	251
67	71.7	64.1	83.9	107.5	67.9	80.5	12.6	154.7	50.5	1192	85.9	25.8	7.05	437,000	1.178	352
68	69.9	63.3	80.1	108.9	66.6	78.0	11.4	182.0	39.2	1135	90.5	27.3	7.20	366,000	1.162	407
69	70.7	62.7	88.3	110.8	66.7	81.1	14.4	216	72.5	1632	103.0	31.1	7.19	465,000	1.206	484
70	68.3	62.8	83.8	110.4	65.5	78.2	12.7	294	55.0	1461	104.5	31.6	7.33	384,000	1.182	648
71	69.3	61.2	96.2	110.7	65.2	87.2	22.0	377	211	3060	126.1	38.1	7.34	658,000	1.329	829
72	70.5	63.7	100.5	118.9	67.1	90.3	23.2	512	186.8	3440	134.6	40.6	7.16	765,000	1.342	1150
73	68.0	63.5	95.0	110.8	65.8	85.2	19.4	692	188.9	2980	139.1	42.1	7.30	597,000	1.284	1530
74	69.0	64.0	99.6	113.4	66.5	90.9	24.4	844	305	4210	156.0	47.1	7.22	782,000	1.362	1882

temperatures as listed in Kent's Mechanical Engineers' Handbook.

In making the correlation, an exponent of $-^1/_3$ was assumed for the L/D factor in Equation [2], and four sets of cooling-test data for the different annular combinations were selected with approximately equal Reynolds numbers. The function

$$\left(\frac{hD}{k}\right)\left(\frac{c_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D}{L}\right)^{-1/3}$$

was plotted against (D_1/D_2) values for these four sets on logarithmic paper. Different straight lines were drawn to join the four points. The slopes of the lines were found to vary from 0.8 to 1, the majority being in the neighborhood of 0.8. As a result, it was decided to use a value of 0.8.

The function

$$\left(\frac{hD}{k}\right)\left(\frac{c_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D}{L}\right)^{-1/3}\left(\frac{D_1}{D_2}\right)^{-0.8}$$

was plotted against

$$\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)\left(\frac{\mu_a}{VD\rho}\right)$$

on logarithmic paper for eight sets of cooling and heating tests on the four combinations with approximately equal Reynolds numbers. A straight line was drawn to join the points and a slope of 0.05 was found for the line.

The function

$$\left(\frac{hD}{k}\right)\left(\frac{c_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D}{L}\right)^{-1/3}\left(\frac{D_1}{D_2}\right)^{-0.8}\left[\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)\left(\frac{\mu_a}{VD\rho}\right)\right]^{-0.05}$$

was plotted against $\frac{VD\rho}{\mu}$ on logarithmic paper for all 74 runs. The plotted points fell within a strip on the graph, showing the definite existence of a correlation; however, the spread of the data was rather large.

Finally the exponent of (D/L) was changed to -0.4 and the function

$$\left(\frac{hD}{k}\right)\left(\frac{c_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D_1}{D_2}\right)^{-0.8}\left[\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)\left(\frac{\mu_a}{VD\rho}\right)\right]^{-0.05}$$

was plotted against $\frac{VD\rho}{\mu}$. A satisfactory correlation resulted.

Combining the Reynolds number, the function

$$\left(\frac{hD}{k}\right)\left(\frac{c_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D}{L}\right)^{-0.4}\left(\frac{D_1}{D_2}\right)^{-0.8}\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)^{-0.05}$$

was plotted against the Reynolds number as shown in Fig. 4.

A close examination shows that the slope of a line drawn through the cooling data is slightly steeper than a similar line for the heating data. The difference between the two is so small that it can be neglected.

The equation of the single line in Fig. 4 was found to be

$$\left(\frac{hD}{k}\right)\left(\frac{c_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D}{L}\right)^{-0.4}\left(\frac{D_1}{D_2}\right)^{-0.8}\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)^{-0.05} = 1.02\left(\frac{VD\rho}{\mu_a^2}\right)^{0.45}$$

Solving for the Nusselt number gives

$$\frac{hD}{k} = 1.02\left(\frac{VD\rho}{\mu_a}\right)^{0.45}\left(\frac{c_p\mu_a}{k}\right)^{1/3}\left(\frac{\mu_a}{\mu_w}\right)^{0.14}\left(\frac{D}{L}\right)^{0.4}\left(\frac{D_1}{D_2}\right)^{0.8}\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)^{0.05}$$

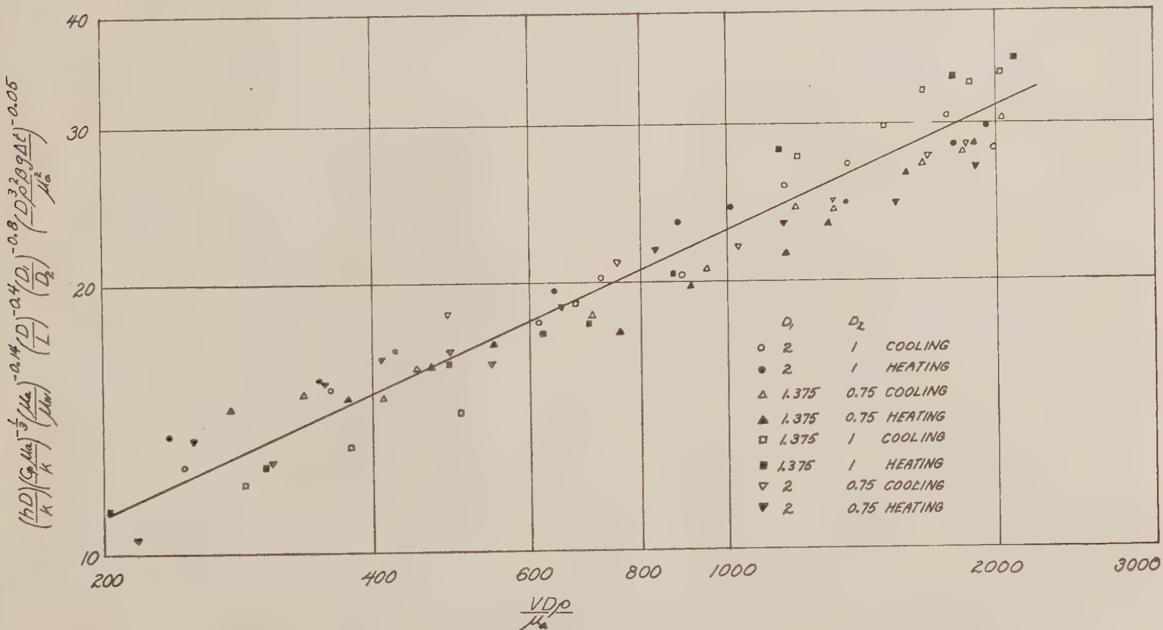

Fig. 4 Recommended Correlation of Data for Heating and Cooling Tests

Equation: $\left(\frac{hD}{k}\right)\left(\frac{C_p\mu_a}{k}\right)^{-1/3}\left(\frac{\mu_a}{\mu_w}\right)^{-0.14}\left(\frac{D_1}{D_2}\right)^{-0.8}\left(\frac{D}{L}\right)^{-0.4}\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a^2}\right)^{-0.05} = 1.02\left(\frac{DV\rho}{\mu_a}\right)^{0.45}$

Discussion and Conclusion

The following equation

$$\frac{hD}{k} = 1.02\left(\frac{VD\rho}{\mu_a}\right)^{0.45}\left(\frac{c_p\mu_a}{k}\right)^{1/3}\left(\frac{\mu_a}{\mu_w}\right)^{0.14}\left(\frac{D}{L}\right)^{0.4}\left(\frac{D_1}{D_2}\right)^{0.8}\left(\frac{D^3\rho^2\beta g\,\Delta t}{\mu_a{}^2}\right)^{0.05} \quad \ldots\ldots [3]$$

was found to correlate the test data with an average deviation of ± 6.6 per cent and a maximum deviation of $+14.1$ per cent. In view of the variables encountered in such a problem and the difficulty of making precise temperature measurements, it is considered satisfactory and is recommended for computing the heat-transfer coefficient at the inner wall of an annulus for laminar flow. Since the tests were conducted over a range of Reynolds number from 200 to 2000, the relation is only valid for this range of Reynolds number. However, as the effects on heat transfer in the laminar region should be the same for all Reynolds numbers below the lower critical value, as is the case for flow in round pipes, it can be reasonably expected that the equation may also hold for Reynolds numbers smaller than 200.

It should be noted that the exponent for (**P**) was assumed to be $^1/_3$. The Prandtl number is connected solely w th the properties of the fluid. As water was the only fluid used, the exponent for (**P**) was not verified for all fluids. Nevertheless, it is safe to presume that this exponent should be somewhere around $^1/_3$, as this was the exponent used successfully in the correlation of the heat-transfer data for both laminar and turbulent flow in round pipes, and for turbulent flow in annuli.

The Grashof number (**G**) in Equation [3] accounts for the effect of natural convection. This effect should decrease with an increase in the flow rates or Reynolds number. This was proved qualitatively by the temperature-distribution data.

The temperature at any section was found to be the highest at the top and the lowest at the bottom. In 1931 Drew and Ryan (15) studied the distribution of heat about the circumference of a pipe in a stream of fluid flowing across the pipe. Baker and Mueller (16) investigated the temperature variation around the perimeter of a tube when pure and mixed vapors were condensing. They found that the top surface attained the highest temperature and the bottom the lowest. Since only four temperature readings were obtained at each section, no quantitative analysis can be made.

The percentage of difference between temperatures at the top and bottom of any one section decreased with an increase in the Reynolds number. This means that the effect of natural convection decreased for a given increase of the Reynolds number.

The end effect constitutes a variable, the influence of which could not be determined quantitatively from the experimental results. It is felt that the end effect on the heat-transfer rate is very small for these experiments. The (L/D) ratio for the four annular combinations varies from 179.8 to 54. However, the test data fell into a satisfactory correlation. This should indicate that the end effect for different (L/D) ratios within this range were of the same order of magnitude at the same Reynolds number.

The difference between the heat loss or gain for the water flowing in the annulus and the heat loss or gain for the water flowing in the inner pipe was used as a relative index of the heat balance. Due to the facts that the area of the external insulation surface was not known accurately, and that the temperature differences between the outer insulation surface and the air were small, no effort was made to compute the heat-transfer loss by convection and radiation from the insulation surface. Approximate computations could be made for this loss, which when taken into account would decrease the difference between the heat gained or lost by the fluid in the annular space and the heat gained or lost by the fluid in the inner pipe. It was felt that the difference between the heat gained or lost by the water in the annular space and that in the pipe would serve as an index of the reliability of the tests. The average difference for all of the tests was approximately 5.8 per cent. The results obtained on the first test conducted showed a difference of 31.6 per cent. The data for one other test checked to within 16.9 per cent. The differences for seventy-two tests were less than 11 per cent, the average for these tests being 5.22 per cent.

The heat-transfer coefficients for the inner pipe were computed in order to check the experimental values. These data correlated in a satisfactory manner with those already published.

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(Owing to emergency travel conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Heat Transfer in the Locomotive Boiler

By LAWFORD H. FRY,[1] NEW YORK, N. Y.

It is shown that the total heat absorption in a locomotive boiler for any assumed rate of heat release and weight of gases of combustion can be predicted with all of the accuracy practically necessary. The over-all heat transfer is considered as taking place in two separate operations which can be represented by two formulas, a fourth-power equation of the usual type for the radiation in the firebox, and a double logarithmic equation for the convection in the flues. This form of equation was suggested by Fessenden (1),[2] and its usefulness has been extended by the coefficients developed by the present author. It may not represent the exact physical reactions of the process of heat transfer, but it has the pragmatic sanction of giving practically accurate results under many different conditions. Actually that is all that can be claimed for any other heat-transfer formula even if it uses the Reynolds number. Fessenden pointed out, "It should be noted that the apparently rational idea that the rate of heat transfer is directly proportional to the temperature difference is, after all, only an assumption that has been accepted for so long that it has come to be regarded as an axiomatic statement of fact."

THE locomotive boiler with its water-enclosed firebox and its fire-tube barrel still follows the lines laid down by George and Robert Stephenson when they built the *Rocket* in 1829. This survival is sometimes criticized by advocates of other forms of motive power who think that we should be modern at all costs. Actually the retention of the locomotive-type boiler is the result of hard common sense supported by the inherent advantages of the arrangement. Change in design is not necessarily good engineering. Retention of a circular form for a wheel is still good practice.

Study of the details of operation of the locomotive boiler leads inevitably to recognition of the fact that in its modern form the locomotive boiler is an extremely compact and effective mechanism for the production of steam. Some efficiency of combustion is deliberately sacrificed to obtain the high rate of heat release necessary. It must be remembered that the boiler is part of a mobile power plant capable of running at 100 mph and able to pass through the restricted limits of a railroad loading gage. Evaporative capacities of over 100,000 lb of superheated steam per hr are obtained, and heat-release rates up to 300,000 Btu per hr per cu ft of firebox volume may be reached at maximum outputs. It is obvious that such rates of heat release are not compatible with high combustion efficiencies. The efficiency of heat absorption with which this paper is chiefly concerned is, however, generally satisfactory.

In studying the processes by which part of the heat released in the firebox is transferred to the boiler heating surfaces and taken up by the water and steam, the action may be considered as taking place in two major and individual operations. In the firebox, part of the heat released is taken up by direct radiation, while the remainder is carried into the tubes and flues by the gases of combustion. Of this gas-carried heat, part is transferred to the evaporative and to the superheating surfaces by convection while the remainder is lost through the smokebox as sensible and latent heat in the gases of combustion.

The purpose of this paper is to present formulas which represent the two processes of heat transfer and to show how they can be combined for use in estimating the amount of heat that will be taken up by a boiler of given dimensions under any known or assumed operating conditions.

In applying the formulas the boiler design is characterized by the area of the firebox heating surface, and by the number, diameters, and lengths of the flues, tubes, and superheater pipes. The operating conditions are characterized by the rate of heat release and by the weight of gases of combustion produced per hour. These dimensions and conditions are necessary and sufficient for determining the rate at which the boiler will take up heat to be used in producing and superheating the steam.

Over-All Heat Transfer

Computation of the over-all heat transfer falls into two parts. In the first the rate of heat release, the rate of gas production, and the area of the firebox surface are used to find the so-called "equilibrium temperature" of the firebox. This is the temperature at which the sum of the heat radiated direct to the firebox surface and of the sensible heat carried by the gases at that temperature is just equal to the total available heat released by combustion.

In the second part of the computation the gases are assumed to enter the flue bundle at the equilibrium temperature and to lose temperature at a rate given by the double logarithmic formula, Equation [2]. This equation takes into account the rate of gas flow, the gas-swept perimeter and the free gas area of the flue bundle, the temperature of the receiving surface, and the length of the flues.

Details of the formulas and of their application in practice are given later. Attention is here directed to the results obtained by applying the formulas to four different locomotive boilers of the dimensions shown in Table 1, for which extensive boiler test results are available. Outlines of two of the boilers are shown in Fig. 1. Two boilers have Type A superheaters with 4 pipes in each flue, while the other two boilers have Type E superheaters

TABLE 1 PRINCIPAL DIMENSIONS OF BOILERS

Test series	A	B	C	D
Flues, number, and OD in in...	69—5 1/2	198—3 1/2	170—3 1/2	40—5 1/2
Tubes, number, and OD in in..	184—2 1/4	50—2 1/4	120—2 1/4	236—2 1/4
Length between flue sheets, ft..	17.9	20.4	18.9	19.1
Superheater type..........	AS	ED	E	A
Superheater pipes OD, in......	1 1/2	1 3/16	1 3/16	1 1/2
Superheater pipes, number in flue..................	4	2	2	4
Evaporative surface (fireside), sq ft..................	3719	3986	3908	3686
Superheater surface (fireside), sq ft..................	2029	2640	2052	1205
Gas-swept perimeter (p), in....	3477	3820	3750	3068
Free gas area (a), sq in.......	1522	1368	1389	1382
Ratio P/a..................	2.28	2.80	2.70	2.22
Firebox volume (net), cu ft....	613	391	475	410
Firebox heating surface, sq ft..	490	373	406	311
Grate area, sq ft............	92.0	75.8	70.0	69.3

[1] Director of Research, Steam Locomotive Research Institute, Inc. Fellow A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

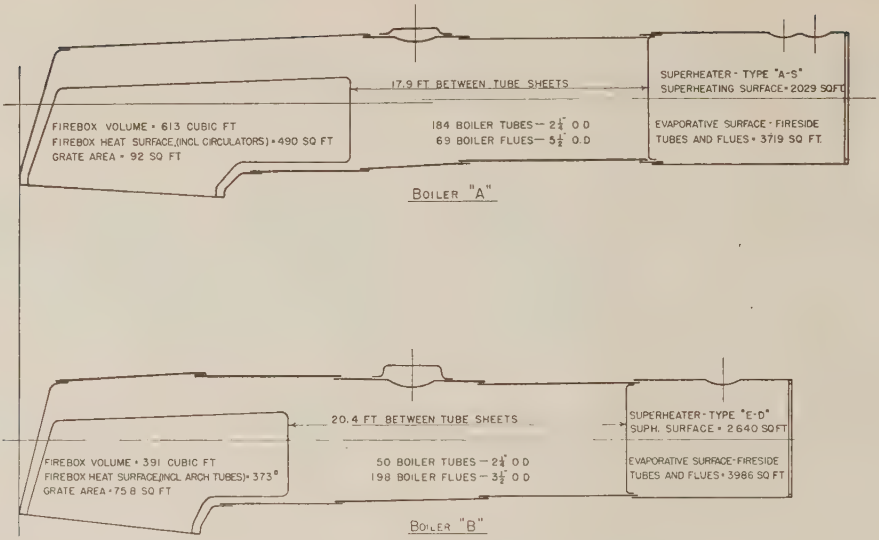
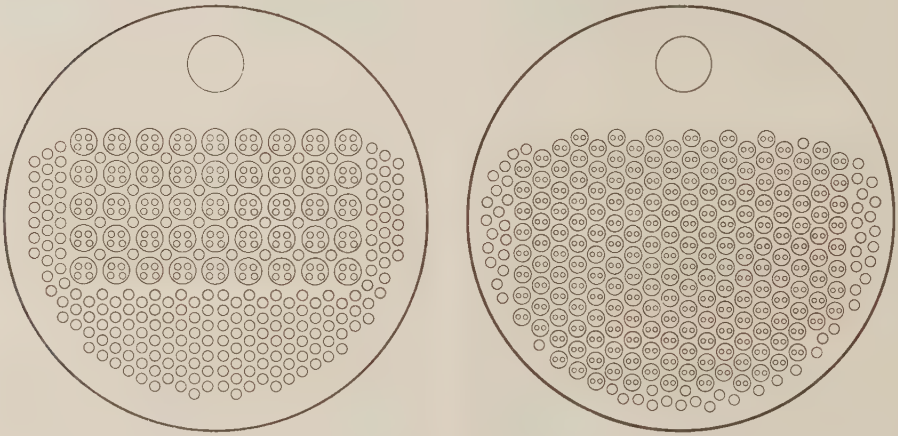


Fig. 1 Comparison of Boiler Dimensions



Boiler "A" Boiler "B"

Fig. 2 Representative Arrangements of Superheater Flues and Tubes

(Boiler "A," equipped with Type A superheater with 4 pipes in each flue; boiler "B," equipped with Type E superheater with 2 superheater pipes in each flue.)

with 2 superheater pipes in each flue. Representative arrangements of the superheater flues and the tubes are shown in Fig. 2. Computations for heat transfer have been carried out for twenty-seven tests taken largely at random. The test data for rate of heat release and weight of gases of combustion have been used, and the rates of heat transfer in firebox and in flues and the resulting smokebox temperatures have been computed by means of the formulas. The results are given in Tables 2 and 3 for comparison with the measured values. Agreement between measured and computed values is generally satisfactory, as the greatest divergence is hardly greater than the element of uncertainty in the measured values.

Heat Distribution

Table 2 includes the experimental data to which the formulas are applied and gives the computed and measured firebox and smokebox temperatures. The relation between measured and computed temperatures is also shown in Figs. 3 and 4.

Table 3 is based on the same data, measured and computed, that are contained in Table 1, but the data are organized to show how the heat released in the firebox is distributed. It may be noted that rates of firing range from the low rate of 35 lb of dry coal per sq ft of grate area per hr up to the high rate of 242 lb per sq ft per hr. Column 4 gives the computed equilibrium temperature, and column 5 the corresponding rate of radiation per square foot of firebox heating surface. This rate of radiation is given in column 5 in terms of Btu per hour and in column 6 is translated into equivalent evaporation in pounds of water per hour from and at 212 F per square foot of firebox surface. These values run up to 126 lb per sq ft per hr at the maximum firing rates. At normal working rates they are about 90 to 100 lb per sq ft per hr. There is no direct method of checking these

TABLE 2 EXPERIMENTAL DATA AND COMPARISON OF COMPUTED VALUES WITH MEASURED VALUES FOR FIREBOX AND SMOKEBOX TEMPERATURES

Series	Test no.	Firing rate, lb DCF per SFG per hr	Firebox temperature, F — Measured — Center	Firebox temperature, F — Measured — Flue sheet	Firebox temperature, F — Computed equilibrium temperature	Smokebox temperature, F — Measured	Smokebox temperature, F — Computed	Per sq ft firebox heating surface — Heat released, Btu per hr	Per sq ft firebox heating surface — Weight gases of combustion, lb per hr
1	2	3	4	5	6	7	8	9	10
A	1441	47	2491	1530	1840	576	570	109 500	126
	1440	68	2476	1579	1950	617	605	140 000	153
	1436	102	2643	2083	2090	686	664	208 000	235
	1437	123	2630	2163	2075	713	683	219 000	254
	1439	151	2566	1858	2230	741	721	271 000	288
	1438	182	2734	2058	2300	768	760	312 000	324
	1444	207	2642	2058	2380	775	760	334 000	324
	1442	242	2703	2150	2490	807	780	367 000	328
B	4F	56	..	1540	1915	526	515	147 000	175
	4D	62	..	1530	1970	562	525	161 000	188
	7F	85	..	1600	2120	604	557	223 000	250
	7H	97	..	1750	2190	607	562	237 000	251
	10C	114	..	1710	2260	655	595	285 000	300
	10D	115	..	1780	2260	628	585	274 000	282
	10E	134	..	1940	2320	642	590	304 000	307
	13G	139	..	1940	2410	660	600	314 000	290
	13M	161	..	2160	2400	667	607	344 000	334
	13F	182	..	2040	2480	693	620	394 000	374
	13K	208	..	2350	2500	660	634	405 000	376
C	517A	58	2173	1647	1700	507	540	104 500	148
	510A	81	2123	1692	1900	549	565	150 000	185
	519A	122	2067	1762	2015	590	612	204 000	248
	527A	175	2369	1669	2320	647	660	274 000	327
D	607A	35	1750	..	1715	500	505	86 500	106
	623A	107	2324	1646	2070	632	645	242 000	298
	651A	139	2466	1895	2310	676	690	312 000	323
	610A	173	..	..	2485	698	715	378 000	35)

NOTE: DCF = dry coal fired. SFG = square feet grate area.

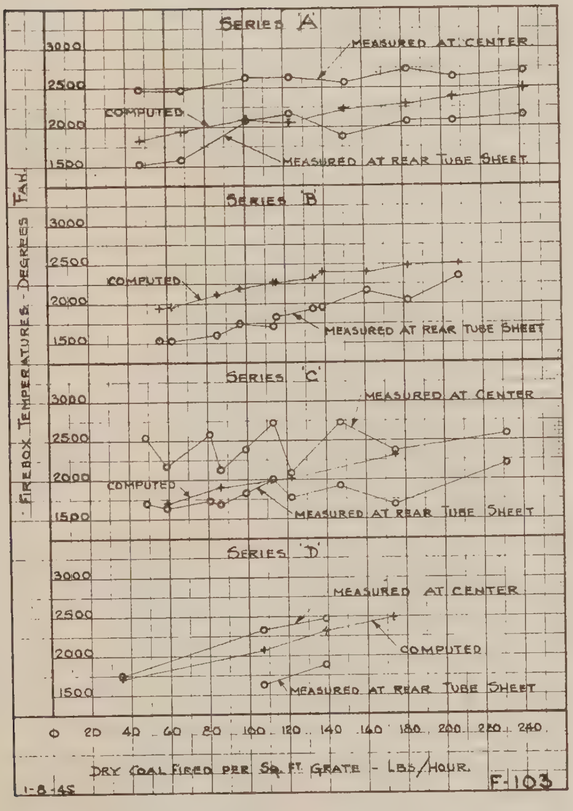


FIG. 3 RELATION BETWEEN MEASURED AND COMPUTED TEMPERATURES

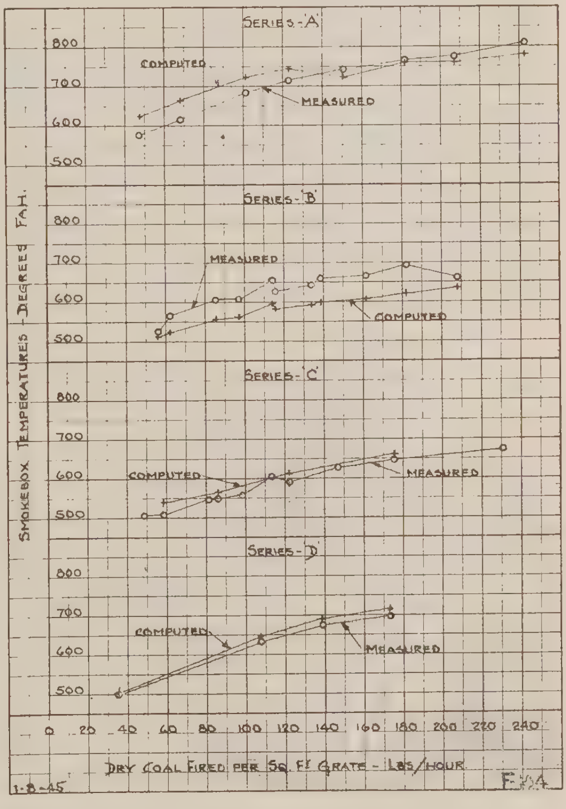


FIG. 4 RELATION BETWEEN MEASURED AND COMPUTED TEMPERATURES

TABLE 3 COMPUTED RADIATION AND EQUIVALENT EVAPORATION IN FIREBOX TOGETHER WITH PERCENTAGE DISTRIBUTION OF HEAT RELEASED

Series	Test no.	Firing rate, lb DCF per SFG per hr	Computed equilibrium temperature, F	Heat radiated, equilibrium temperature, Btu per hr per sq ft	Equivalent firebox evaporation, lb per sq ft per hr	Heat distribution in per cent of heat released — Absorbed — Firebox	Absorbed — Flues, tubes, superheater	Lost in smokebox — Sensible heat	Lost in smokebox — Latent heat
1	2	3	4	5	6	7	8	9	10
A	1441	47	1840	44 000	45	40.0	41.0	15.5	3.5
	1440	68	1950	53 000	55	38.0	42.0	15.7	4.3
	1436	102	2090	67 000	69	32.2	45.2	18.8	3.8
	1437	123	2075	65 000	67	30.0	45.5	20.0	4.5
	1439	151	2230	83 000	86	30.6	45.4	19.2	4.8
	1438	182	2300	92 000	95	29.6	45.1	20.5	4.8
	1444	207	2380	103 000	106	30.8	45.8	18.3	5.1
	1442	242	2490	120 000	123	32.6	44.1	17.8	5.5
B	4F	56	1915	51 000	52	34.6	47.0	15.0	3.4
	4D	62	1970	55 000	57	34.2	46.6	15.5	3.7
	7F	85	2120	70 000	72	31.4	48.9	15.1	3.6
	7H	97	2190	78 000	80	32.9	48.1	15.2	3.8
	10C	114	2260	87 000	90	30.5	49.0	16.5	4.0
	10D	115	2260	87 000	90	31.7	49.0	15.3	4.0
	10E	134	2320	95 000	98	31.2	49.4	15.5	3.9
	13G	139	2410	108 000	110	34.4	46.8	14.7	4.1
	13M	161	2400	106 000	109	30.8	49.0	15.7	4.5
	13F	182	2480	119 000	123	30.2	49.8	15.7	4.3
	13K	208	2500	122 000	126	30.2	50.4	14.6	4.8
C	517A	58	1700	33 000	34	31.6	47.4	16.7	4.3
	510A	81	1900	49 000	50	32.6	47.3	16.0	4.1
	519A	122	2015	59 500	61	29.2	49.2	17.2	4.4
	527A	175	2320	95 000	98	34.6	42.4	18.2	4.8
D	607A	35	1715	35 000	36	40.7	41.0	14.2	4.1
	623A	107	2070	65 000	67	26.8	50.5	18.6	4.1
	651A	139	2310	93 000	96	29.8	49.0	17.0	4.2
	610A	173	2485	120 000	124	31.7	48.2	15.6	4.5

NOTE: DCF = dry coal fired. SFG = square foot grate area.

figures with evaporations actually obtained, but they are in line with values usually accepted.

Columns 7, 8, 9, 10 split up heat released into percentages representing, respectively, in column 7 the heat radiated to the firebox at equilibrium temperature, in column 8 the heat taken up by the evaporative and superheater surface of the tube bundle in cooling the gases from equilibrium to smokebox temperature, in column 9 the heat lost in sensible heat of gases at smokebox temperature, and in column 10 the heat lost in latent heat of the vapor. Except for high values at the very low firing rates, the distribution of the heat shows no great variation with the rate of firing and is very similar for the boilers of Series B, C, and D. The boiler of Series A with shorter flues shows a slightly lower percentage of heat absorbed in the flues.

Mean Firebox Equilibrium Temperature

The "mean equilibrium temperature" is defined as the temperature at which the amount of heat carried by the combustion gases, together with the amount radiated to the firebox surface, is just equal to the quantity of heat released by combustion. Its value in any given case can be found by using Table 4 or Fig. 5, provided that the experimental data furnish information as to the rate of available heat released in the firebox as well as the weight of the gases of combustion. Table 4 covers temperatures from 1400 F to 2800 F by steps of 10 deg F and for each temperature gives the sensible heat carried by the gas as well as the rate in Btu per square foot per hour at which radiated heat will be taken up by the firebox heating surface. The sensible heat in Btu above 32 F carried by 1 lb of gas is derived from the values given by Heck (6) for the products of combustion of coal with 20 per cent excess air. The original data show that the percentage of excess air has little effect on the specific heat of the mixed gases of combustion and that the figures of Table 4 are satisfactory in practice for any gas composition encountered in a coal-burning locomotive.

The values for radiated heat absorbed by the firebox heating surface are derived from the so-called Stefan-Boltzmann fourth-power equation

$$H_R = 1600\ [(T/1000)^4 - (t/1000)^4] \quad \ldots\ldots\ldots [1]$$

where

H_R is heat absorbed by radiation, in Btu per hour per square foot of firebox surface

T is absolute temperature of flame in firebox, in degrees Rankine

t is absolute temperature of receiving surface, in degrees Rankine. The temperature of the firebox surface is taken to be the temperature of the water back of the firebox surface.

Table 4 is computed for a water temperature of 380 F (200 psi) and can be used for any pressures normally encountered in locomotive practice. An increase of temperature to 600 F (1600 psi) would reduce the tabular values by 1000 Btu per sq ft per hr, which is negligible except for the lowest transmitting temperatures.

The Stefan-Boltzmann fourth-power relation is generally accepted as representative of radiation heat transfer and the factor 1600 has been found to give satisfactory results for the conditions encountered in locomotive fireboxes.

Fig. 5 has been plotted to answer graphically any question as to the mean equilibrium temperature for any combination of heat available and weight of gas per square foot of heating surface per hour.

In using Table 4 and Fig. 5, computations must be based not on total heat released, but on heat available, exclusive of the latent heat of the vapor. Usually the value given in the experimental data for heat released will be based on the upper calorific value of the fuel and will therefore include the heat involved in vaporizing the H_2O produced by combustion. The latent heat of this vapor is not available for radiation and is not included in the values for gas-carried heat in Table 4. It must therefore be deducted from the heat released to find the "heat available." Each pound of hydrogen burned produces 9 lb of vapor which will account for 8730 Btu of latent heat. Consequently, for each per cent of hydrogen in the fuel, 87.3 Btu must be deducted from the total heat released per pound of fuel to find the "net available heat" to be used in computing the equilibrium temperature.

For example, in test No. 1439 in Series A with 288 lb of gas pro-

TABLE 4 RADIATION AND HEAT-CONTENT FACTORS FOR DETERMINING FIREBOX TEMPERATURE WITH COAL FUEL AND A RECEIVING SURFACE TEMPERATURE OF 380 F[a]

	1400 F		1500 F		1600 F		1700 F		1800 F		1900 F		2000 F	
	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.
0F	18200	363.3	22800	392.5	28000	421.9	34000	451.6	40890	481.5	48800	511.6	57800	541.9
10F	18700	366.2	23300	395.4	28500	424.9	34700	454.6	41600	484.5	49600	514.6	58700	545.0
20F	19200	369.1	23800	398.3	29100	427.8	35300	457.5	42400	487.5	50500	517.7	59700	548.0
30F	19600	372.0	24300	401.3	29700	431.1	36000	460.5	43200	490.5	51400	520.7	60700	551:1
40F	20000	374.9	24800	404.2	30300	433.7	36700	463.5	44000	493.5	52200	523.7	61700	554.1
50F	20500	377.8	25300	407.1	30900	436.7	37300	466.5	44700	496.5	53100	526.7	62700	557.2
60F	20900	380.8	25800	410.1	31500	439.7	38000	469.5	45500	499.5	54000	529.8	63700	560.2
70F	21400	383.7	26300	413.1	32100	442.6	38700	472.5	46300	502.6	54900	532.8	64700	563.3
80F	21800	386.6	26900	416.0	32700	445.6	39400	475.5	47100	505.6	55900	535.9	65800	566.3
90F	22300	389.5	27400	418.9	33300	448.6	40200	478.5	47900	508.6	56800	538.9	66800	569.4
100F	22800	392.5	28000	421.9	34000	451.6	40890	481.5	48800	511.6	57800	541.9	67900	572.5

	2100 F		2200 F		2300 F		2400 F		2500 F		2600 F		2700 F	
	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.	Rad. factor	Heat cont.
0F	67900	572.5	79300	603.2	92000	634.1	106200	665.2	122000	696.3	139500	727.6	158700	759.1
10F	69000	575.5	80500	606.3	93400	637.2	107700	668.3	123600	699.4	141300	730.8	160700	762.3
20F	70100	578.6	81700	609.3	94700	640.3	109200	671.4	125400	702.6	143200	733.9	162800	765.4
30F	71200	581.6	83000	612.4	96100	643.4	110800	674.5	127100	705.7	145100	737.1	164900	768.5
40F	72300	584.7	84200	615.5	97500	646.5	112300	677.6	128800	708.8	147000	740.2	167000	771.7
50F	73400	587.8	85500	618.6	98900	649.6	113900	680.7	130500	711.9	148900	743.3	169100	774.9
60F	74600	590.9	86800	621.7	100400	652.7	115500	683.8	132300	715.1	150800	746.5	171200	778.0
70F	75700	593.9	88000	624.8	101800	655.8	117100	687.0	134100	718.2	152700	749.7	173300	781 2
80F	76900	597.0	89300	627.9	103300	658.9	118700	690.1	135900	721.4	154700	752.8	175500	784.3
90F	78100	600.1	90700	631.0	104700	662.1	120400	693.2	137700	724.5	156700	755.9	177700	787.5
100F	79300	603.2	92000	634.1	106200	665.2	122000	696.3	139500	727.6	158700	759.1	....	790.7

[a] If the temperature of the receiving surface is increased to 600 F, the tabulated figures should be reduced by 1000 Btu.

NOTE: The "radiation factor" represents, for each temperature, the rate at which heat measured in Btu per hour is taken up by each square foot of firebox surface.

The "heat-content factor" gives the amount of sensible heat carried by each pound of the mixed gases of combustion. Latent heat is not included.

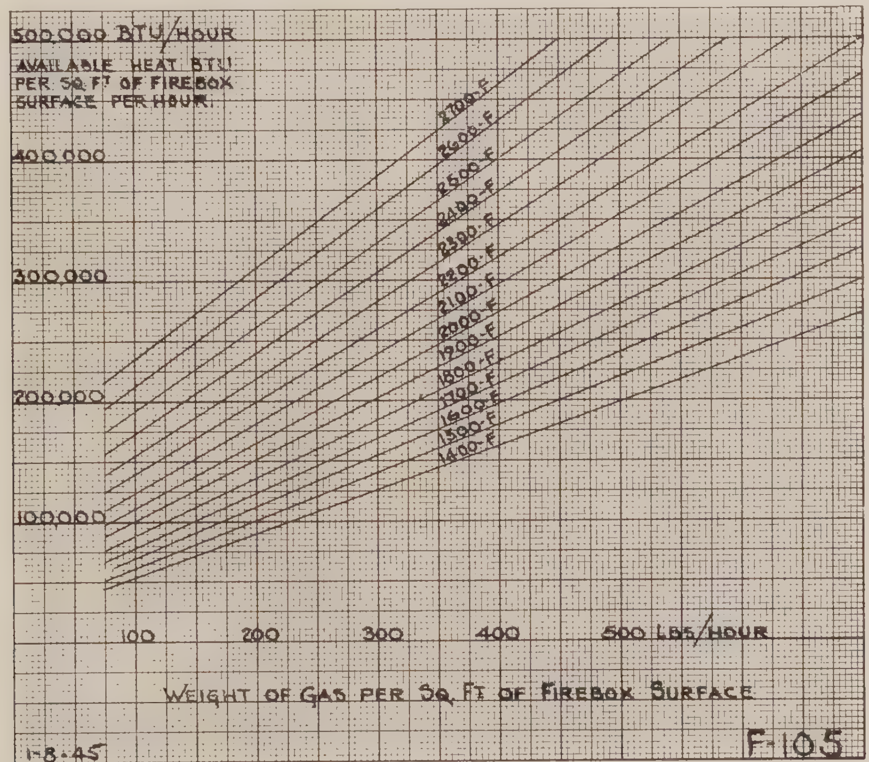


FIG. 5 EQUILIBRIUM TEMPERATURE IN RELATION TO HEAT AVAILABLE AND WEIGHT OF GAS

duced and 271,000 Btu released per sq ft of firebox heating surface per hr, the latent heat accounts for 13,000 Btu per sq ft per hr, leaving 258,000 Btu available. Inspection of Fig. 5, followed by check with Table 4, shows that at a temperature of 2230 F the heat released per hour per square foot of firebox surface will be accounted for as follows:

	Btu
Heat radiated at 2230 F	83000
Sensible heat in 288 lb of gas at 612 Btu per lb	175000
Heat available	258000
Latent heat	13000
Heat released	271000

On this basis the mean equilibrium temperature is taken to be 2230 F. It may be noted that the experimental data and the type of computation do not justify an attempt to determine the equilibrium temperature to a value closer than the nearest 10 deg F.

In test No. 1439 just quoted, the measured firebox temperatures were 2566 F at the center and 1858 F at the flue sheet. On the basis of radiation proportional to the fourth power of the absolute temperature, the mean between 2566 F and 1858 F is 2280 F, which is close to the computed mean equilibrium temperature of 2230 F. The measured values for the firebox temperatures are not sufficiently reliable to allow any final conclusions to be drawn, but it is of interest to note that where the measured values seem to be most consistent there is quite close agreement between the mean fourth-power absolute temperature and the computed equilibrium temperature. This supports the view that radiation at the mean equilibrium temperature, as given in Table 4, corresponds to the heat actually taken up by the firebox. It should be noted that the drop in firebox temperature between center and flue sheet is not due in any large degree to transfer of heat by convection. The hydraulic depth of the firebox is too large to provide any effective rate of heat transfer. The lower final temperature is due to the lower rate of heat release. With a firing rate of 100 lb of dry coal per hr, the hourly supply of combustible may be represented by 78 lb of carbon, 5 lb of hydrogen, and 248 lb of oxygen. These combustible elements are diluted by 920 lb, which is 2.8 times their weight, of inert nitrogen. As they pass through the firebox they combine until when the flue sheet is reached the combustible elements may be reduced to 16 lb of carbon and 63 lb of oxygen, diluted by 14 times their weight of inert gas consisting of nitrogen and the products of combustion. It is obvious that as these final conditions are approached the rate of heat release will be much slower than that at the center of the firebox.

Heat Transfer by Convection in Flue Bundle

The heat taken up by the flues from the gases of combustion as they pass from firebox to smokebox is found by computing the drop in temperature of the gases. The double logarithmic formula shown as Equation [2] is used for this purpose.

This form of equation, by which heat transfer is measured in temperature drop and not by the rate of transfer per square foot of surface per degree of temperature difference per hour, is due to Fessenden (1, 3, 4) who showed that it could be applied to represent satisfactorily a variety of experimental data. The author has carried the application a step further and shown (2, 5) that if the coefficient M were related to the rate of gas flow and to the hydraulic depth of the flue by Equations [3], [4a], and [4b], given later in this paper, the formula had a remarkably wide range of usefulness. It has been shown to be applicable to flues ranging in diameter from $^1/_2$ in. to 4 in. and of circular and of annular cross section, with gas temperatures ranging from room temperature up to 2500 F, and with heat transmission either from a hot gas to a cooler liquid surrounding the flue or to a cooler gas from a hot liquid. As the formula covers a broad territory with an accuracy sufficient for all practical purposes, it has useful possibilities even if its pedigree is not entirely unimpeachable from the standpoint of theoretical physics.

Fessenden pointed out that the real justification of the formula rests not so much upon the rationality of its development, as upon the manner in which it satisfies the experimental data gathered from many different sources. The fundamental assumption leading to Equation [2] might be stated in a number of ways, but the simplest is that the loss of heat from an element of gas passing along a tube takes place in accordance with the exponential formula for a damped process. This leads to the equation

$$dQ = C \cdot e^{-m.x} \qquad [1]$$

where

dQ is heat lost by element of gas traversing flue length x.

Hedrick and Fessenden (1) show that this can be developed into the form

$$\text{lolog } (T_2/t) = \text{lolog } (T_1/t) - M_x \qquad [2]$$

where

lolog means "the logarithm of the logarithm of"
T_1 is absolute equilibrium temperature in degrees Rankine
T_2 is absolute smokebox temperature in degrees Rankine
t is absolute mean effective temperature of receiving surface in degrees Rankine
x is length of flues in feet
M is coefficient depending only on mean hydraulic depth of tube bundle and on rate of gas flow per inch of perimeter of receiving surface

Using data from many sources, the author has found that the coefficient M is linked with the rate of gas flow and with the flue dimensions by the equation

$$\log M = B - m \log W/p \qquad [3]$$

where

W is total weight of gases of combustion in pounds per hour
p is total perimeter of receiving surface, including flues, tubes and superheater pipes, in inches

The auxiliary coefficients B and m depend only upon the ratio of (p), the gas-swept perimeter of the flue through which the gas flows to (a), the free area of this flue. The ratio of perimeter to area is of course the reciprocal of the hydraulic depth. It has been used as a parameter instead of the hydraulic depth because it varies in the same direction as the heat-absorbing value of a flue. A high value of the p/a ratio corresponds to a flue with a high capacity for heat absorption.

With values of p in inches and of a in square inches, values of the coefficients B and m, corresponding to the various values of the ratio p/a, are given in Table 5. These values are derived from the equations

$$\log (B - 1.3) = 9.384 - 0.54 \log (p/a) \qquad [4a]$$

$$\log m = 9.583 - 0.37 \log (p/a) \qquad [4b]$$

Equation [2] involves t the absolute temperature of the receiving surface of the flue. Experience shows that for work with locomotive boilers a satisfactory mean value can be obtained by adding to the mean value of the steam or water surface of the flue a differential which varies with the heat offered per linear inch of the gas-swept perimeter of the flue, as shown in Fig. 6.

In the case of the evaporative surface of the flues, the water temperature is that of the saturated steam. In the superheater the steam temperature varies from saturated to the final superheat, and for the whole superheater surface an arithmetic mean may be taken. When evaporative and superheater surfaces are taken together a weighted mean is taken, allowing for the surface and temperature of both surfaces. In the present work both surfaces have been taken together and only the final smokebox temperature has been computed. The earlier paper referred to shows that the equations can be applied to the evaporative and superheating surfaces separately and a step-by-step computation used to find the progressive temperature drop along the flue, and to determine separately the respective amounts of heat taken up by the evaporative and by the superheating surface.

TABLE 5 COEFFICIENTS FOR COMPUTING HEAT TRANSFER IN FLUES

(p/a = perimeter/area; B and m are coefficients)

p/a	B	m	p/a	B	m	p/a	B	m
0.50	8.867	0.494	2.00	9.053	0.296	6.20	9.350	0.194
0.51	8.868	0.491	2.10	9.062	0.291	6.40	9.362	0.192
0.52	8.870	0.487	2.20	9.071	0.286	6.60	9.373	0.190
0.53	8.872	0.483	2.30	9.080	0.281	6.80	9.384	0.188
0.54	8.874	0.480	2.40	9.089	0.277	7.00	9.394	0.186
0.55	8.876	0.477	2.50	9.098	0.273	7.20	9.404	0.184
0.56	8.878	0.474	2.60	9.108	0.269	7.40	9.414	0.183
0.57	8.880	0.471	2.70	9.116	0.265	7.60	9.423	0.181
0.58	8.881	0.467	2.80	9.124	0.261	7.80	9.433	0.179
0.59	8.883	0.464	2.90	9.132	0.258	8.00	9.444	0.177
0.60	8.884	0.461	3.00	9.140	0.255	8.20	9.455	0.176
0.65	8.892	0.449	3.20	9.154	0.249	8.40	9.465	0.174
0.70	8.900	0.437	3.40	9.170	0.242	8.60	9.474	0.172
0.75	8.908	0.426	3.60	9.185	0.237	8.80	9.484	0.170
0.80	8.915	0.416	3.70	9.199	0.233	9.00	9.494	0.169
0.85	8.922	0.406	4.00	9.213	0.229	9.20	9.504	0.168
0.90	8.929	0.390	4.20	9.227	0.225	9.40	9.513	0.166
1.00	8.942	0.383	4.40	9.241	0.221	9.60	9.522	0.165
1.10	8.955	0.370	4.60	9.254	0.217	9.80	9.532	0.164
1.20	8.968	0.358	4.80	9.266	0.214	10.00	9.542	0.163
1.30	8.980	0.347	5.00	9.278	0.211	10.20	9.550	0.162
1.40	8.991	0.337	5.20	9.290	0.208	10.40	9.557	0.161
1.50	9.002	0.329	5.40	9.303	0.205	10.50	9.553	0.160
1.60	9.013	0.322	5.60	9.315	0.203			
1.70	9.023	0.315	5.80	9.329	0.200			
1.80	9.034	0.308	6.00	9.338	0.197			
1.90	9.044	0.302						

The smokebox temperatures reported in Table 2 have been obtained by taking the combined perimeter and area and the mean temperature for evaporative and superheating surface, and by assuming that the gases entered the flues at the equilibrium temperature. Agreement between the computed and the measured values meets practical requirements.

The widest deviation is found in test No. 13 F, Series B, with a smokebox temperature of 620 F computed and 693 F measured. This represents a difference of 19 Btu of sensible heat per lb of gas, and as the gas flow is 374 lb for a heat release of 394,000 Btu per hr per sq ft of firebox surface, the difference amounts to 1.8 per cent of the heat released. This is hardly greater than the factor of uncertainty in the measured values. Agreement in the three other series of tests is considerably closer so it is felt that the proposed method can be used with confidence in predicting the over-all heat-absorbing capacity of a locomotive boiler. It should be noted that in the computation the gases of combustion are assumed to leave the firebox and to enter the flues at the equilibrium temperature, while in reality the gases at the flue sheet are probably from 100 to 200 deg F below the equilibrium temperature. The justification for this is that the formula as set up gives satisfactory results and that, in estimating the heat transfer, the equilibrium temperature can be computed while the actual flue-sheet temperature is not known.

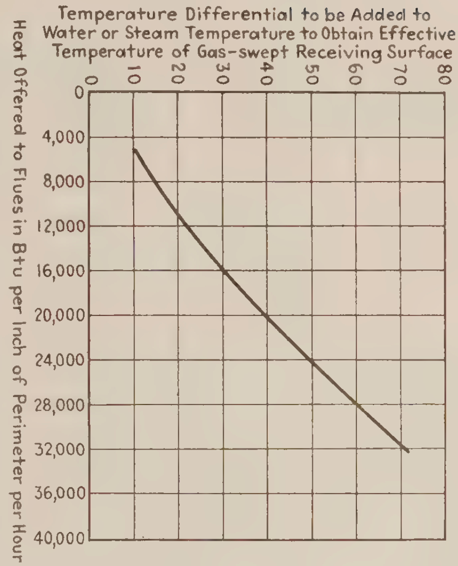


FIG. 6 TEMPERATURE DIFFERENTIAL FOR FULL LENGTH FLUES

If a close study of the heat transfer in the flues were to be made, it would be desirable to adjust the differential given in Fig. 6, so that it would correspond to heat transfer from the exact initial flue-gas temperature. This would require an increase of about 60 per cent in the values of the temperature differential given in Fig. 6.

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(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

The Influence of Tube Shape on Heat-Transfer Coefficients in Air to Air Heat Exchangers

By F. H. Green[1] and L. S. King,[1] Los Angeles, Calif.

Heat-transfer coefficients are reported for air passing through round tubes, through partially flattened tubes, and through partially flattened tubes which have also been dimpled. It was found that flattened tubes maintained fully turbulent flow and, consequently, a higher heat-transfer coefficient at lower Reynolds number values than did round tubes. The addition of dimples to the flattened tubes maintained fully turbulent flow from the transition point and increased the heat-transfer coefficient substantially. The addition of the dimples also improved the relation between heat transfer and pressure drop in the flattened tubes.

Nomenclature

The following nomenclature is used in the paper:

A = tube internal cross section area, sq ft
A_i = total area of inside surface of tubes in unit, sq ft
A_0 = total area of outside surface of tubes in unit, sq ft
D_H = hydraulic diameter, ft
D_0 = outside diameter of tube, ft
E_i = effectiveness on air inside tubes $= \dfrac{t_1 - t_2}{t_1 - t_3}$
E_0 = effectiveness on air outside tubes $= \dfrac{t_4 - t_3}{t_1 - t_3}$
G = mass velocity, lb/(hr)(sq ft)
h_i = coefficient of heat transfer inside tubes, Btu/(hr)(sq ft)(deg F)
h_0 = coefficient of heat transfer outside tubes, Btu/(hr)(sq ft)(deg F)
i_1 = enthalpy of steam, Btu per lb
i_2 = enthalpy of condensate, Btu per lb
k = thermal conductivity of fluid, Btu/(hr)(sq ft)(deg F per ft)
L = length of tube or tube section, ft
$\mathbf{N}$ = Nusselt number $= \dfrac{hD_H}{k_{avg}}$
$\mathbf{R}$ = Reynolds number $= \dfrac{D_H\, G_{max}}{\mu_{avg}}$
$\sigma \Delta P$ = air pressure drop corrected to standard conditions at center of unit, in. H_2O
q = rate of heat transfer, Btu per min
T = absolute temperature, deg F + 460
Δt = temperature change, deg F
Δt_m = log-mean temperature difference, deg F
t_1 = inlet temperature of air inside tubes, deg F
t_2 = outlet temperature of air inside tubes, deg F
t_3 = inlet temperature of air outside tubes, deg F
t_4 = outlet temperature of air outside tubes, deg F
U = over-all heat-transfer coefficient, Btu/(hr)(sq ft) (deg F)
w_a = mass rate of air flow, lb per min
w_s = mass rate of steam flow, lb per min
μ = absolute viscosity, lb/(hr)(ft)

Introduction

Aircraft-engine supercharger intercoolers are designed to reduce the temperature of the air leaving the supercharger before it enters the carburetor. These intercoolers use cooling air obtained from the outside air and forced through the intercooler by the ram pressure created by the forward motion of the airplane. The first units manufactured by the authors' company were constructed with round aluminum tubes about $^1/_4$ in. diam with the supercharged air flowing through the tubes while the cooling air flowed across the tubes. This type is satisfactory, however, only for certain limited types of installations. In order to maintain a sufficient temperature difference between the average cooling air and the average supercharged air, it is necessary to have a fairly high rate of flow of cooling air. The actual amount of cooling air required for a given intercooler effectiveness of course depends also on the over-all heat-transfer coefficient. However, at high air flows, the pressure drop outside the tubes becomes excessive in a round-tube unit. Since the drag imposed on the airplane by the intercooler is a function of the product of the pressure drop of the cooling air by the amount of cooling air required and since the drag which is required for operating the intercooler increases substantially at high altitudes, it became necessary to develop an improved form of low-drag intercooler for modern high-altitude high-speed airplanes.

The first step in this development was to partially flatten the tubes so that their axes would be parallel to the cooling-air-flow direction. This permitted a greater flow of cooling air at the same pressure drop and reduced the cooling-air drag. Although the outside film coefficient was probably reduced in many cases, the inside coefficient, which with round tubes had been smaller than the outside coefficient, was raised by the decrease in the internal cross section of the tube. The net result was an increase in over-all heat-transfer coefficient for a given horsepower without reduction in surface per tube. Also, flattening the tubes allowed them to be placed closer together, thus forming a lighter-weight and more compact unit.

In an attempt to reduce still further the cooling-air drag horsepower, the flattened tubes were dimpled with small creases about $^1/_2$ in. apart running crosswise on the flattened portion of the tubes. At the same time, the outside dimension perpendicular to the flattened portion of the tube was increased to avoid increasing the pressure drop of the supercharged air inside the tubes too greatly by the introduction of the dimples.

The net result of this dimpling was greater than anticipated. In one typical case the cooling-air flow was reduced 30 per cent at an increase in pressure drop of only 25 per cent, resulting in a net saving of 12.5 per cent in horsepower required. Since the reduced cooling-air flow required resulted in a log-mean temperature difference only 87 per cent as great as before, it was apparent that this change had made a substantial increase in heat-transfer coefficient, most of which was estimated to be an increase in the coefficient for the air flowing inside the tube, since the air flow

[1] Heat Transfer Engineer, AiResearch Manufacturing Company.

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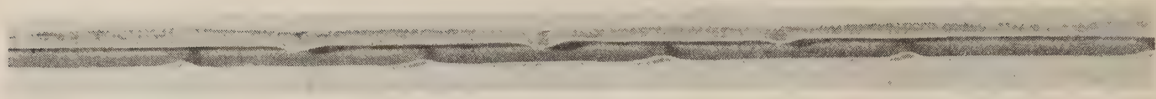

Fig. 1 Longitudinal Cross Section of Dimpled Tube

outside had been reduced. The tests to be described were performed in an attempt to obtain information which would correlate these results with the usual methods of computing heat-transfer coefficients.

Apparatus Used in Tests

The apparatus tested consisted of three tubular-type aircraft-engine intercoolers of approximately the same external dimensions. One unit was constructed with round tubes, one with partially flattened tubes, and one with partially flattened and dimpled tubes. The flattened and flattened dimpled tubes had the same flattening dimension and were placed with a no-flow clearance or cooling-air passage which was the same clearance as the diagonal clearance used in the round-tube unit. Major dimensions of the test units are given in Table 1. A longitudinal cross section of a portion of a dimpled tube is shown in Fig. 1.

TABLE 1 MAJOR DIMENSIONS OF TEST UNITS

	Round	Flattened	Dimpled
Tube length, net, in.	12	12	12
Tube outside diameter at headers, in.	0.247	0.247	0.247
Dimension perpendicular to flats, in.	....	0.1215	0.1215
Depth of dimple, in.	....	....	0.030
Dimple spacing, in.	....	....	0.5
Minimum space outside tubes, in.	0.0885	0.0885	0.0885
Hydraulic diameter inside, ft	0.0192	0.0136	0.0105
Hydraulic diameter outside, ft	0.0144	0.0144	0.0144
Number of active tubes	567	659	652
Surface area inside tubes, sq ft	34.3	39.8	39.4
Surface area outside tubes, sq ft	36.7	42.6	42.1

For the first tests the units were mounted as shown in Figs. 2 and 3, with air being blown through the tubes while steam condensed around the tubes. The steam was fed at low pressure and slightly superheated to provide complete dryness and allow determination of its enthalpy from measurements of temperature and pressure alone. A small portion of the steam was bled off the bell at the bottom of the unit in order to keep a steady flow of steam past the tubes and prevent the accumulation of water in the lower passages. The condensate ran to a pipe coil in which it was subcooled slightly and then passed to the weighing bucket. A sight glass was used to keep a constant water level in the lower part of the apparatus and thus prevent loss of steam. The inlet steam temperature and pressure were measured with a mercury-in-glass thermometer and a mercury manometer.

The air inlet temperature was measured with two thermometers in the inlet duct just ahead of the unit. The outlet temperature was measured with four thermometers in the outlet duct just beyond the unit. The outlet air temperature was also measured after the air had passed through a baffled mixing box. It was found that when the temperature at the outlet of the mixing box was corrected for the experimentally determined drop of temperature in the mixing box, this corrected temperature agreed very well with the average of the four outlet thermometers placed immediately after the unit.

Most of the air flows were measured with a flat-plate sharp-edged orifice in a duct following the mixing box. The largest air flows were computed from the pressure drop across the unit under test by extrapolation of the pressure drop versus flow curve drawn from the runs where the flow was determined by the orifice. The apparatus was thoroughly insulated with glass wool at all parts where needed.

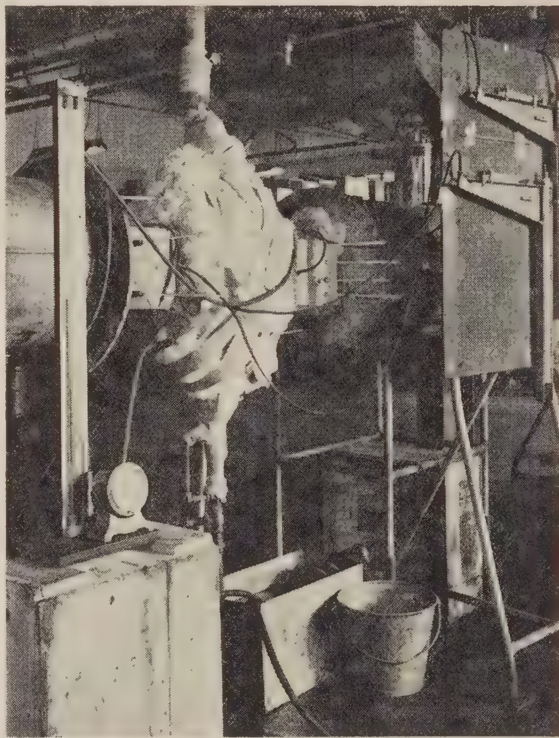

Fig. 2 Setup for Steam-Air Test

For the other series of tests the units were mounted as shown in Fig. 4. Hot air was used inside the tubes while room-temperature air was passed outside the tubes. Inlet air temperatures were measured with one thermometer on the cooling-air side and with three thermometers on the hot-air side. Outlet air temperatures were measured after the air had passed through the mixing boxes, and the temperatures were corrected for the known loss in the boxes. The air flows were measured with orifice plates for the lower air flows and were determined from unit pressure drops for the higher air flows. Because of the relatively high resistance of the unit as compared with that of the duct, a uniform flow distribution was obtained in both the cooling air and the hot paths.

Experimental Procedure

The first series of tests was run to determine the heat-transfer coefficient to air passing inside the tubes. Each of the three units was treated in the same manner. Air was passed through the tubes at a range of velocity to give values of **R** ranging from approximately 1400 to the highest available with the fan used as shown in Table 2. Steam was supplied to the unit under approximately 2 psig pressure and with approximately 10 deg F superheat. The air flow and steam temperature and pressure were kept constant within very close limits, and the system was allowed to balance thoroughly before readings were taken. Readings of all quantities except condensate flow rate were then

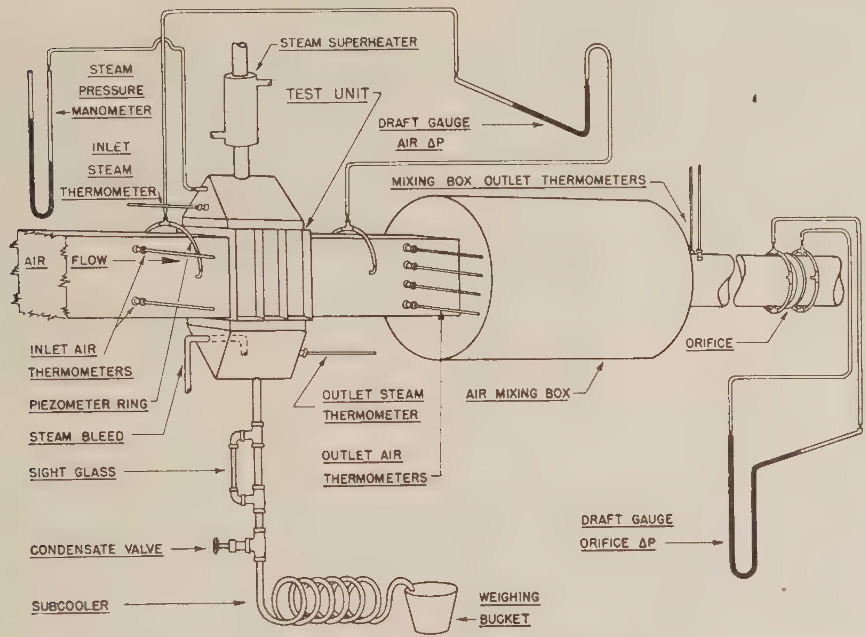

Fig. 3 Setup for Steam-Air Test

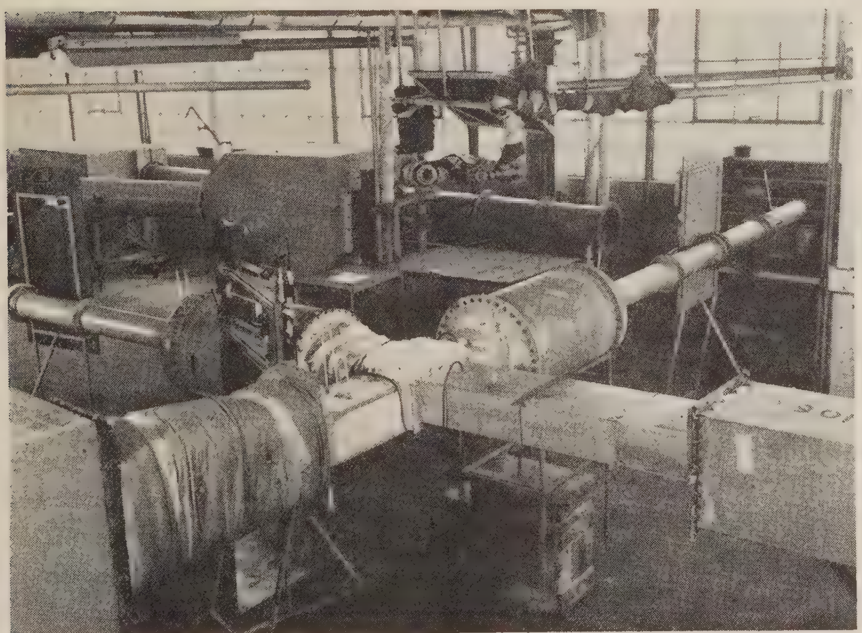

Fig. 4 Setup for Air-Air Test

taken every 2 min for a period of 10 min. At the end of this period,the condensate was weighed. Since the condensate level was kept constant in the sight glass during the run, it was possible to determine from the amount of adjustment necessary in the condensate valve whether the condensate rate was constant. The heat balances obtained on the higher air-flow points show some discrepancies due to the necessity for bleeding steam from the outlet header at these points in order to keep the condensate blown off the tubes and to obtain the highest heat transfer to the air inside the tubes. The extensive experience of the laboratory of the authors' company in measuring air heat gains and losses indicates that the errors involved in these measurements did not exceed 3 per cent. For this reason, all computations of heat-transfer coefficients have been based on the air-side heat gains.

TABLE 2 RANGE OF PRINCIPAL VARIABLES

	Round tubes Max	Round tubes Min	Flattened tubes Max	Flattened tubes Min	Dimpled tubes Max	Dimpled tubes Min
	Air-Steam Test—Inside Tubes					
w_a	117.5	9.65	90.0	9.5	76.5	9.6
R	17600	1450	11600	1230	9870	1239
t_1	111.4	70.3	113.3	79.5	119.2	71.0
t_2	173.8	150.7	193.6	182.3	209.1	196.9
Δt_m	93.2	71.7	72.5	58.0	52.1	41.0
h_i	39.1	4.61	38.1	6.09	55.1	9.62
G	42900	3520	40000	4230	44200	5650
q	2245	224	1635	264	2320	302
	Air-Air Test—Inside Tubes					
w_a	97.0	16.2	...	...	44.5	10.1
R	14550	2430	...	...	5740	1305
t_1	257.8	240.5	...	...	256.7	246.0
t_2	239.1	155.6	...	...	205.0	118.2
E_i	0.486	0.111	...	...	0.901	0.253
	Air-Air Test—Outside Tubes					
w_a	39.1	7.4	...	...	112.0	14.3
R	4650	880	...	...	9300	1185
t_3	109.1	83.3	...	...	103.0	86.0
t_4	243.0	125.9	...	...	202.4	124.6
E_o	0.918	0.206	...	...	0.705	0.083
Δt_m	93.7	44.8	...	...	75.5	51.3
h_i	33.0	5.1	...	...	35.1	10.1
h_o	62.0	15.5	...	...	74.0	10.9
U	23.3	4.1	...	...	21.4	5.8

To determine heat-transfer coefficients to air outside the tubes, the units were operated as intercoolers with hot air inside the tubes and cool air outside the tubes. The air flows were varied throughout the range of the apparatus as shown in Table 2. The flows were maintained until steady conditions were observed at all points and then all instruments were read as rapidly as possible by three observers. This procedure was repeated until there were two sets of readings at each point which agreed within an effectiveness of 0.01 with each other. At least 30 such points were taken on each unit.

Method of Calculation

In the steam test the enthalpy of the entering steam was determined from the readings of temperature and pressure by reference to the steam tables (1).[2] It was assumed that the tube-wall temperature would be the same as the condensing temperature of the steam. This assumption probably leads to much less error than would have been introduced if thermocouples had been used to determine tube-wall temperatures in the small thin-wall tubes. The enthalpy of the condensate was determined from the condensing temperature of the steam and the steam tables. By experiment it was determined that the heat loss through the insulation of the unit in the steam tests was approximately 7 Btu per min, therefore the net heat loss of the steam was determined as

$$q = w_s(i_1 - i_2 - 7) \qquad [1]$$

Air inlet and outlet temperatures were computed by taking the arithmetic average of the respective thermometer readings. Air flows were determined from pressure-drop and temperature readings at the orifice and were computed from graphs based on reference (2). The air-side heat gained was therefore

$$q = 0.244\, w_a \Delta t \qquad [2]$$

The over-all heat-transfer coefficient U was determined from the inside surface area of the tubes, from the log-mean temperature difference based on the condensing temperature of steam at the observed pressure and the inlet- and outlet-air temperatures and from the air-side heat gain as determined from Equation [2]. The over-all heat-transfer coefficient was therefore

$$U = \frac{60\, q}{A_i \times \Delta T_m} \qquad [3]$$

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Since the steam-side coefficient was probably of the order of 2000 Btu/(hr)(sq ft)(deg F) very slight error was introduced by assuming that the air-side coefficient was equal to the over-all coefficient. Also this error had the effect of giving a slightly lower inside coefficient which was desirable for design purposes. A plot of the air-side coefficient inside the tubes of the three units versus Reynolds number is shown in Fig. 5. A plot of the Nusselt number versus Reynolds number for the three units is shown in Fig. 6. In computing Reynolds number and Nusselt number the hydraulic diameters were based on the minimum cross section of the tube, that is, in the dimpled-tube unit the

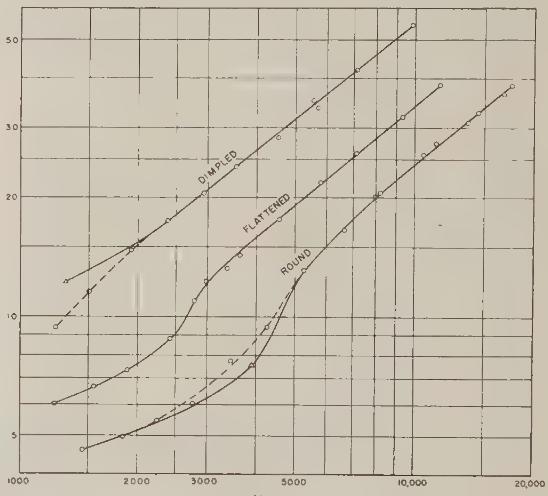

Fig. 5 Comparison of Experimental Heat-Transfer Coefficients for Air Inside Tubes

(Dashed lines show steam-air test results which were modified by air-air test results shown by triangles.)

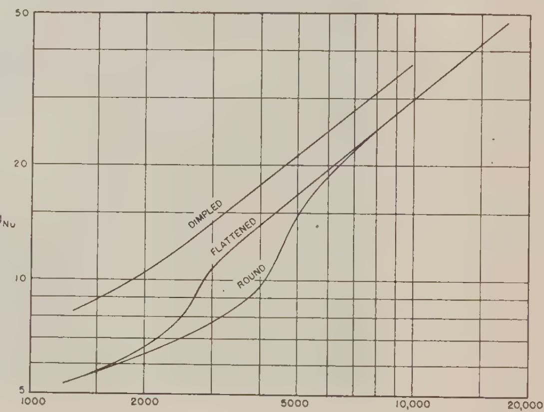

Fig. 6 Comparison of Experimental Values of Nusselt Number for Air Inside Tubes

internal cross section was determined under the dimple. The hydraulic diameter was then computed as

$$D_H = \frac{4A}{P} \qquad [4]$$

The mass velocity G was computed from the same minimum cross section and the conductivity and viscosity of the air were taken as 0.0155 Btu/(hr)(sq ft)(deg F/ft) and 0.047 lb/(hr)(sq ft), respectively, at an assumed average temperature of 140 F.

In the intercooler test the air temperatures and air flows were measured in the same manner as in the steam test. The air heat gains and losses were also determined in a similar manner in order to check the heat balance. The over-all heat-transfer coefficient in the intercooler test was determined from the log-mean temperature difference corrected for crossflow (4). The amount of heat transfer was assumed to be the average of the computed heat gained and heat lost. The surface area used was the average of the internal and external surfaces. The external heat-transfer coefficients were then determined from the following equation

$$\frac{1}{h_0 A_0} = \frac{2}{U(A_i + A_0)} - \frac{1}{h_i A_i} \qquad [5]$$

These outside coefficients are plotted in Figs. 7 and 8.

Correlation of Test Results

Fig. 6, which shows Nusselt number versus Reynolds number for flow inside the tubes, indicates that for the round tubes fully turbulent flow is not established until the Reynolds number

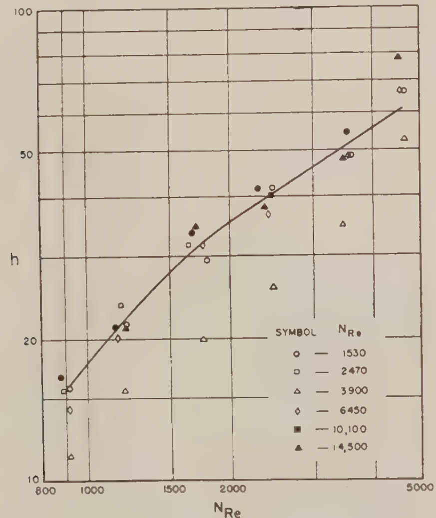

Fig. 7 Experimental Heat-Transfer Coefficients for Air Outside Round Tubes, Showing Results With Various Values of Reynolds Number Inside the Tubes

reaches 8000 and that laminar flow appears to continue up to a Reynolds number of about 2500. In contrast, it is noted that turbulent flow in the flattened tubes starts at approximately 3500 Reynolds number. The dimpled tubes appear to maintain fully turbulent flow from a Reynolds number of 2000.

The graph shows good correlation between the round tubes and flattened tubes tested in both the fully turbulent and fully laminar flow regions. This curve lies only slightly below the average curve given by McAdams (3). The position of this curve seems confirmed by other experiments with very small hydraulic diameters (5).

Both the dimpled-tube curve and the common curve for round and flattened tubes have approximately the same slope in the fully turbulent-flow regions, namely, 0.82.

It will be noted that the curve for dimpled tubes lies 26 per cent above the curve for flattened tubes. A suggested explanation of this is as follows:

Schack (6) states that the heat-transfer coefficient should vary with the length of the pipe, the exponent expressing this variation being —0.05. A recalculation of his figures shows that this exponent should have been stated as —0.07. Now, a dimpled tube might be considered as consisting of a series of short tubes with a new one starting at each dimple. Since, in the unit tested there are 21 dimples per tube, therefore 22 short tubes are represented. The Nusselt number for the dimpled-tube unit was found to be 26 per cent higher than for nondimpled tubes, thus

$$\left(\frac{1}{22}\right)^n = 1.26 \qquad [6]$$

where n is the exponent expressing the effect of tube length. Solving, n equals —0.07 which checks with the recalculation of Schack's result. Thus a logical justification may be established for use of the minimum cross-sectional area in determination of hydraulic diameter and mass velocity in dimpled tubes.

Fig. 5 shows the film coefficient inside the tubes for all three units. In drawing Fig. 6, the lines in Fig. 5 which pass through the points marked by triangles were used to determine the Nusselt number. The method of obtaining these points is explained as follows:

Fig. 7 shows the heat-transfer coefficients outside of round tubes. In this plot Reynolds number has been based on the hydraulic diameter of the diagonal passes between the tubes instead of on the tube diameter, in order to provide a more direct comparison with the results of the tests on the dimpled-tube unit in which the Reynolds number is also based on the passage available. It may be noted that the points plotted for a value of **R** of 3900 inside the tubes fall consistently below the average of all the other test points. It was therefore assumed that the steam-test points at values of **R** of 3450 and 4300 inside the tubes were in error since these were based on two runs, whereas the intercooler result is based on ten runs. By assuming outside coefficients at each of the cooling-air flows as read from the average line in Fig. 7, the inside coefficients were recalculated and averaged, and this average was plotted as the intercooler test point at a value of **R** of 3900 shown in Fig. 5. The same procedure was used in obtaining the intercooler test

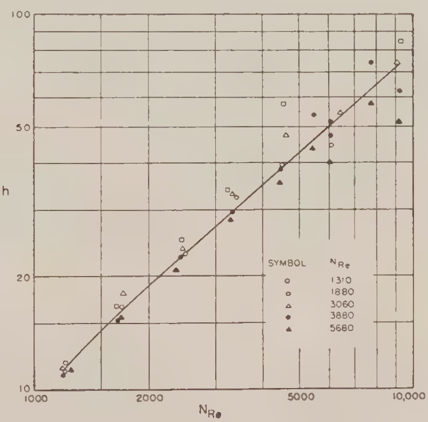

Fig. 8 Experimental Heat-Transfer Coefficients for Air Outside Dimpled Tubes, Showing Results With Various Values of Reynolds Number Inside the Tubes

point for the dimpled-tube unit shown at a value of **R** of 1300 in Fig. 5.

For heat-transfer coefficients outside round tubes, Fig. 7, extrapolation of Grimison's (7) values gives the following estimated equation for a no-flow spacing of 2.4 D_0 and a cooling-air spacing of 0.6 D_0 as used in the test unit

$$h = \frac{0.296\, k\, G_{max}^{0.61}}{\mu^{0.61}\, D_0^{0.39}} \qquad [7]$$

The test results in Fig. 7 average 27 per cent above the values given by Equation [7] in the region where **R** is greater than 2000.

Fig. 8 shows heat-transfer coefficients versus Reynolds number for air outside the dimpled-tube unit. In this case the arrangement of the dimpled tubes is such that an approximately straight passage of rectangular cross section is provided for air outside the tubes. The hydraulic diameter of this passage was determined from Equation [4]. It will be noted that the slope of the line in the turbulent-flow region is approximately 0.8 as compared to a slope of approximately 0.6 with the round-tube unit. This would seem to indicate that with this type of construction heat-transfer coefficients outside the tubes should be computed on the same basis as for flow inside the tubes.

There is seen to be a rather large spread of points in the higher values of **R** in Fig. 8. However, this spread does not indicate a very large experimental error. If heat-transfer coefficients outside the tubes are read from the curve in Fig. 8, and heat-transfer coefficients inside the tubes are read from Fig. 5, and the over-all coefficient U is computed for each of the test points, it is found that the computed U has an average variation of ± 5 per cent from the values of U obtained in the intercooler test. It is probable that these coefficients for air outside dimpled tubes are too high and it is hoped that further work can be done on this subject.

Fig. 9 shows heat-transfer coefficients inside the tubes plotted against the air pressure drop corrected to a standard density of 0.07651 pcf at the center of the unit. This air pressure drop is

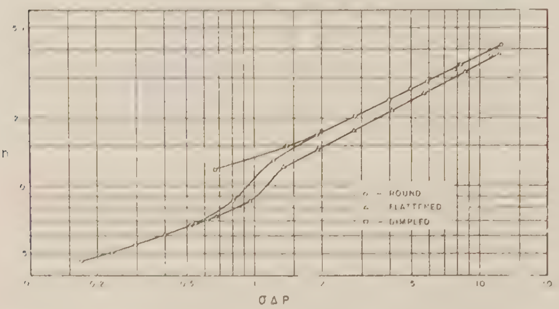

Fig. 9 Comparison of Experimental Heat-Transfer Coefficients and Pressure Drop for Air Inside Tubes

the over-all pressure drop as measured by static fittings located on the sides of the ducts 4 in. upstream and 4 in. downstream of the test unit, as shown in Fig. 3.

Conclusions

1 The transition from laminar to turbulent flow extends over the range **R** = 2000 to **R** = 8000 for small round tubes. If the tubes are flattened the transition region extends only from **R** = 2000 to **R** = 3500 while if the flattened tubes are dimpled the transition takes place at **R** = 2000.

2 For any given Reynolds number, flattening a round tube increases the heat-transfer coefficient substantially, while dim-

TABLE 3 ROUND TUBE AIR-STEAM TEST DATA

Run number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Steam																
Inlet temperature, (avg) deg F	227.5	229.1	228.1	229.3	229.2	230.7	232.5	231.5	228.8	229.1	230.4	224.0	224.0	231.5	217.8	229.8
Pressure, psia	16.6	15.0	16.5	15.0	16.6	16.5	16.5	16.5	16.7	16.7	16.7	16.7	16.5	16.7	16.4	16.7
Condensing temperature, deg F	218.2	213.0	218.0	213.0	218.1	218.0	218.0	218.0	218.4	218.4	218.4	218.4	218.0	218.4	217.5	218.4
Enthalpy, Btu per lb	1161	1161	1161	1161	1161	1161	1162	1162	1161	1161	1161	1160	1160	1162	1158	1161
Enthalpy condensate, Btu per lb	186	181	186	180	186	186	186	186	186	186	186	186	185	186	185	186
Enthalpy change, Btu per lb	975	980	975	981	975	975	976	976	975	975	975	974	975	976	973	975
Condensate flow, lb per min	0.238	0.263	0.297	0.323	0.407	0.510	0.641	0.753	1.053	0.975	1.081	1.520	1.566	1.84	1.81	2.20
Heat loss of steam, Btu per min	232	258	289	317	397	497	626	735	1028	950	1053	1480	1525	1795	1757	2142
Insulation loss, Btu per min	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7
Net heat loss of steam, Btu per min	225	251	282	310	390	490	619	728	1021	943	1046	1473	1518	1788	1750	2135
Air																
Outlet temperature, (avg) deg F	172.0	160.2	159.0	153.5	161.3	157.4	165.9	169.6	159.1	168.7	173.0	156.0	162.3	153.2	164.6	152.0
Inlet temperature, (avg) deg F	76.8	76.2	78.7	84.7	91.4	85.7	92.7	102.0	76.2	97.6	111.4	72.7	94.2	74.0	103.9	73.7
Temperature rise, deg F	95.2	84.0	80.3	68.8	69.9	71.7	73.2	67.6	82.9	71.1	61.6	83.3	68.1	79.2	60.7	78.3
Air flow, lb per min	9.65	12.2	14.9	18.4	23.0	28.5	35.1	44.5	53.9	54.7	70.3	75.5	91.0	96.5	112	117.5
Air heat gain, Btu per min	224	250	290	309	386	491	616	734	1090	959	1070	1533	1513	1865	1660	2245
Discrepancy, per cent	0.4	0.4	3.4	0.3	0.5	1.6	1.3	0.8	6.3	0.7	0.9	3.9	0.3	4.1	5.4	4.9
Air pressure drop, $\sigma \Delta P$, in. H_2O	0.17	0.23	0.30	0.40	0.55	0.81	1.22	1.91	...	2.79	...	4.94	...	7.74	...	...
Log-mean temperature difference, deg F	85.0	88.2	93.2	89.3	87.3	91.8	83.4	77.4	94.7	80.2	71.7	98.2	85.4	99.5	79.5	100.6
Over-all coefficient, Btu/(hr)(sq ft)(deg F)	4.61	4.96	5.48	6.06	7.73	9.49	13.2	16.6	20.1	20.7	25.7	27.4	31.0	32.7	36.6	39.1
Nusselt number, dimensionless	5.71	6.15	6.80	7.51	9.58	11.8	16.3	20.6	24.9	25.7	31.8	34.0	38.4	40.5	45.4	48.5
Reynolds number, dimensionless	1450	1830	2240	2760	3450	4270	5260	6670	8100	8200	10500	11300	13650	14500	16800	17600

TABLE 4 FLATTENED TUBE AIR-STEAM TEST DATA

Run number	1	2	3	4	5	6	7	8	9	10	11	12	13
STEAM													
Inlet temperature, (avg) deg F	228.7	228.7	228.3	229.2	228.1	229.1	228.4	228.7	227.7	228.0	226.9	226.9	226.4
Pressure, psia	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.6
Condensing temperature, deg F	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.1
Enthalpy, Btu per lb	1161	1161	1161	1161	1161	1161	1161	1161	1161	1161	1160	1160	1160
Enthalpy condensate, Btu per lb	186	186	186	186	186	186	186	186	186	186	186	186	186
Enthalpy change, Btu per lb	975	975	975	975	975	975	975	975	975	975	974	974	974
Condensate flow, lb per min	0.285	0.328	0.369	0.438	0.495	0.516	0.562	0.678	0.788	0.873	1.068	1.438	1.581
Heat loss of steam, Btu per min	278	320	360	427	482	503	548	660	768	851	1029	1400	1540
Insulation loss, Btu per min	7	7	7	7	7	7	7	7	7	7	7	7	7
Net heat loss of steam, Btu per min	271	313	353	420	475	496	541	653	761	844	1022	1393	1533
AIR													
Outlet temperature, (avg) deg F	193.6	188.2	184.2	182.3	187.6	190.7	191.7	186.8	187.6	190.3	188.0	182.5	184.1
Inlet temperature, (avg) deg F	79.5	81.3	84.9	89.8	95.6	101.5	106.5	92.0	100.0	113.3	109.5	99.7	109.7
Temperature rise, deg F	114.1	106.9	99.3	92.5	92.0	89.2	85.2	94.8	87.6	77.0	78.5	82.8	74.4
Air flow, lb per min	9.5	12.0	14.6	18.7	21.5	23.1	26.2	28.1	35.5	45.0	55.2	72.4	90.0
Air heat gain, Btu per min	264	313	353	422	482	503	524	650	760	845	1057	1462	1635
Discrepancy, per cent	2.6	0.0	0.0	0.5	1.5	1.4	3.2	0.5	0.1	0.1	3.3	4.7	6.2
Air pressure drop, $\sigma \Delta P$, in. H_2O	0.40	0.52	0.68	0.96	1.20	1.36	1.67	1.92	2.78	4.10	5.64	8.70	12.3
Log-mean temperature difference, deg F	65.9	70.3	72.5	72.3	66.1	61.6	59.1	68.0	64.5	58.0	61.2	68.8	64.1
Over-all coefficient, Btu/hr(sq ft)(deg F)	6.03	6.70	7.33	8.79	11.0	12.3	13.3	14.4	17.7	21.9	25.9	32.0	38.3
Nusselt number, dimensionless	5.29	5.88	6.43	7.71	9.63	10.8	11.7	12.6	15.6	19.2	22.7	28.1	33.6
Reynolds number, dimensionless	1230	1550	1880	2420	2780	2980	3380	3620	4580	5800	7130	9340	11600

TABLE 5 DIMPLED TUBE AIR-STEAM TEST DATA

Run number	1	2	3	4	5	6	7	8	9	10	11	12	13
STEAM													
Inlet temperature, (avg) deg F	226.5	226.8	226.4	227.2	226.9	226.3	226.0	229.9	225.7	229.2	229.3	228.8	227.2
Pressure, psia	16.6	16.6	16.6	16.6	16.6	16.6	16.6	16.8	16.6	16.7	16.7	16.7	16.6
Condensing temperature, deg F	218.1	218.1	218.1	218.1	218.1	218.1	218.1	218.8	218.1	218.6	218.6	218.6	218.1
Enthalpy, Btu per lb	1160	1160	1160	1161	1160	1160	1160	1161	1160	1161	1161	1161	1161
Enthalpy condensate, Btu per lb	187	186	186	186	186	186	186	187	186	187	186	187	186
Enthalpy change, Btu per lb	973	974	974	975	974	974	974	974	974	974	975	974	975
Condensate flow, lb per min	0.317	0.380	0.473	0.520	0.588	0.747	0.836	1.36	0.964	1.69	1.65	2.20	2.20
Heat loss of steam, Btu per min	309	370	460	507	572	728	813	1323	938	1645	1610	2142	2144
Insulation loss, Btu per min	7	7	7	7	7	7	7	7	7	7	7	7	7
Net heat loss of steam, Btu per min	302	363	453	500	565	720	806	1316	931	1638	1603	2135	2137
AIR													
Outlet temperature, (avg) deg F	208.6	208.9	209.1	208.8	208.6	207.1	206.5	202.5	205.7	199.9	200.3	196.9	197.4
Inlet temperature, (avg) deg F	79.7	82.5	87.7	96.3	108.0	99.8	113.0	71.0	119.1	71.4	74.3	73.1	73.2
Temperature rise, deg F	128.9	126.4	121.4	112.5	100.6	107.3	93.5	131.5	86.6	128.5	126.0	123.8	124.2
Air flow, lb per min	9.6	11.7	14.9	18.5	22.8	27.3	35.1	43.1	44.2	55.2	55.2	76.2	76.5
Air heat gain, Btu per min	302	361	442	508	559	716	800	1382	935	1730	1697	2297	2320
Discrepancy, per cent	0.0	0.8	2.5	1.6	1.1	0.7	0.8	4.8	0.4	5.3	5.5	7.0	8.0
Air pressure drop, $\sigma \Delta P$, in. H_2O	0.67	0.95	1.40	1.99	2.82	3.97	5.89	8.25	8.62	12.5	12.5	21.4	21.3
Log-mean temperature difference, deg F	48.2	47.0	45.4	43.7	41.0	45.1	42.4	59.6	41.7	62.2	61.0	64.7	63.9
Over-all coefficient, Btu/hr(sq ft)(deg F)	9.41	11.6	14.7	17.5	20.5	23.9	28.4	35.3	33.7	42.3	42.4	54.0	55.3
Nusselt number, dimensionless	6.38	7.83	9.96	11.9	13.9	16.2	19.3	23.9	22.8	28.5	28.5	36.8	37.4
Reynolds number, dimensionless	1240	1510	1920	2380	2940	3520	4530	5560	5700	7120	7120	9830	9870

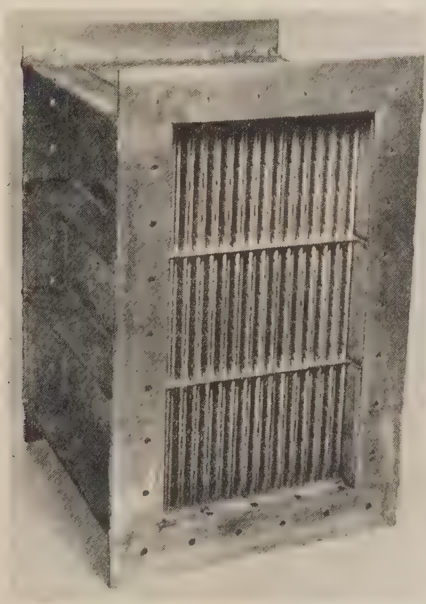

FIG. 10 TEST UNIT WITH DIMPLED TUBES

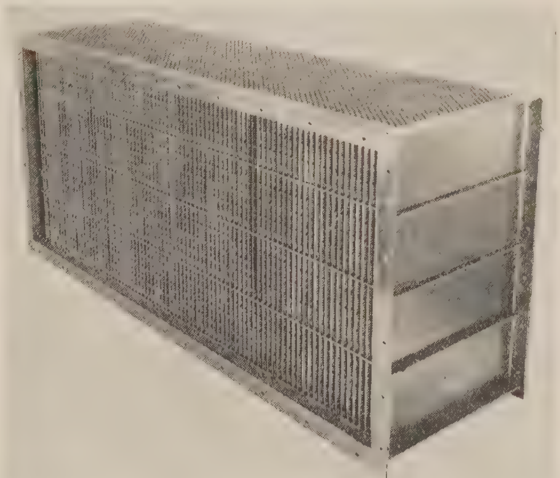

FIG. 11 LARGE AIRCRAFT INTERCOOLER WITH DIMPLED TUBES

pling the flattened tube increases the heat-transfer coefficient to an even higher value. This statement also applies to equal weight flows of air, since the Reynolds number for a tube of given perimeter at a given weight flow is not varied by flattening the tube or by dimpling it.

3 For the same air-pressure drop, dimpled tubes have the same or higher heat-transfer coefficient than round tubes, while plain flattened tubes have a lower heat-transfer coefficient, except at very low air-pressure drops.

4 The use of dimpled tubes in intercoolers permits a reduction in the amount of cooling air required and therefore leads to a lower drag on the airplane.

Acknowledgments

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(*Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946*)

Heat Transfer From a Cylindrical Surface to Air in Parallel Flow With and Without Unheated Starting Sections[1]

By MAX JAKOB[2] and W. M. DOW,[3] CHICAGO, ILL.

The present investigation sponsored by the Illinois Institute of Technology, Armour Research Foundation, and Institute of Gas Technology, deals with the heat transfer of air in parallel flow to a surface, a process of great practical importance which, for instance, occurs with all kinds of fins or on the skin of an airplane in flight. Previous knowledge of that process was based on a few sets of experiments which were performed with plane surfaces and led to a considerably higher heat transfer than a reliable theory, due to H. Latzko. In particular, the influence of nonheated starting sections seemed to require a new investigation. Compared with the use of plane plates by previous investigators, the use of an electrically heated cylindrical specimen has the following advantages: A cylinder can be easily placed in the center of an air jet and is free of the edge losses of a plate; for both reasons, air jets of moderate diameter can be used. Uniform heating is easier to provide, heat losses to the back are easier to control, and noses of different shape and cylindrical starting sections can readily be used for studying the behavior of the so-called boundary layer of the fluid which is developing along the surface, first streamlined and then turbulent, and in which all resistance against heat transfer is concentrated. The experiments were performed with specimens of 1.3 in. diam and 9 to 20 in. total length, the ratio of the heated length to the total length being varied from 40 to 90 per cent. Spherical, ellipsoidal, and conical nosepieces were used. The air velocity was varied from 10 to 150 fps and it is assumed that the results can be theoretically extrapolated above sound velocity, using the principle of similarity.

Nomenclature

The following nomenclature is used in the paper:

A = area of heating surface, sq ft
b = thickness of boundary layer, ft
C, C_0, C_1, C_2, C_3 are constant factors
c_p = specific heat at constant pressure of air, B $\mathrm{lb_m}^{-1}$ F^{-1}
$F = N/M$ = correction factor
H = velocity head in feet of air, ft
H_w = velocity head, in. of water
h = coefficient of heat transfer by convection (mean taken over heating length, L_{th},) B hr^{-1} ft^{-2} F^{-1}
h_x = local coefficient of heat transfer by convection at distance x from leading edge, B hr^{-1} ft^{-2} F^{-1}
$h_{x,m}$ = mean of h_x, taken from $x = x_1$ to $x = x$, B hr^{-1} ft^{-2} F^{-1}
J = mechanical equivalent of heat, ft lb/B
k = thermal conductivity of air, B hr^{-1} ft^{-1} F^{-1}
L_{st} = hydrodynamic starting length, ft
L_{th} = thermal or heating length, ft
L_{tot} = total length, ft
M, N are integrals, defined in text, sq ft/sec
m and n are constant exponents
$N_{Nu} = \dfrac{hL_{tot}}{k}$ = Nusselt number (total mean)
$(N_{Nu})_{max}$ = Nusselt number, corresponding to q_{max}
$(N_{Nu})_{min}$ = Nusselt number, corresponding to q_{min}
$(N_{Nu})_0$ = Nusselt number for $L_{st}/L_{tot} = 0$
$(N_{Nu})_x = \dfrac{h_x x}{k}$ = local Nusselt number
$(N_{Nu})_{x,m} = \dfrac{h_{x,m} x}{k}$ = mean Nusselt number for distance x from leading edge
$N_{Nu}' = \dfrac{h(L_{tot} - x_0)}{k}$
$N_{Pr} = \dfrac{\rho c_p \nu}{k}$ = Prandtl number
$N_{Re} = \dfrac{v_a L_{tot}}{\nu}$ = Reynolds number with L_{tot} as characteristic length
$(N_{Re})_{cr}$ = critical Reynolds number
$(N_{Re})_x = \dfrac{v_a x}{\nu}$ = Reynolds number with x as characteristic length
$N_{Re}' = \dfrac{v_a(L_{tot} - x_0)}{\nu}$
$(N_{Re}')_x' = \dfrac{v_a x'}{\nu}$
q = rate of heat flow, B/hr
q' = rate of heat transferred by convection over length $x - x_1$ and unit width, B/hr
q_c = rate of heat transferred by convection from cylindrical surface, B/hr
q_i = rate of heat input, B/hr
q_l = sum of heat losses, B/hr
q_{max} = rate of heat transferred for certain values of L_{tot} and L_{st}, B/hr
q_{min} = rate of heat transferred for the same L_{tot}, but L_{st} = 0, B/hr

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[1] Based partly on a thesis submitted by W. M. Dow to the Graduate School, Illinois Institute of Technology, in partial fulfillment of the requirement for the degree of Master of Science, for studies completed in the Institute of Gas Technology.

[2] Research Professor of Mechanical Engineering, Illinois Institute of Technology; Consultant in Heat Research, Armour Research Foundation, Chicago, Ill. Mem. A.S.M.E.

[3] Fellow of the Institute of Gas Technology, Chicago, Ill., now Ensign, U. S. Navy.

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q_p = rate of heat transferred by convection from plane surface, B/hr
r = radius of cylinder, ft
t = air temperature at distance y from surface, °F
t_a = temperature of bulk of air stream, °F
t_f = $(t_a + t_s)/2$ = film temperature, °F
t_i = $t_a + \theta_i$ = impressed temperature of unheated surface, °F
t_s = average surface temperature, °F
v = air velocity at distance y from surface, ft/sec
v_a = velocity of bulk of fluid (air), ft/sec
v_b = air velocity at distance b from surface, ft/sec
W = width of flat surface, ft
x = distance from leading edge, general, ft
x_0 = distance from leading edge, defined by Fig. 1, ft
x_1 = L_{st} (see this symbol), ft
x_{cr} = distance from leading edge where critical Reynolds number occurs, ft
x' = $x - x_0$, ft
y = perpendicular distance from surface, ft
θ = $t - t_a$ = temperature excess at distance y from surface, F
θ_{ad} = temperature increment by frictionless adiabatic stopping of flow, F
θ_i = temperature increment by adiabatic stopping of flow, including effect of friction, F
θ_s = $t_s - t_a$ = temperature difference between heating surface and bulk of air stream, F
ν = kinematic viscosity of fluid (air) at bulk temperature t_a, sq ft/sec
ν_f = kinematic viscosity of air at film temperature t_f, sq ft/hr
ρ = density of air, lb_m/cu ft
$\varphi(N_{Pr})$ = function of N_{Pr}

Introduction

While the heat transfer between a cylindrical surface and a fluid has been abundantly studied for the flow in a tube and across a cylinder, no experiments seem to have been done for the parallel flow on the surface of a cylinder on the outside of a tube, and only a few sets of experiments have been performed concerning the free stream parallel to a plane heating surface. As a consequence, the numerous correlations of heat transfer for flow inside of tubes and channels and across cylinders have a much better foundation than the few correlations known for the flow parallel to a heating or cooling surface when the fluid is not bounded by the walls of a tube or channel.

The present experimental investigation was undertaken to secure and correlate data of heat transfer for this sort of flow, including transition-state conditions and the influence of hydrodynamic starting sections.

Referring to Fig. 1, it may be remembered that at the leading edge the fluid is suddenly caused to adhere to the solid surface. As a result a thin layer is formed in which the viscosity forces of the fluid bring about a change of velocity. As the stream moves over the surface, the retarding effect of the viscosity forces gradually penetrates deeper and deeper into the surrounding fluid so that the boundary layer increases in thickness. For some distance the flow in that layer is laminar; in a transition range it gradually becomes turbulent; in the turbulent stage, finally, the thickness of the layer increases more rapidly than it did in the laminar range. Within the boundary layer, however, still a very thin laminar film exists next the surface, being a permanent feature of any fully established flow.

The boundary layer of a free stream parallel to a surface can be characterized by a dimensionless Reynolds number xv_a/ν,

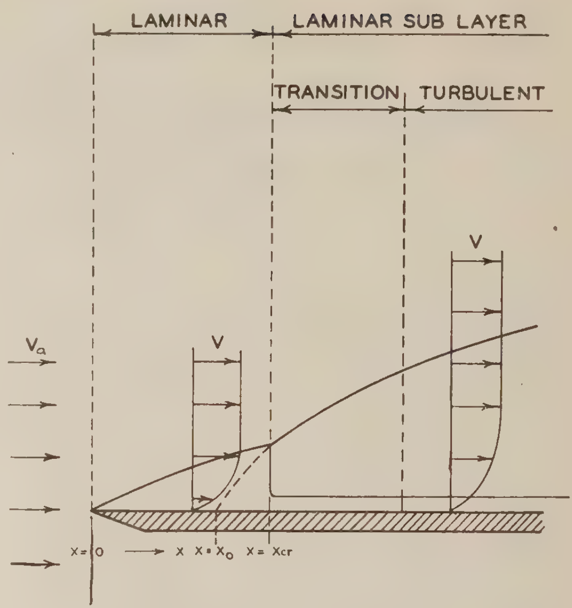

Fig. 1 Boundary-Layer Development Along a Flat Plate

where x is the distance from the leading edge ($x = 0$ in Fig. 1), v_a is the unaltered velocity, and ν is the kinematic viscosity of the fluid. The transition from laminar to turbulent flow is not sharp; it depends on original disturbances in the turbulent stream ahead of the surface, the shape of the leading edge, and the roughness of the surface. Critical Reynolds numbers, determining the place, $x = x_{cr}$, where laminar flow ceases in the layer, have been measured from 100,000 to 500,000, depending on the starting conditions, 300,000 being the ordinary value according to van der Hegge Zijnen (1)[4] and Hansen (2). With blunt leading edges, however, turbulent boundary layers have been obtained straight from the start (3),[5] $x = 0$.

Due to the well-known analogy between momentum transfer and heat transfer, the mode of heat transmission depends on whether the flow in the boundary layer is laminar or turbulent. A difference in the coefficient of heat transfer by convection can also be expected if heating does not begin at the leading edge, but is preceded by a hydrodynamic starting length. Whereas the development of the boundary layer and its effect on heat transfer will not necessarily be the same for cylindrical and plane surfaces, the difference will be considerable only if the radius of the surface curvature is relatively small (4).[6]

Considering the described features of flow and heat transfer along a surface, the objectives of the investigation are as follows:

1 To determine the heat transfer between a cylindrical surface and air in parallel flow, laminar or turbulent, under conditions where only a small influence of the surface curvature is to be expected.

2 To determine the effect of different hydrodynamic starting lengths on the heat-transfer coefficient.

3 To compare the results with theoretical and experimental results for plane surfaces reported in the literature.

Compared with the use of plane plates by previous investiga-

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.
[5] Loc. cit., p. 139.
[6] Loc. cit., p. 249.

tors the use of a cylindrical specimen has the following advantages:

1 A cylinder can be easily placed in the center of an air jet without disturbing the uniform velocity field ahead of the apparatus.

2 With a plane plate, heat losses to the side have to be considered, and for this reason the plate must be rather wide. Such losses do not exist in a cylinder and therefore its diameter can be relatively small so that an air jet of moderate diameter can be used.

3 Uniform heating is easier to provide and losses to the backside of the heater are smaller and easier to control than with plane plates.

4 A cylindrical specimen is particularly convenient for studying the influence of the hydrodynamic starting length.

With these objectives and advantages in mind a cylindrical specimen was designed with an electric heating coil inside and thermocouples just below the outer surface to insure accurate temperature measurements without interference with the air stream. Wooden nosepieces of different shape and length served to close the upstream end of the specimen and to vary the hydrodynamic starting conditions.

In the experiments air was pressed through a channel by a blower and discharged by a nozzle into the room and against the nose of the specimen which was placed in the air stream with its axis parallel to the flow lines. Under these conditions the main air-stream velocity was measured with a Pitot tube; the heat input to the specimen was determined from the wattage of the heating coil by electric instruments; and the temperatures of the main air stream and the specimen surface were measured by thermocouples. These are the data necessary for calculating the coefficient of heat transfer by convection. Experimental runs were made at various velocities and under different hydrodynamic starting conditions. The results were correlated in graphs and formulas, discussed and compared with those of previous investigators.

Description of Apparatus

For the time being the experiments had to be restricted to one heating cylinder. In order that the results might be applicable to the case of a plane surface with reasonable approximation, a diameter of 1.3 in. seemed to be appropriate, considering the range of Reynolds numbers which could be covered by the experimental arrangement.

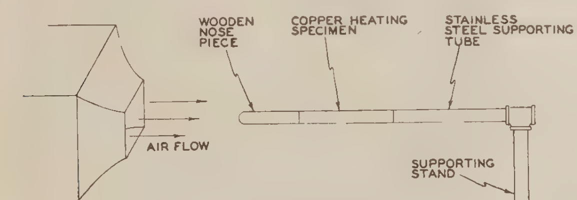

Fig. 2 General Arrangement of Apparatus

Fig. 2 is a self-explanatory sketch of the general setup of the apparatus. The nozzle indicated in this sketch had a square throat, 10 in. × 10 in. Details of the specimen with supporting assembly are given in Figs. 3 and 4. The main parts are a copper tube 8 in. long, 1.300 in. OD, 0.950 in. ID, and a stainless-steel tube 0.840 in. OD, 0.622 in. ID, on which nichrome wire, B and S gage No. 24 B (0.0201 in. diam) was wound. This heating coil was insulated by glass cloth and placed inside the copper tube.

Extreme care was taken to insure a smooth and continuous surface junction between the wooden nosepiece and the copper tube, as irregularities at this point would greatly influence the character of the boundary layer. The surface junctions were made with paper rings inserted in grooves as shown in Figs. 3 and 4. Melted paraffin was then painted on the junction and carefully scraped flush with the surface of the copper tube and the wooden nosepiece. A Pitot tube constructed according to Prandtl's design (5),[7] in connection with an inclined manometer, was provided for the velocity-head measurements.

All thermocouples used in the investigation were made of copper and constantan, B & S gage No. 28 (0.0126 in. diam). The cold junctions were placed in melting ice at 32 deg F. By means of a mercury-contact selector switch, each thermocouple could be connected to a Leeds and Northrup portable precision potentiomeeter which allowed readings of 0.001 mv.

One thermocouple served to the measurement of the air-flow temperature. It was inserted into a $^1/_8$-in-diam copper tube which was placed parallel to the air flow for a distance of 2 in. and was sealed at the upstream end.

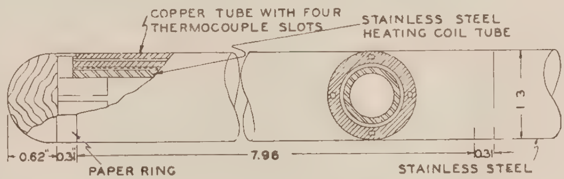

Fig. 3 Details of Copper Heating Specimen

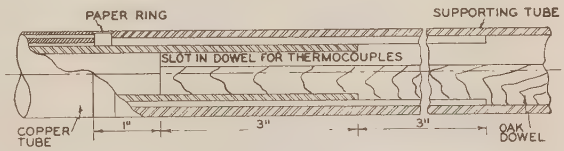

Fig. 4 Details of Supporting Assembly

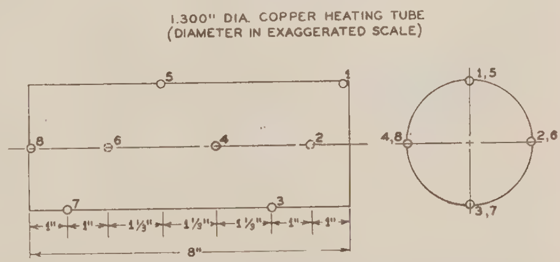

Fig. 5 Locations of Surface Thermocouples in Copper Heating Specimen

In order to avoid disturbances on the surface, the eight thermocouples serving for the measurement of the surface temperature were placed in, and brought out axially through four slots of square cross section (0.080 in. side length) in the wall of the heating tube as shown in Fig. 3. The thermocouples fitted snugly in the axial slots but were removable for ease of calibration and manipulation. Locations of the hot joints of the thermocouples are shown in Fig. 5.

Thermocouples for secondary temperature measurements were placed in the wooden nosepieces, Fig. 6, and on the rear part of the heating-coil tube, Fig. 7. Each of the latter thermocouples was wrapped around the tube once so that the wires were in an

[7] Loc. cit., p. 170.

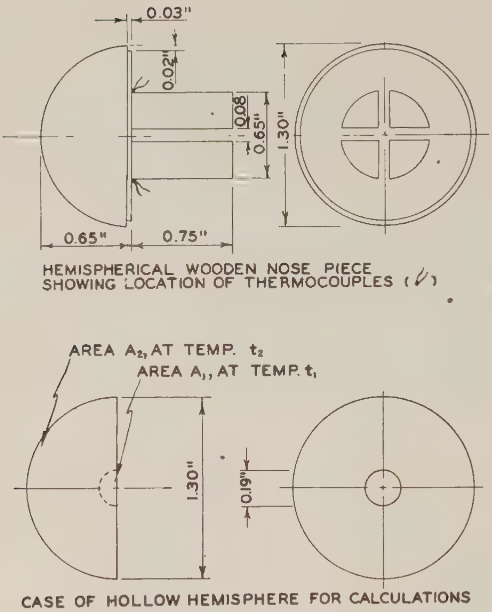

Fig. 6 Hemispherical Wooden Nose Piece

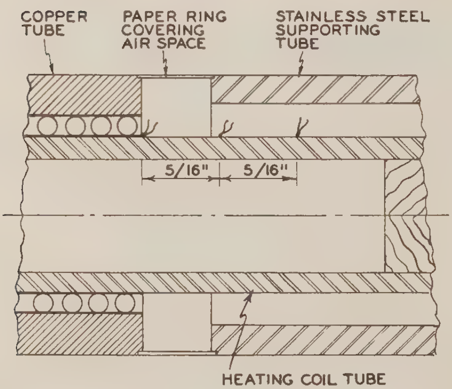

Fig. 7 Location of Thermocouple on Heating Coil Tube

isothermal zone for about 1 in. from the joint. The purpose of these thermocouples will be explained in the following section.

A hemispherical nosepiece of 0.65 in. diam is shown in Fig. 3. The six different starting pieces used in the experiments are described in Table 1.

TABLE 1 STARTING PIECES

Specification	L_{st} ft	L_{tot} ft	L_{st}/L_{tot}
Cylinder with hemispherical nose............	1.026	1.693	0.606
Cylinder with hemispherical nose............	0.689	1.356	0.508
Cylinder with hemispherical nose............	0.354	1.021	0.347
Conical piece (4 in. long)...................	0.187	0.854	0.220
Ellipsoidal nose............................	0.092	0.759	0.122
Hemispherical nose..........................	0.075	0.742	0.101

In this table L_{st} is the hydrodynamic starting length of the specimen, defined as the ratio of the surface area of the starting piece to the perimeter of the heating cylinder. The total length, L_{tot}, is defined as the sum of the starting length L_{st}, and the thermal or heating length L_{th}, of 8 in.[8]

Test Procedures

The velocities were measured by the Pitot tube, using the equation

$$v_a = \sqrt{2gH} = 18.275 \sqrt{H_w/\rho} \qquad [1]$$

where

v_a = velocity, ft/sec
H = velocity head as used in test, ft of air
H_w = velocity head, in. of water
ρ = density of air, lb_m/cu ft

H_w was read directly on the inclined manometer.

Preliminary velocity traverses through the air jet, conducted in horizontal and vertical direction, revealed that for a distance of 2 ft downstream from the nozzle, the air velocity varied less than 1 per cent within an 8-in. core. Therefore, in the main experiments the velocity of the air stream was determined at one point only, midway between the ends of the heating coil and 2 in. from the surface of the specimen.

The electromotive force of the thermocouples was checked at the water boiling point. The indications of the couples, when placed in saturated steam, were compared with the values of Keenan and Keyes's steam tables (6). The deviations from the manufacturer's calibration table were less than 0.1 F at 212°F.[9]

As the velocity of the air, also its temperature, were always taken at a point midway between the ends of, and 2 in. from the heating surface. No correction was applied for the influence of velocity on temperature measurement, which may have reached 0.7 F at the highest velocities, corresponding to about 1 per cent in the measured coefficient of heat transfer.

The measurement of the temperature of the exposed surface is a delicate operation in any heat-transfer experiment. The thermocouple wires in the axial slots below the surface parted from the junctions virtually along isothermal lines, the maximum variation of the surface temperature in axial direction being generally below and only exceptionally slightly above 1 F/in. Therefore the junctions were not sensibly affected by conduction in the axial direction. On the other hand, the influence of the slots in the copper wall on the cross-sectional temperature field had to be studied. Obviously, the length of the heat-flow path near the slots was increased and the distribution of the surface temperature around the tube was not uniform any more. In order to find the surface temperatures from the indications of the thermocouples in the slots, the temperature field in a cross section of the copper tube was determined by Lehmann's graphical method (7).[10] Fig. 8 shows the temperature field in the tube wall for the most drastic conditions encountered in the experimental work. The maximum possible deviation of the thermocouple indication from the undisturbed surface temperature was less than 0.1 F and consequently the thermocouple readings were taken as the surface temperatures.

The heat input was determined by electric-resistance and current measurements. Direct current was used in the heating coil. Its resistance was measured by a Wheatstone bridge, the slight increase of the resistance of the nichrome wire with temperature being taken from Knowlton's handbook (8). A millivoltmeter

[8] The slight difference between the sixth ratio given in Table 1, and L_{st}/L_{tot}, as calculated from the lengths given in Fig. 3, is due to some details in design not indicated in the figure.

[9] In order to distinguish temperature differences from temperatures the unit symbols F and °F, respectively, are employed in this paper. A similar procedure is recommended when the centigrade temperature scale or an absolute temperature scale is used.

[10] Loc. cit., p. 187.

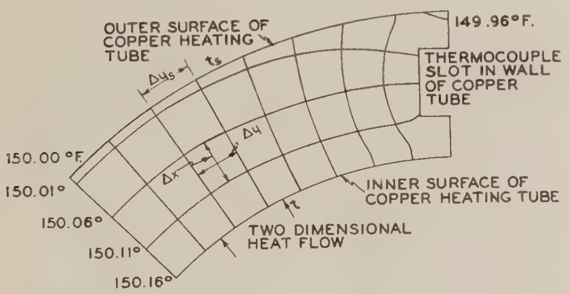

FIG. 8 TEMPERATURE FIELD IN COPPER TUBE WALL

with shunt served to determine the heating current. Voltmeter readings were also taken for each run as a check.

Any portions of the heat input which are not given up from the heating surface to the flowing air by convection will be called heat losses. They might be due to the following:

(*a*) Radiation from the heating surface of the specimen to the room.

(*b*) Conduction and convection from the paper rings at both ends of the specimen.

(*c*) Conduction through the wooden nosepiece.

(*d*) Conduction through the downstream end of the heating-coil tube.

(*e*) Conduction and convection through the air space at the downstream end of the specimen.

Loss (*a*) was calculated by Kirchhoff's and Stefan-Boltzmann's laws. Loss (*b*) was found, considering the paper rings as straight sheets protruding from a copper base at known temperature and exposed to the same air flow and at the same coefficient of heat transfer as the heating surface of the specimen. The calculation of the heat loss (*c*) was reduced to the problem of radial heat conduction in a hollow hemisphere with the inner surface area equal to the cross-sectional areas of the wooden prongs holding the nosepiece in the heating-coil tube. The inner temperature was measured for several representative runs by means of the thermocouples indicated in Fig. 6.

The heat loss (*d*) down the heating-coil tube was calculated from the temperature gradient, measured by the three thermocouples, shown in Fig. 7, the cross-sectional area of that tube and the thermal conductivity of the stainless steel. From the decrease of the temperature gradient the loss (*e*) was found.

The four losses, (*a*), (*b*), (*c*), and (*e*) together were less than 1.5 per cent of the total heat input. They were calculated for several representative runs throughout the entire velocity range of the experiments, and an average correction of 1.4 per cent was applied to each run.

The heat loss (*d*) was the only one of some weight. It amounted from 1.4 to 5.3 per cent of the heat input and had to be determined individually for each run.

Denoting the area of the heating surface by A, the rate of heat input by q_i, and the sum of the heat losses by q_j, the average surface temperature by t_s, and the temperature of the bulk of the air stream by t_a, the coefficient of heat transfer by convection is found from the equation

$$h = \frac{q_i - q_j}{A(t_s - t_a)} \qquad [2]$$

EXPERIMENTAL RESULTS

The experimental results are represented in Figs. 9 to 13, inclusive. They comprise 92 experiments under seven different starting conditions. Six of these were obtained by using the different starting pieces described in Table 1, and the seventh occurred when no particular care was taken to insure a continuous surface in gluing the paper ring in place. Sixteen runs were taken with this somewhat rough junction and twelve when the junction had been carefully smoothed with wax.

The range of the experiments is shown in Table 2.

TABLE 2 RANGE OF EXPERIMENTS

Item	Symbol	Units	Minimum	Maximum
Lengths ratio	L_{st}/L_{tot}		0.101	0.606
Air velocity	v_a	ft/sec	9.8	148.0
Air temperature	t_a	°F	43.0	83.5
Reynolds number	N_{Re}		45300	1450000
Heat input	q_i	B/hr	42.9	362.0
Heat losses	q_j	B/hr	2.0	9.0
Heat transferred by convection	q_c	B/hr	39.5	353.5
Surface temperature	t_s	°F	110.8	146.9
Temperature difference	$t_s - t_a$	F	55.8	86.6
Coefficient of heat transfer by convection	h	$Bhr^{-1}ft^{-2}F^{-1}$	2.55	26.2
Nusselt number	N_{Nu}		126	2660
Nusselt number for $L_{st}/L_{tot} = 0$	$(N_{Nu})_0$		146	2420

For each run the Reynolds number, N_{Re}, and Nusselt number, N_{Nu}, were calculated from the following equations of definition

$$N_{Re} = \frac{v_a L_{tot}}{\nu} \qquad [3]$$

and

$$N_{Nu} = \frac{h L_{tot}}{k} \qquad [4]$$

where ν is the kinematic viscosity and k the thermal conductivity of the air.

Since the characteristics of the boundary layer at any point depend on the total length from the start of hydrodynamic action to the point under consideration, the two dimensionless groups were formed with L_{tot} and not L_{th} as characteristic length. The former has been previously defined as $L_{tot} = L_{st} + L_{th}$. The definition of the starting length L_{st} is based on the fiction that every surface element of the starting piece was placed on a cylindrical surface having the radius of the heating cylinder. This is correct for the cylindrical sections of the starting pieces, but somewhat arbitrary, though reasonable for the nose portions. It can be assumed that L_{st} is a fair equivalent of the starting length of a plane surface.

Also the choice of temperatures at which ν in Equation [3] and k in Equation [4] should be taken is rather arbitrary. However, it is of secondary importance only because the temperature differences used were not too high (see Table 2). Since the Reynolds number is influenced by no other temperature but t_a in the starting section and partly by the same in the heating section, the kinematic viscosity ν was taken at this temperature. The thermal conductivity k, on the other hand, enters the process only in the heating section and depends a good deal on the film temperature $t_f = (t_a + t_s)/2$. Therefore it was taken at this temperature.

Figs. 9 to 13, inclusive, show plots of N_{Nu} versus N_{Re} for the various cases of a constant ratio, L_{st}/L_{tot}. The influence of the Prandtl number does not appear since all runs were made with air at virtually the same Prandtl number.

According to these figures the transition to turbulence was generally finished at $N_{Re} = 300{,}000$, with the rough junction, however, already at $N_{Re} = 100{,}000$ (see Fig. 12). It is striking that the relatively small roughness of the joint was sufficient to increase N_{Nu} by almost 10 per cent.

Fig. 13 shows the thermal behavior of the specimen in the laminar range and the starting of the transition in the runs where the ellipsoidal and the long conical nosepieces were used.

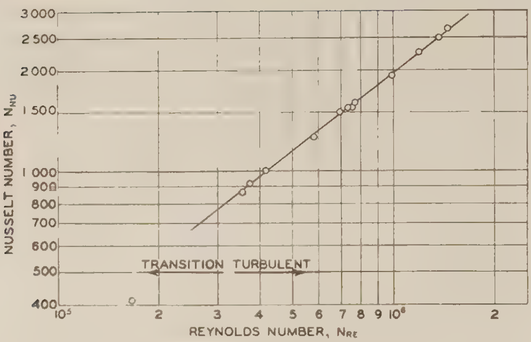

FIG. 9 EXPERIMENTAL DATA FOR L_{st}/L_{tot} = 0.606

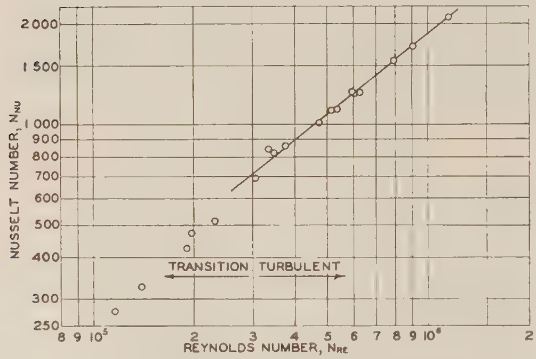

FIG. 10 EXPERIMENTAL DATA FOR L_{st}/L_{tot} = 0.508

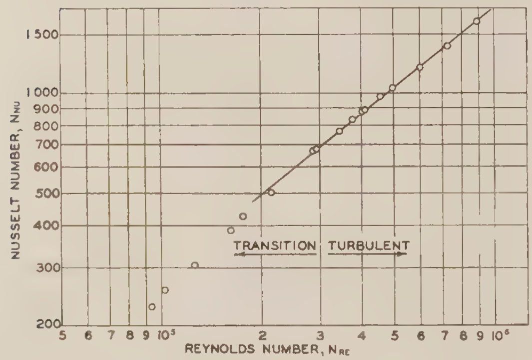

FIG. 11 EXPERIMENTAL DATA FOR L_{st}/L_{tot} = 0.347

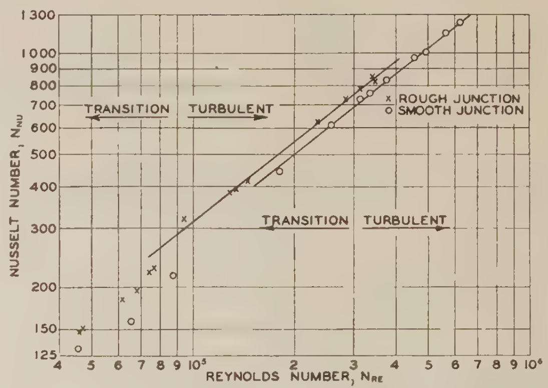

FIG. 12 EXPERIMENTAL DATA FOR L_{st}/L_{tot} = 0.101

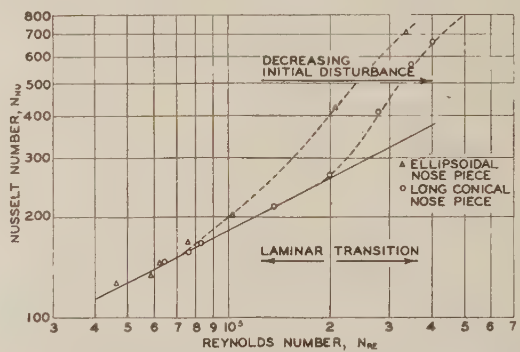

FIG. 13 EXPERIMENTAL DATA FOR L_{st}/L_{tot} = 0.122 AND 0.220 WITH ELLIPSOIDAL AND CONICAL NOSEPIECES

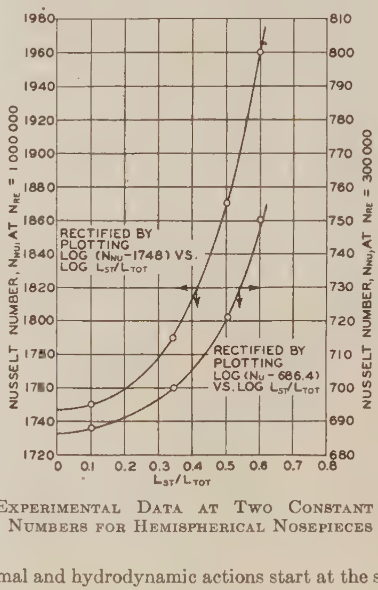

FIG. 14 EXPERIMENTAL DATA AT TWO CONSTANT REYNOLDS NUMBERS FOR HEMISPHERICAL NOSEPIECES

CORRELATION AND ANALYSIS OF THE RESULTS

Mean Heat Transfer. In Fig. 14 the Nusselt number is plotted versus L_{st}/L_{tot} for two constant values of N_{Re} in the turbulent range. These curves can be represented by an equation of the type

$$N_{Nu} = C_0(N_{Re})^m[1 + C_1(L_{st}/L_{tot})^n] = C(N_{Re})^m \quad [5]$$

or for N_{Re} = const

$$N_{Nu} = C_2 + C_3(L_{st}/L_{tot})^n \quad [6]$$

Determining the constants from the experimental data, the following equation for the turbulent region of the boundary layer was obtained

$$N_{Nu} = 0.0280(N_{Re})^{0.80}[1 + 0.40(L_{st}/L_{tot})^{2.75}] \quad [7]$$

If the thermal and hydrodynamic actions start at the same point, Equation [7] reduces to

$$N_{Nu} = 0.0280(N_{Re})^{0.80} \quad [8]$$

Fig. 15 shows a correlation of the experimental data based on these equations.

Whereas Equations [5] to [8], inclusive, hold only for fully established turbulence, there will also be an influence of the hydrodynamic starting length on the heat transfer for a laminar boundary layer. However, the present experimental arrangement allowed going back to Reynolds numbers in the streamline range only when $L_{st}/L_{tot} \gtreqless 0.220$. In this range the influence upon the turbulent layer would be only 0.6 per cent as a maximum, according to Equation [7]; no influence could be

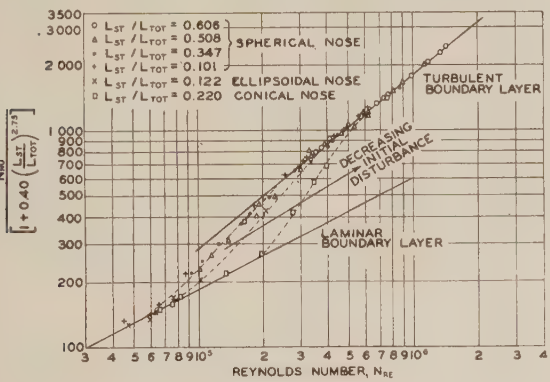

FIG. 15 FINAL CORRELATIONS OF EXPERIMENTAL DATA

observed in the mentioned range of laminar layers. The results of these tests were represented by the equation

$$N_{Nu} = 0.590(N_{Re})^{0.5} \quad \text{[9]}$$

Fig. 15 shows clearly the effect of the laminar and turbulent layers on the heat transfer. As could be predicted from the theory of boundary layers, there is no sharp critical Reynolds number but a transition, depending on the initial disturbances and the shape of the nosepieces. Increasing N_{Re}, the transition starts and ends earlier with spherical than with ellipsoidal nose and last with the long conical nosepiece. The cylindrical part of the starting length had no appreciable influence on the transition when the nosepiece was the same (spherical). In the fully developed turbulent range the points for all starting pieces fall into one line.

Local Heat Transfer. Information about the influence of hydrodynamic starting sections on the local coefficients of heat transfer is of considerable interest[11] and can be obtained from Equations [5] and [7]. However, since differentiation of an empirical equation is a delicate matter, the accuracy of the result should not be overestimated.

Considering a surface of arbitrary total length, L_{tot}, and constant starting length, L_{st}, and referring to the denotations in Fig. 16, Equation [5] can be written as

$$(N_{Nu})_{x,m} = C_0\,(N_{Re})_x{}^m\left[1 + C_1\left(\frac{x_1}{x}\right)^n\right] \quad \text{[10]}$$

where x is an arbitrary distance from the leading edge in the range from $x = x_1 = L_{st}$ to $x = L_{tot}$, the subscript x stands for this distance, the subscript x, m for the mean between $x = x_1$ and $x = x$.

Hence Equation [10] can be written as

$$\frac{h_{x,m}\,x}{k} = C_0\left(\frac{v_a x}{\nu}\right)^m\left[1 + C_1\left(\frac{x_1}{x}\right)^n\right] \quad \text{[11]}$$

[11] Capt. M. Tribus, Air Technical Service Command, Wright Field, Dayton, Ohio, directed our attention to this fact.

Let q' be the rate of heat transferred over the length $x - x_1$, from a surface of unit width, by virtue of the temperature difference $\theta_s = t_s - t_a$; then

$$q'/\theta_s = h_{x,m}\,(x - x_1) \quad \text{[12]}$$

Substituting $h_{x,m}$ from Equation [11] and differentiating with respect to x

$$h_x = \frac{dq'}{\theta_s dx} = kC_0\left(\frac{v_a}{\nu}\right)^m x^{m-1}\left[m + (1-m)\frac{x_1}{x} - (n-m)C_1\left(\frac{x_1}{x}\right)^n + (1+n-m)C_1\left(\frac{x_1}{x}\right)^{n+1}\right] \quad \text{[13]}$$

q' is a function of x and $h_{x,m}$; but $h_{x,m}$, for given x_1, is a function of x only. Hence q' is a function of x and dq' is a perfect differential.

Substituting the numerical values of C_1, m, and n from Equation [7]

$$(N_{Nu})_x = \frac{h_x \cdot x}{k} = C_0(N_{Re})_x{}^{0.8}\left[0.8 + 0.2\,\frac{x_1}{x} - 0.78\left(\frac{x_1}{x}\right)^{2.75} + 1.18\left(\frac{x_1}{x}\right)^{3.75}\right] \quad \text{[14]}$$

The local Nusselt number at the start of the heating section is obtained by taking $x = x_1 = L_{st}$. Substituting this in Equation [14]

$$(N_{Nu})_{x=L_{st}} = 1.4\,C_0(N_{Re})_{x=L_{st}}{}^{0.8} \equiv (N_{Nu})_{L_{st}/L_{tot} = 1} = 0.0392\,(N_{Re})_{x=L_{st}}{}^{0.8}$$

as could be expected, considering the form and figures of Equation [7]

For $L_{st} = 0$ or $L_{tot} = \infty$, that is, $x_1/x = 0$

$$(N_{Nu})_{x=\infty} = 0.8\,C_0(N_{Re})_{x=\infty}{}^{0.8} \equiv 0.8\,(N_{Nu})_{L_{st}/L_{tot} = 0} = 0.0224(N_{Re})_{x=\infty}{}^{0.8}$$

These are the limits in which $(N_{Nu})_x$ may vary owing to the influence of starting length.

The local Nusselt number at the end of the heating length is obtained by taking $x = L_{tot}$. In the present experiments L_{st}/L_{tot} was varied from 0.101 to 0.606. For these two cases Equation [14] yields the local Nusselt numbers

$$(N_{Nu})_{x=L_{tot}} = 0.819\;C_0(N_{Re})^{0.8} \text{ and } 0.905\;C_0(N_{Re})^{0.8}$$

respectively, to be compared with the mean Nusselt numbers $N_{Nu} = 1.0007\;C_0(N_{Re})^{0.8}$ and $1.1005\;C_0(N_{Re})^{0.8}$, respectively.

The latter value is still considerably below the highest possible maximum $1.4\;C_0(N_{Re})^{0.8}$, whereas the lower limit $C_0(N_{Re})^{0.8}$ has virtually been reached in the present experiments.

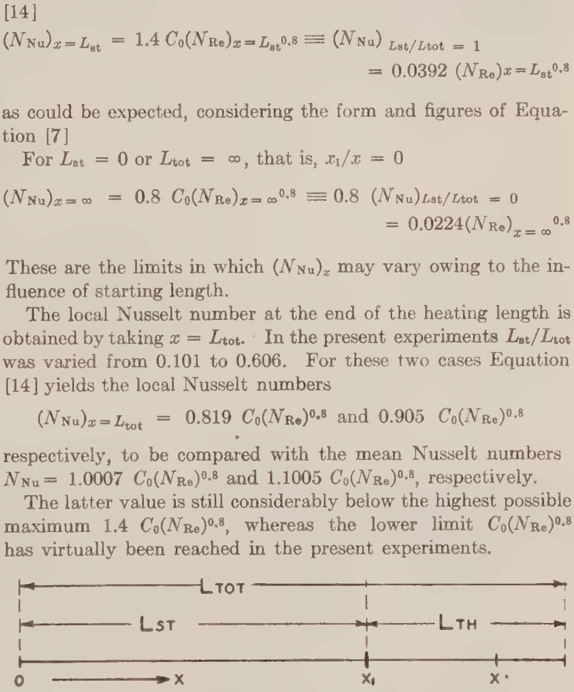

FIG. 16 DENOTATIONS OF LENGTH SECTIONS

Influence of Surface Curvature in General. It will have to be proved by further experiments how far the present results can be applied to surfaces of other curvatures and especially to plane plates. However, an estimate of the influence of curvature on the heat transfer can be obtained in the following way:

It is known that the temperature field and velocity field in a fluid flowing along a surface are almost similar. This had been assumed already by Osborne Reynolds and has been proved ex-

perimentally for the flow in a tube by Pannell (9) and on a plane surface by Éliás (10). Let y be the co-ordinate in the direction perpendicular to the surface, with $y = 0$ at the surface and $y = b$ at the imagined interface between boundary layer and bulk of the flow. The velocities and temperatures at those limits may be $v = 0$ and $v = v_b$, $t = t_s$ and $t = t_a$, respectively. Then because of the existing analogy

$$\frac{\theta}{\theta_s} = \frac{t - t_a}{t_s - t_a} = \frac{v_a - v}{v_a}$$

The rate of heat given up from a plane surface of width W to air in parallel flow between the limits $x = L_{st}$ and $x = x$ is

$$q_p = \int_0^b W\rho c_p v(t - t_a)dy = \int_0^b W\rho c_p \theta_s(v - v^2/v_a)dy \quad [15]$$

The corresponding rate of heat given up by a cylindrical surface of circumference $W = 2\pi r$ to air in parallel flow is

$$q_c = \int_0^b \rho c_p 2\pi(r + y)v(t - t_a)dy = \int_0^b W\rho c_p\theta_s(1 + y/r)(v - v^2/v_a)dy \quad [16]$$

Using the same symbols for v and b in both equations involves the assumption that these quantities are not different for flow parallel to a cylinder or to a plane plate. This assumption seems to give a fair approximation, even for relatively thick boundary layers, according to an observation of Eckert and Weise which will be discussed later on.

Then, according to Equations [15] and [16], the coefficient of heat transfer on a cylinder can be found by multiplying the coefficient for a plane surface by the factor

$$F = 1 + \frac{N - M}{M} = \frac{N}{M} \quad [17]$$

where

$$M = \int_0^b (v - v^2/v_a)dy \text{ and } N = \int^b (1 + y/r)(v - v^2/v_a)dy$$

Influence of Surface Curvature in the Laminar Range. For the laminar section of the boundary layer, the velocity distribution is exactly known from the calculations of Blasius. According to Pohlhausen (11) a close approximation is obtained by the equations

$$v = \left(\frac{2y}{b} - \frac{2y^3}{b^3} + \frac{y^4}{b^4}\right) v_a \quad [18]$$

and

$$b = 5.83\, x(N_{Re})_x^{-0.5} \quad [19]$$

Equation [19] represents the true thickness of the boundary layer and not the so-called displacement thickness for which, according to Pohlhausen, the factor 1.75 instead of 5.83 would have to be entered.

Substituting from Equation [18] in [17] and integrating yields

$$M = \frac{142}{315} bv_a$$

$$N - M = \frac{5}{126}\frac{b^2}{r} v_a$$

and

$$F = 1 + \frac{25}{284}\frac{b}{r} \quad [20]$$

where r is the radius of the surface, 0.054 ft in our case.

According to Equations [19] and [20], the correction factor F increases linearly with x and inverse linearly with $\sqrt{(N_{Re})_x}$. The extreme case in our experiments occurred with $L_{tot} = 0.759$ ft (ellipsoidal nosepiece) at $N_{Re} = 46{,}200$ (see Fig. 13). By susbstitution in Equations [19] and [20] $b = 0.0207$ ft and $F = 1.0338$ were found. Hence in our most extreme case a 3.4 per cent higher value had to be expected for the cylinder than for the plane plate.

From Fig. 13 it is seen that this is just the order of magnitude of scattering of our data. So, the differences of the coefficients of heat transfer for cylindrical and plane surfaces, if any, are in the limits of our accuracy. Should, however, really no difference exist in our range, then this would have the following consequence:

A layer of the bulk flow of unit thickness adjacent to the surface of the boundary layer at $y = b$ has a greater volume on the cylinder than on the plane surface of equal area. Hence its wiping capacity will also be greater and the thickness b may be reduced. Therefore, b in Equation [16] would be smaller than in Equation [15]. However, if Equation [18] were still holding for the velocity distribution, $(dv/dy)_{y=0}$ and, consequently, also $-(dt/dy)_{y=0}$ would become steeper which would mean a greater heat transfer. Hence in this case Equation [18] would have to be replaced by an equation giving a less steep velocity slope at $y = 0$.

Influence of Surface Curvature in the Turbulent Range. Referring to the turbulent section of the boundary layer, again Equations [15], [16], and [17] can be used. Equation [18] may be replaced by

$$v = (y/b)^{1/7}v_a \quad [21]$$

This is von Kármán's well-known equation which, notwithstanding its approximative character, may be used here, for the following reason: According to experiments about the flow in cylindrical tubes by Nikuradse (12), the exponent in Equation [21] decreases from $^1/_6$ to less than $^1/_{10}$ with increasing Reynolds number (13). However, it is not known how the Reynolds numbers for unbounded flow on surfaces are correlated to those for the flow in tubes. The range of turbulent flow covered in the present experiments ($N_{Re} \approx 300{,}000$ to 1,500,000) may correspond to the range between the so-called upper critical number in tubes, $N_{Re} \approx 10{,}000$ or 20,000 (see (14)[12] and (15)[13]) to $N_{Re} \approx 75{,}000$, in which range the exponent of Equation [21] equals about $^1/_7$.

Applying an equation of von Kármán (16), the thickness of the boundary layer can be expressed by

$$b = 0.366\, x'(N_{Re}')_{x'}^{-0.2} \quad [22]$$

where

$$x' = x - x_0$$

$$(N_{Re}')_{x'} = \frac{v_a x'}{\nu}$$

and $x = x_0$ is defined as that distance from the leading edge at which fully established turbulence should start in order to bring the boundary layer to the same thickness b at $x = x_{cr}$ which it actually attains in the laminar range (see Fig. 1). In other words, extending the turbulence curve in Fig. 1 to the left, would lead to $b = 0$ at a fictitious distance $x = x_0$.

Hence for $x = x_{cr}$ Equations [19] and [22] would yield the same b so that

$$5.83\, x_{cr}\left(\frac{v_a x_{cr}}{\nu}\right)^{-0.5} = 0.366\,(x_{cr} - x_0)\left[\frac{v_a(x_{cr} - x_0)}{\nu}\right]^{-0.2}$$

Solving for x_0 yields

[12] Loc. cit., p. 34.
[13] Loc. cit., p. 172.

$$x_0 = x_{cr}\left[1 - 31.82\left(\frac{v_a x_{cr}}{\nu}\right)^{-3/8}\right] \qquad [23]$$

From Equations [22] and [23] by some transformation

$$b = 0.366\, x \left\{1 - \frac{x_{cr}}{x}\left[1 - 31.82\left(\frac{v_a x_{cr}}{\nu}\right)^{-3/8}\right]\right\}^{0.8}\left(\frac{v_a x}{\nu}\right)^{-0.2}$$

or

$$b = 0.366\, x \left\{1 - \frac{(N_{Re})_{cr}}{(N_{Re})_x}\left[1 - 31.82\,(N_{Re})_{cr}^{-3/8}\right]\right\}^{0.8}(N_{Re})_x^{-0.2} \qquad [24]$$

The thickness b increases with increasing x and with decreasing $(N_{Re})_{cr}$ up to a maximum which would occur at $(N_{Re})_{cr} = 2908$.

One extreme case in our experiments occurred with a cylindrical starting piece having a hemispherical nose ($L_{tot} = 1.69$ ft) and with $(N_{Re})_{cr} \approx 50{,}000$ and $N_{Re} \approx 1{,}500{,}000$ (see Fig. 15). Using these values in Equation [24], $b = 0.0353$ ft was found.[14]

Substituting now from Equation [21] in [17] and integrating

$$M = \frac{7}{72}\, b v_a$$

$$N - M = \frac{7}{240}\,\frac{b^2}{r}\, v_a$$

and

$$F = 1 + \frac{3}{10}\,\frac{b}{r} \qquad [25]$$

With $b = 0.0353$ and $r = 0.054$, $F = 1.196$ was obtained.

The other extreme occurred with the spherical nose without cylindrical starting piece ($L_{tot} = 0.74$ ft) and with $N_{Re} = (N_{Re})_{cr} = 50{,}000$. Here $b = 0.01347$ ft and $F = 1.075$ were found.

This means that for the used cylindrical surface, heat-transfer coefficients, 7.5 to 19.6 per cent higher than for a plane surface may be expected. Should actually no differences exist, then the same consequences would have to be considered as drawn when dealing with the corresponding differences for a laminar boundary layer.

Comparison With Previous Work

The present results will be compared with theoretical and experimental results of previous investigators. Most of them can be reduced to starting length zero and represented by the form

$$N_{Nu} = C_0 (N_{Re})^m \qquad [26]$$

Values of C_0 and m, according to different workers, are shown in Tables 3 and 4, each in chronological sequence. For comparison it may first be noted that the present experiments led to $C_0 = 0.590$ and $m = 0.50$ for laminar boundary layer, and to $C_0 = 0.0280$ and $m = 0.80$ for turbulent layer.

Strangely enough, the first notable investigations on this subject date less than 25 years back and are purely theoretical.

For the laminar range Pohlhausen (17) in 1921 derived the following equation

$$N_{Nu} = 0.664 (N_{Re})^{1/2} (N_{Pr})^{1/3} \qquad [27]$$

Substituting $N_{Pr} = 0.71$ which holds for the average temperature of the present experiments, Equation [27] takes the form of Equation [26] with $C_0 = 0.592$ and $m = 0.50$ in excellent agreement with the result of the present experiments. Incidentally, Pohlhausen's derivation is one of the few existing direct solutions of Stokes's equations of viscous flow in combination with the equation of thermal conduction in moving fluids. The attempts made by Boussinesq, King, and Aichi before Pohlhausen, and by Lévêque afterward, failed because of oversimplified assumption. Lévêque, for instance, assumed a linear velocity distribution, instead of Equation [18]. These mathematical attacks of the problem have been thoroughly described by Drew (18).

For turbulent flow, Latzko (19), also in 1921, derived the following equation for the rate of heat flow:

$$q_{Lst/Ltot=0} = 0.0356\, L_{th} W \rho c_p v \theta_s (N_{Re})^{-0.2}$$

which can readily be converted to

$$N_{Nu} = 0.0356\,(N_{Re})^{0.80} N_{Pr} \qquad [28]$$

This equation has the same exponent as was found in our experiments. Since Latzko's derivation is restricted to $N_{Pr} = 1$, it holds only approximately when $N_{Pr} \neq 1$. However, substituting $N_{Pr} = 0.71$, $C_0 = 0.0253$ was obtained, our experimental value of C_0 being about 11 per cent higher, in satisfactory agreement with the presented theoretical analysis concerning the influence of surface curvature to the results.

Latzko's derivation is based on von Kármán's form of Equation [24], that is, $(N_{Re})_{cr}/(N_{Re})_x$ is neglected compared to 1. Instead of this, the fictitious point $x = x_0$ (see Fig. 1) may be taken as zero point of a system, denoted by prime signs; then

$$N_{Nu} = \frac{hL_{tot}}{k} = \frac{h(L_{tot} - x_0)}{k}\,\frac{L_{tot}}{L_{tot} - x_0} = N_{Nu}'\,\frac{L_{tot}}{L_{tot} - x_0}$$

and from Equation [28], using $N_{Pr} = 0.71$

$$N_{Nu}' = 0.0253\,(N_{Re}')^{0.8}$$

After some transformations

$$N_{Nu} = 0.0253 \left\{1 - \frac{(N_{Re})_{cr}}{N_{Re}}\left[1 - 31.82 (N_{Re})_{cr}^{-3/8}\right]\right\}^{-0.2} N_{Re}^{0.8} = C_0 N_{Re}^{0.8} \qquad [29]$$

The expression in curved brackets increases with N_{Re} and with decreasing $(N_{Re})_{cr}$ up to the previously mentioned maximum. Hence again the extreme occurred with $N_{Re} = 1{,}500{,}000$ and $(N_{Re})_{cr} = 50{,}000$. The calculation yields $C_0 = 0.0254$, compared to Latzko's value 0.0253. Hence $C_0 = 0.0280$, observed with our cylindrical surface, is about 10 per cent higher than the theoretical value for a plane surface, in satisfactory agreement with the average (13.5 per cent) of the theoretical values, calculated at the end of the preceding section.

TABLE 3 CONSTANT C_0 AND EXPONENT m OF EQUATION [26] FOR LAMINAR BOUNDARY LAYER

Authors	Kind of work	Surface	C_0	m
Pohlhausen	Theoretical	Plane	0.592	0.50
Fage and Falkner	Experimental	Plane	0.750	0.50
Jakob and Dow	Experimental	Cylindrical, diam, 1.3 in.	0.590	0.50

TABLE 4 CONSTANT C_0 AND EXPONENT m OF EQUATION [26] FOR TURBULENT BOUNDARY LAYER

Authors	Kind of work	Surface	C_0	m
Latzko	Theoretical	Plane	0.0253	0.80
Juerges	Experimental	Plane	0.0452	0.775
Juerges	Experimental[a]	Plane	0.0322	0.80
Colburn	Correlation	Plane	0.0320	0.80
Seibert	Theoretical	Plane	0.0423	0.786
Seibert	Theoretical[a]	Plane	0.0349	0.80
Jakob and Dow	Experimental	Cylindrical, diam, 1.3 in.	0.0280	0.80

[a] Results reduced to $m = 0.80$.

[14] This procedure involves the assumption that Equation [22] can be used through and past the transition region (see Fig. 1); the same assumption will be made in deriving Equation [29]. Within the transition region, however, it yields only approximation results.

Only one year after these two papers the first publication of Juerges' experiments appeared (20), followed in 1924 by a detailed report (21). He measured the heat transfer from a vertical plane copper plate, 1.64 ft × 1.64 ft, to air flowing parallel to the surface at velocities up to 100 ft/sec, surface temperatures from 115 to 140 deg F, and air temperature of 68 deg F. The hydrodynamic starting length in these experiments was 1.02 ft. For a polished plate Juerges found

$$h = 0.4497\, v^{0.775} \qquad [30]$$

which can be generalized into our standard form, Equation [26], by using the term in square brackets of Equation [7]. In this way $C_0 = 0.0452$ and $m = 0.775$ were obtained. In order to simplify the comparison with our values, Juerges' results were also represented with $m = 0.80$. This led to $C_0 = 0.0322$ and to values of N_{Nu} which differ less than ±2 per cent from the values obtained by Equation [30] between $N_{\mathrm{Re}} = 300{,}000$ and $1{,}500{,}000$, but are 15 per cent higher than our values based on $C_0 = 0.0280$.

Much of this difference is probably due to a blunt leading edge of the starting surface.[15] As mentioned in the first section of this paper, turbulence will start immediately past such an edge with the effect that considerably more heat is transferred in the thermal section. Therefore, Juerges' results cannot be directly compared with the present ones in which a laminar boundary layer preceded the turbulent layer. Reference is made to Fig. 12 which shows that even a small roughness in the starting section can have an effect on the Nusselt number of similar order of magnitude as was observed between Juerges' and the authors' experiments.

In the experiments of Éliás (10) which covered almost the same range of v_a and θ_s as those of Juerges, a starting piece 3.94 in. long, with sharpened leading edge and heating lengths of 3.94 to 15.76 in. were used.

Éliás represented his experimental results by curves of the velocity and temperature distribution in the boundary layer and used these curves to determine the heat carried past in the air stream by graphical integration, according to the first part of Equation [15].

From Éliás' reported values we calculated the heat-transfer coefficients and the Reynolds and Nusselt numbers with L_{tot} as characteristic length. Again the starting length was considered by means of Equation [7].

Éliás' and Juerges' data, reduced to $L_{st} = 0$, are compared with the present results in Fig. 17. It is seen that most of Éliás' points fall in between Juerges' and our line. They show an unusual steep increase with Reynolds number ($m > 0.9$). Seibert (22) considers this as evidence of not yet finished transitions and in fact, the points of Fig. 17 fit rather well with our observations in the state of transition shown in Fig. 15. The instability due to transition and the indirect determination of heat convection from flow and temperature measurements in the boundary layer may account for the considerable scattering of the points.[16]

In contrast to the steep increase of Éliás' values, Taylor and Rehbock (23) arrived at an improbably small exponent ($m = 0.725$), using Juerges' data and their own experiments, per-

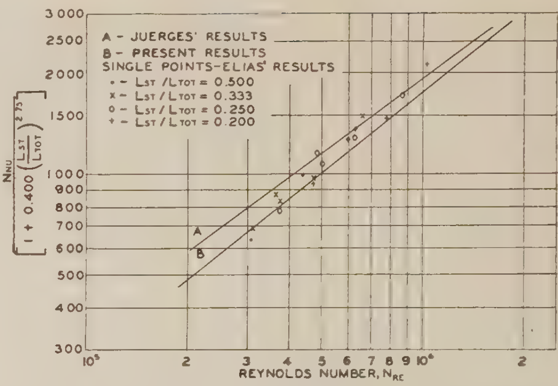

Fig. 17 Comparison of Present Results for a Turbulent Boundary Layer With Previous Experiments With Flat Plates

formed on a 6-in-sq copper plate at 75 to 235 ft/sec air velocity. Unfortunately, they did not indicate the value of the hydrodynamic starting length in their experiments. For this reason a comparison with present results is not possible.

Fage and Falkner (24) determined the heat transfer to air from platinum foils, 0.00127 cm thick, 1.27 cm wide, and from 0.333 to 1.27 cm long, in the laminar-boundary range. The air velocity was varied from 20 to 70 ft/sec, the air temperature was 72° F, the foil temperature was about 340°F. For the exponent m the usual value of 0.5 was found, for the constant C_0, however, the high value 0.750. This may be ascribed to the fact that with such short heating surfaces, the starting-section conditions prevail. Latzko (19)[17] analyzing an analogous case in the turbulent range found that in the starting section the constant C_0 should be higher than in established turbulence, whereas the exponent would remain unchanged. The same may hold for the laminar range.

Colburn (25) accepted Pohlhausen's equation and made a correlation according to which Fage and Falkner's results would be in agreement with this equation. Because we had found their values to be 27 per cent too high, we asked Dr. Colburn for an explanation of this discrepancy, and he found that due to a slide-rule error, all points representing Fage and Falkner's values in his Fig. 20 should be raised by 21 per cent. This explains the greatest part of the mentioned difference.

Colburn further correlated the experimental results of Juerges and Éliás in the turbulent region and expressed them by the following equation

$$\frac{h\, L_{\mathrm{th}}}{k} = 0.0321 \left(\frac{v_a L_{\mathrm{th}}}{\nu}\right)^{0.80}$$

Using[18] Juerges' $L_{st}/L_{\mathrm{tot}} = 0.383$ and converting from ν_f at film temperature (about 98°F) as used by Colburn, to ν at bulk temperature (about 68°F), correcting also for the influence of the starting length, according to Equation [7], Colburn's equation is reduced to our standard form with $m = 0.80$ and

$$C_0 = 0.0321 (L_{\mathrm{tot}}/L_{\mathrm{th}})^{0.2} [1 + 0.40 (L_{st}/L_{\mathrm{tot}})^{2.75}]^{-1} (\nu/\nu_f)^{0.8} = 0.0320$$

Seibert (22) also adopted Pohlhausen's theory in the laminar range. For turbulence he used the modification by which Prandtl had extended his original theory to the case of $N_{\mathrm{Pr}} \neq 1$ and in particular, ten Bosch's (3) extensions of this theory. Employing

[15] In Fig. 2 of Juerges' paper no sharpening or smoothing of the edge is indicated. Since many minor details are dealt with in the paper, the author would certainly have mentioned any sort of treatment of the leading edge.

[16] The accuracy of the experiments is unfavorably influenced by the difficulty to co-ordinate a considerable number of measured local velocities and temperatures to the exact values of y in a total range of boundary thickness of about 1/2 in. only. Moreover, the surface temperature was measured by thermocouples placed on the back side of the heating box. It is very questionable whether sufficient symmetry existed on both sides; the surface thermocouples themselves may have disturbed such symmetry.

[17] Loc. cit., p. 283, Fig. 7A.

[18] Éliás' range of ratios was $L_{st}/L_{\mathrm{tot}} = 0.2$ to 0.5.

Seibert's exponent $m = 0.786$, but correcting[19] for the difference in N_{Pr}, the value $C_0 = 0.0423$ was obtained. His results can be covered also by using $m = 0.80$ and $C_0 = 0.0349$. Comparison then shows that Seibert's theoretical values of N_{Nu} for plane plates are 25 per cent higher than those measured on a cylinder in the present work. This is at variance with Latzko's theoretical results which cannot be expected to be seriously in error at a Prandtl number so close to unity. However, Seibert's relation is based on different assumptions which would need to be checked, as he states himself.

Seibert also indicated the following formula for the transition state

$$N_{Nu} = 0.00178\ (N_{Re})^{0.986} \qquad [31]$$

It yields too low values compared to the transition lines in Fig. 15.

Also his method of inserting Éliás' results between a maximum and minimum of Nusselt numbers is questionable. The heating section gives up more heat (q_{max}) in actual measurements if an unheated starting length exists, than would be the case if the same total length were held at the same temperature excess θ_s as the heating section was in the actual experiments (q_{min}). Seibert, however, uses a sort of mixing rule for Nusselt numbers in his Equation [20], which is not correct.

Using Equation [7], the following is obtained

$$\frac{q_{min}}{q_{max}} = \frac{[1 - (L_{st}/L_{tot})^{0.8}](1 - L_{st}/L_{tot})}{1 + 0.4(L_{st}/L_{tot})^{2.75}} \qquad [32]$$

For the extreme ratios, employed by Éliás, $L_{st}/L_{tot} = 0.2$ and 0.5, Equation [32] yields $q_{min}/q_{max} = 0.901$ and 0.804, respectively, compared to the ratios $(N_{Nu})_{min}/(N_{Nu})_{max} = 0.945$ and 0.713, respectively, calculated according to Seibert, assuming the same exponent, $m = 0.8$. In general, much insight in the mechanism is not obtained by Seibert's method of asking what the heat transfer in a heated section would have been if the starting length had also been heated, whereas Equation [7], rough and improvable as it may be, considers the influence of the starting length in a straightforward way.

Finally, some theoretical and experimental papers may be mentioned which deal with the laminar part of the boundary layer at very high fluid velocity. The authors are Crocco (26), Hilton (27), Eckert and Weise (28), and Eckert and Drewitz (29). The result of these investigations is that Pohlhausen's equation can be applied up to as high speeds as twice the velocity of sound, provided that the temperature of the gas t_a is replaced by the temperature t_i which the unheated and perfectly insulated surface would assume in the gas stream. This temperature may be called "impressed temperature." For $N_{Pr} = 1$, it is found by adding to the ordinary surface temperature the temperature increment due to adiabatic stopping of the gas stream, $\theta_{ad} = v_a^2/(64.4\ Jc_p)$, where J is the mechanical equivalent of heat.[20] For $N_{Pr} \neq 1$, $\theta_i = \theta_{ad}$. $\varphi(N_{Pr})$ is to be added to t_s; values of the function φ are given in Table 5. They are calculated by means of Pohlhausen's theory; the first nine values have been taken from his original paper (17), the last two from Eckert and Weise's paper (28).

Also the heat transfer in the range of turbulent layer can be calculated with the formulas developed for small velocities if t_i is employed instead of t_a.

Hence it may be assumed that the results of the present paper also can be approximately used for high velocities and for other fluids than air if t_a is replaced by t_i.

[19] Seibert's Equation [17] has been used for this slight correction.

[20] For convenience J and c_p are expressed in the dimensions given in the "nomenclature;" the number 64.4, then, is not dimensionless.

TABLE 5 VALUES OF THE FUNCTION (φN_{Pr})

N_{Pr}	0.6	0.7	0.8	0.9	1.0	1.1	Gases
$\varphi(N_{Pr})$	0.77	0.835	0.895	0.95	1.00	1.05	
N_{Pr}	7.0	10.0	15.0	100	1000		Liquids
$\varphi(N_{Pr})$	2.515	2.965	3.535	6.7	12.9		

Concerning the influence of curvature of the surface in a plane perpendicular to the main flow direction, Eckert and Weise made some tests on wires with 0.2 to 2 mm diam in the range of $v_a = 400$ to 1000 ft per sec air velocity, corresponding to $N_{Re} =$ 50,000 to 500,000. Denoting the temperature excess of the unheated wire by θ_i, it was found that the ratio θ_i/θ_{ad} depended only slightly on the ratio b/r which was varied from 0.1 to 1. In the same range, Hilton measured θ_i/θ_{ad} for a plane plate and found only about 3 per cent higher values than Eckert and Weise on the thin wires. This is in good agreement with the deviation of 3.4 per cent calculated by the present authors for $b/r = 0.384$ in the fourth part of the previous section of this paper.

Summary and Conclusions

1 Systematic measurements of the heat transfer by convection from a cylinder to a fluid flowing parallel to its axis allow a simpler technique than measurements with plane surfaces. They should yield results which can be applied to plane surfaces up to rather high Reynolds numbers when cylinders with reasonably large diameters are used.

2 The present experiments were performed with air flowing parallel to an electrically heated smooth cylinder, 8 in. long, 1.3 in. diam. Wooden nosepieces of different shapes and cylindrical pieces of different lengths served as unheated starting sections. The heated lengths were varied from 40 to 90 per cent of the total length, air velocities from 10 to 150 ft per sec, corresponding to Reynolds numbers from 45,000 to 1,500,000, based on the total length as a characteristic dimension.

3 The results of the measurements could be expressed by the equations

$$N_{Nu} = 0.590\ (N_{Re})^{0.50} \qquad [9]$$

and

$$N_{Nu} = 0.0280\ (N_{Re})^{0.80}[1 + 0.40\ (L_{st}/L_{tot})^{2.75}] \qquad [7]$$

for laminar and turbulent boundary layers, respectively, where N_{Nu} and N_{Re} are Nusselt and Reynolds numbers with the total length L_{tot} as characteristic length, and L_{st} as the starting length. Transition to turbulence started between $N_{Re} =$ 50,000 and 200,000 and was virtually finished between 250,000 and 600,000 (see Fig. 15). The start and end of transition depended mainly on the shape of the nose. Even a slight roughness in the starting section caused a considerable increase of the heat transfer.

4 An equation for the local Nusselt number was derived from Equation [7] by differentiation.

5 The influence of the cylinder curvature has been studied theoretically, considering similarity between velocity and temperature distribution and assuming that the velocity distribution in curved boundary layers is the same as for plane layers. Under these assumptions 8.8 b/r and 30 b/r would be the percentages by which the Nusselt numbers for cylinders would exceed those for plane plates in the ranges of laminar and turbulent layers, respectively, where r is the cylinder radius and the thickness of the boundary layer b is taken from Equations [19] and [24], respectively. It was further shown what changes in the theory might be advisable if such differences should not actually occur.

6 Using these differences in the laminar range of the present experiments, the heat transfer should be higher by 0 to 3 per cent than for a plane boundary layer. This seems to be in agreement with some recent experiments which Hilton and Eckert and Weise performed at much higher air velocities and smaller lengths. Fage and Falkner, however, obtained 25 per cent higher

values in tests on short platinum foils. No difference was found between the present experimental data on cylindrical layers and those from Pohlhausen's generally recognized theory for plane boundary layers.

7 Applying the theory to the turbulent range of the present experiments, a 7 to 20 per cent larger heat transfer would be expected than for plane plates. Actually the present experimental data are about 10 per cent higher than those obtained from Latzko's theory for a plane surface. On the other hand, they are 20 per cent lower than those according to a theory of Seibert which is based on different and not yet proved assumptions. Experiments in the range of turbulent boundary layers on plane surfaces have been performed by Juerges and Éliás. A blunt leading edge seems to have caused relatively high heat-transfer values in Juerges' experiments 15 per cent above the present data. Éliás' results are between Juerges' and the present data.

8 A recent paper of Eckert and Drewitz shows that Pohlhausen's theory holds up to twice the velocity of sound if the fluid temperature is replaced by a temperature impressed on the surface because of adiabatic stopping and friction of the stream. The same is claimed for the turbulent boundary layer.

It may be concluded also that the results of the present investigation can be approximately employed up to much higher than the experimental velocities and applied to other fluids when the equilibrium fluid temperature is replaced by the "impressed temperature" of the surface. However, it is planned to extend the experiments to velocities higher than those used in the present experiments.

9 According to the authors' theoretical reasoning, the heat transfer on plane surfaces should be somewhat smaller than given by Equation [7]. This would be in agreement with Latzko's theory. Seibert's theory for plane surfaces, however, leads to higher values of heat transfer than observed on the cylinder. It is intended to decide this controversial question by experiments with cylinders of different diameters, varying also the starting and heating lengths. For the time being it seems to be safe to use Equation [7] tentatively for any convex smooth surface with arbitrary starting section when the radius of curvature in a plane perpendicular to the main flow direction is more than $^1/_2$ in. The influence of the starting length, in particular, should be nearly the same for cylinders and plane surfaces.

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Temperature-Time Distribution in Rectangular Bars

By W. M. ROHSENOW,[1] M. J. ARONSTEIN,[2] AND A. C. FRANK[2]

Charts for determining temperature-time distribution in flat plates, infinite cylinders, and spheres have been published previously. The purpose of this paper is to amplify the lower end of the curves thus presented, and to give the results in graphical and in tabular form. In addition, the rapid convergence of the series solution is shown, and combined curve sheets for rectangular bars are presented.

Nomenclature

The following nomenclature is used in the paper:

t = temperature
t_s = initial temperature of bar
t_a = temperature of surrounding atmosphere
θ = time
k = coefficient of thermal conductivity
ρ = density
c_p = specific heat at constant pressure
w = half-width of bar in x-direction
u = half-width of bar in y-direction
$a = k/(\rho c_p)$ — thermal diffusivity
h = film coefficient of heat transfer
$b = h/k$
$T = (t - t_a)/(t_s - t_a)$

$$T_x = \sum_{n=1}^{\infty} X_n$$

X_n = defined in Equation [6]

Note: Any consistent set of dimensions may be used.

Introduction

When a homogeneous solid body at a uniform temperature is suddenly exposed to an atmosphere which is held at some constant temperature different from that of the body itself, heat is exchanged between the body and the atmosphere through a film on the surface of the body. The outer parts of the body approach the temperature of the atmosphere more rapidly than do the inner parts. Heat is transferred between the inner and outer parts of the body by conduction. After an infinite amount of time the solid body is uniformly at the temperature of the surrounding atmosphere.

In engineering applications the useful information concerning this transient-state heat-transfer process is of two forms. In one case it may be desirable to know the temperature distribution existing in a solid body at any instant of time; in the other case it may be desirable to know the temperature-time curve for any point in the body. An example of these cases is the hardening of a solid steel body. The solid body is heated to a uniform high temperature; then it is suddenly quenched in oil or water. The inner parts cool more slowly than the outer parts; hence there results a body of varying degrees of hardness throughout its volume. In order to determine the stresses produced in the bar by this unequal cooling, it is necessary to know the temperature distribution at any instant of time; and, in order to determine the degree of hardness at any point in the bar, it is necessary to know the temperature-time curve at that point.

Charts for determining temperature-time distribution in infinite flat plates, infinite cylinders, and spheres have been published by Gurney and Lurie (1, 2),[3] Hottel (3) and Newman (4). The purpose of this paper is to enlarge the lower end of these curves presenting the results in graphical and in tabular form; to show the rapid convergence of the series solution; and to present combined curve sheets for rectangular bars.

Since the number of variables is large and the calculations are laborious when applying the equations resulting from the mathematical solution of the problem to any given case, the series of charts constructed permits a convenient solution under any set of given conditions. From these charts the temperature-time-position relationship for two specific cases, namely, the square bar and the 2 × 1 rectangular bar, was found as a method of illustrating the procedure. It is hoped that these specific solutions along with the solution for the infinite flat plate will give a general idea of the range of values to be encountered.

Infinite Flat Plate

Consider an infinite flat plate (infinite in the Y- and Z-directions) of thickness $2w$, as shown in Fig. 1. The plate is initially at a uniform temperature t_s and is suddenly exposed to an atmos-

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

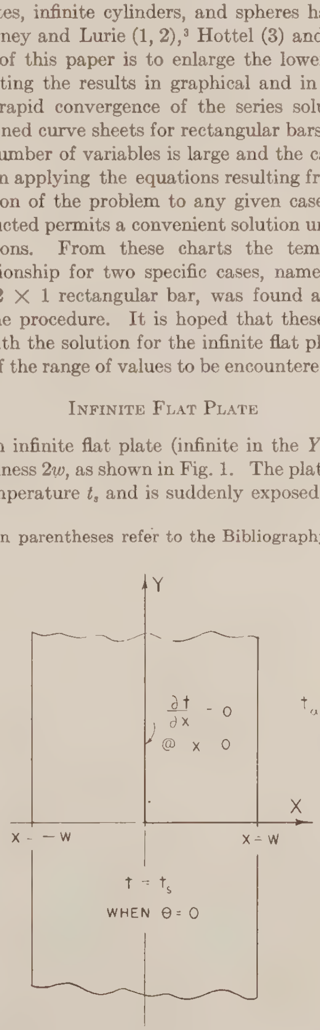

Fig. 1 Cross Section of Infinite Flat Plate

[1] Instructor in Mechanical Engineering, Yale University. Jun. A.S.M.E.

[2] Senior, Navy V-12 College Training Program, School of Engineering, Yale University. Student Mem., A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society, or the U. S. Navy, or Naval Service at large.

phere of temperature t_a, which may be colder or hotter than t_s. The heat-conduction equation for this case is

$$\frac{\partial t}{\partial \theta} = a \frac{\partial^2 t}{\partial x^2} \qquad [1]$$

where

$$a = k/(\rho c_p) \qquad [2]$$

The term a is called the thermal "diffusivity" of the material and is considered to be constant for any given solid material.

The solution of Equation [1] may be found in applied mathematics and in heat-conduction texts such as Carslaw (5). The solution is

$$\frac{t - t_a}{t_s - t_a} = \sum_{n=1}^{\infty} 2e^{-(a\theta/w^2)\delta_n^2} \cdot \frac{\cos \delta_n(x/w)}{\delta_n \csc \delta_n + \cos \delta_n} \qquad [3]$$

where δ_n, called "eigenwerte," are roots of the equation

$$bw \cot \delta_n = \delta_n \qquad [4]$$

The first five roots of Equation [4] are shown as a function of bw in Fig. 2. Note that since all are shown on the same ordinate scale, π must be added to the scaled value of δ_2, 2π to δ_3, etc. This arises from the fact that δ_1 varies from 0 to $\pi/2$, δ_2 from $\pi/2$ to $3\pi/2$, etc. Tabular values of these roots may be found in Gröber and Erk (6) and in Jahnke and Emde (7).

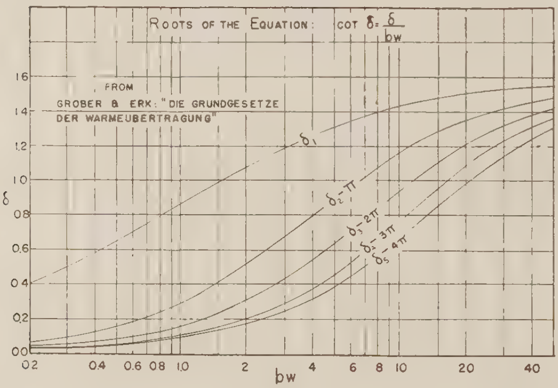

Fig. 2 Values of Eigenwerte, δ

In order to simplify the series notation of Equation [3], let it be written in the form

$$T_x = \sum_{n=1}^{\infty} X_n \qquad [5]$$

where

$$X_n = 2e^{-(a\theta/w^2)\delta_n^2} \cdot \frac{\cos \delta_n(x/w)}{\delta_n \csc \delta_n + \cos \delta_n} \qquad [6]$$

The first three terms of the series, X_1, X_2, and X_3 are shown in Fig. 3. The term X_3 is insignificant for values of $a\theta/w^2$ greater than 0.05, and term X_2 is insignificant for values of $a\theta/w^2$ greater than 0.20. The sum of the first three terms of the series Equation [5], T_x, is plotted in Figs. 4 through 8, for values of x/w = 0, 0.25, 0.50, 0.75, and 1, respectively.

These charts represent the solution of Equation [3] to determine the temperature-time distribution at 5 planes in one half of the infinite flat plate. The computed values obtained by adding

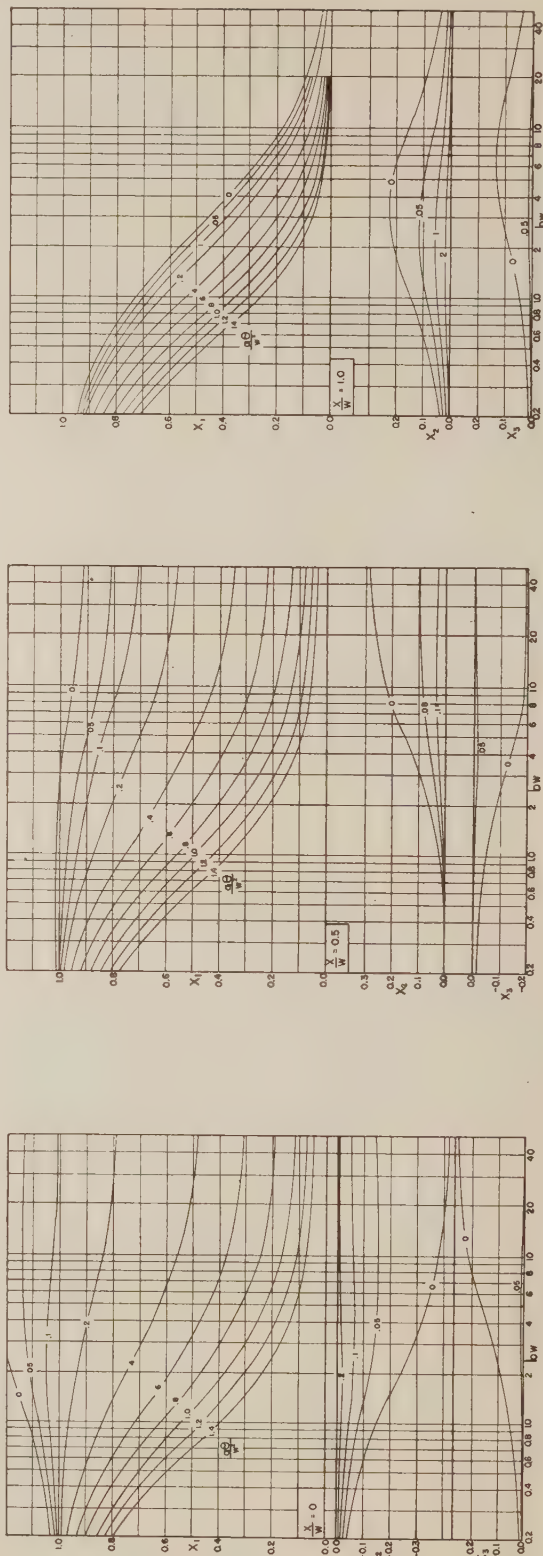

Fig. 3 First Three Terms of Series Solution, T_x

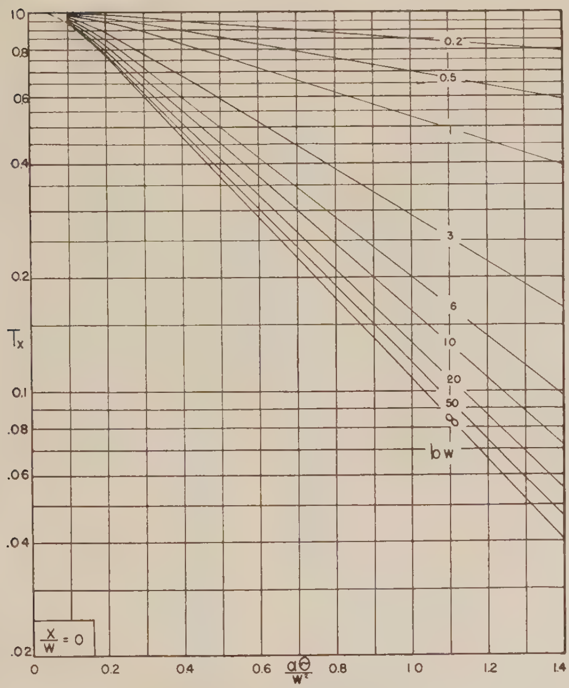

Fig. 4 T_x Versus $a\theta/w^2$ for $x = 0$

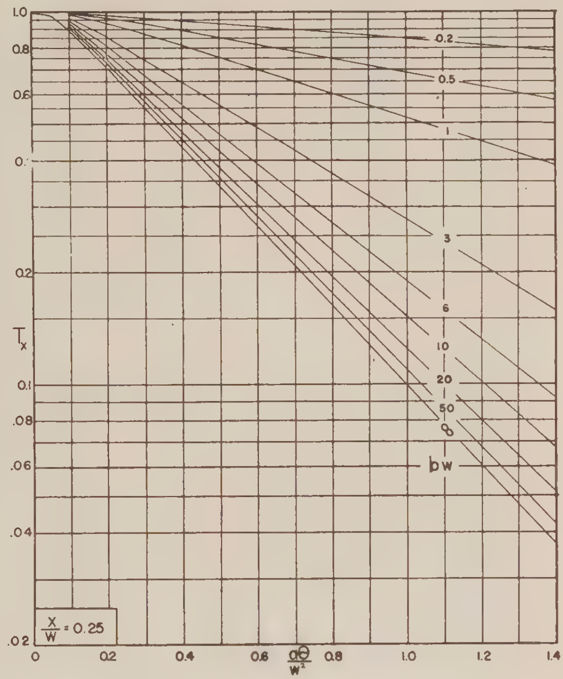

Fig. 5 T_x Versus $a\theta/w^2$ for $x = 0.25$

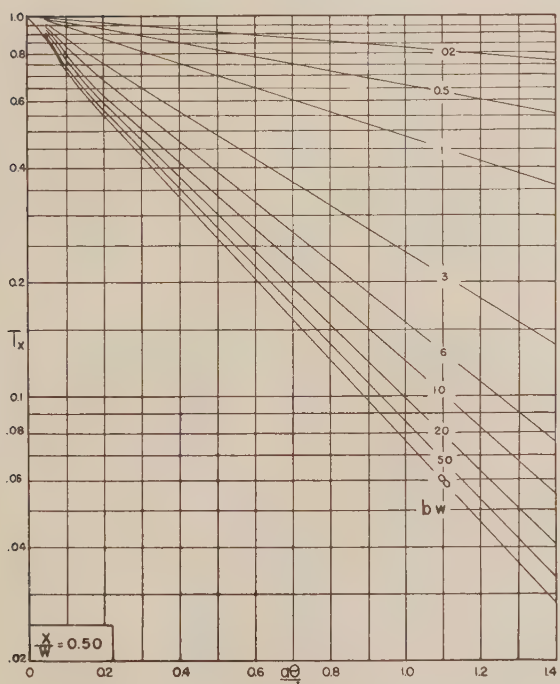

Fig. 6 T_x Versus $a\theta/w^2$ for $x = 0.50$

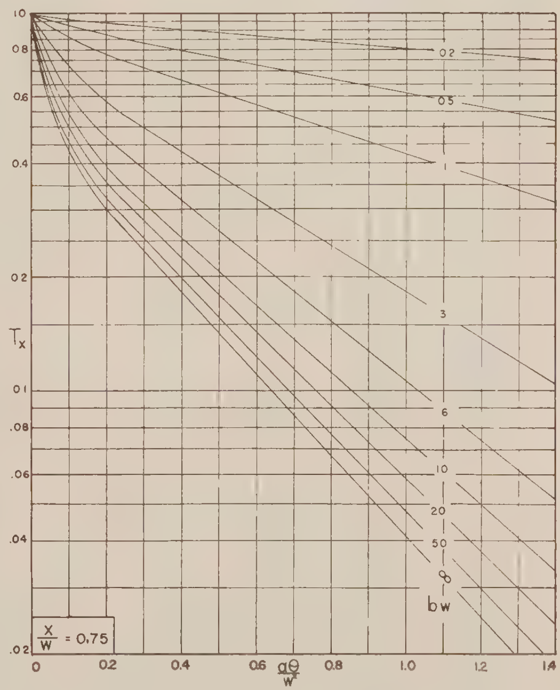

Fig. 7 T_x Versus $a\theta/w^2$ for $x = 0.75$

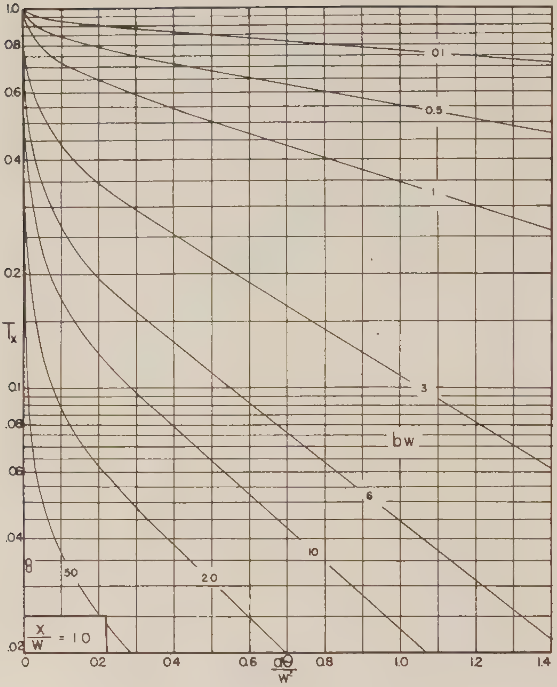

FIG. 8 T_x VERSUS $a\theta/w^2$ FOR $x = 1.0$

the first three terms of the series in Equation [5] are tabulated in Tables 1 to 5, inclusive.

INFINITE RECTANGULAR BAR AND RECTANGULAR BLOCK

The heat-conduction equation for the infinite rectangular bar represented in Fig. 9 is

$$\frac{\partial t}{\partial \theta} = a\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2}\right)\dots\dots\dots\dots\dots\dots [7]$$

By substituting in the partial differential equation, Equation [7], it may be shown that

$$T = T_x \cdot T_y \dots\dots\dots\dots\dots\dots [8]$$

is its solution, where T_x is the solution of

$$\partial t/\partial\theta = a(\partial^2 t/\partial x^2)$$

and T_y is the solution of

$$\partial t/\partial\theta = a(\partial^2 t/\partial y^2)$$

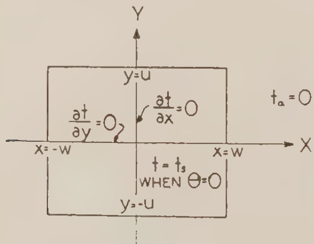

FIG. 9 QUARTER-CROSS SECTION OF INFINITE RECTANGULAR BAR

This is known as Newman's rule (4). Similarly, for a cube the differential equation is

$$\partial t/\partial\theta = a\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial Z^2}\right)\dots\dots\dots\dots [9]$$

the solution of which is

$$T = T_x \cdot T_y \cdot T_z \dots\dots\dots\dots\dots\dots [10]$$

where the T functions are defined as in Equation [8].

The functions T_x, T_y, T_z are given by Equations [3] and [5], and are plotted in Figs. 4 to 8, inclusive. In order to determine the solution T for any point in a bar or a block, evaluate T_x from Equation [5] (or from Figs. 4 to 8, inclusive) for each of the coordinate directions, thus obtaining T_x, T_y, T_z. The product of these three is the solution as indicated in Equation [10]. For an infinite rectangular bar, only T_x and T_y need be obtained.

A set of curves has been constructed in Figs. 11 and 12 for the solution of Equation [8] at the inner points of the infinite rectangular bar shown in Fig. 10. Fig. 11 is for a square bar, and Fig. 12, for a rectangular bar of side dimensions in the ratio of 2 to 1. In the case of a square bar, only 6 of the 9 positions indicated in Fig. 10 need be considered since there are three identical pairs of points because of symmetry.

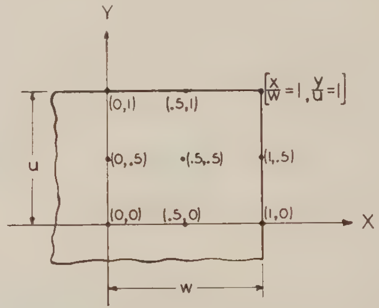

FIG. 10 QUARTER CROSS SECTION OF INFINITE RECTANGULAR BAR SHOWING POINTS FOR WHICH COMBINED SOLUTION IS PRESENTED

It will be noticed that for positions along the x-axis, the curves for the rectangular bar, Fig. 12, lie approximately midway between the curves for the square bar, Fig. 11, and the curves for the infinite flat plate, Figs. 4 to 8, inclusive. These curves are intended to indicate the range of possible values in rectangular-bar problems and to indicate a graphical method of procedure.

ILLUSTRATIVE EXAMPLE

Consider an infinite rectangular steel bar of dimensions w = 0.125 ft and u = 0.250 ft. Determine the value of temperature ratio t/t_s at the point x/w = 0.5, y/u = 0.5 at the time θ = 0.0271 hr. For a steel bar, k = 25 Btu/(hr)(sq ft) (deg F/ft), ρ = 493 lb/cu ft, and c_p = 0.11 Btu/(lb) (deg F); hence the thermal diffusivity is

$$a = 25/(0.11 \times 493) = 0.461 \text{ sq ft/hr}$$

If the film coefficient of heat transfer is

$$h = 200 \text{ Btu/(hr) (sq ft) (deg F)}$$

then

$$b = h/k = 200/25 = 8.0 \text{ ft}^{-1}$$

and

$$bw = 1.0,\ bu = 2.0,\ a\theta/w^2 = 0.8,\ a\theta/u^2 = 0.2$$

From Fig. 6 or Table 3 for

$$a\theta/w^2 = 0.8 \text{ and } bw = 1.0,\ T_x = 0.564$$

TABLE 1 VALUES OF T_x FOR $x/w = 0$

$a\theta/w^2$ / bw	0	.05	.1	.2	.4	.6	.8	1.0	1.2	1.4
.2	1.007	1.001	.999	.988	.956	.922	.888	.856	.824	.794
.5	1.010	1.001	.997	.974	.903	.830	.762	.700	.642	.590
1	1.016	1.001	.994	.952	.832	.719	.620	.535	.461	.398
3	1.038	.998	.982	.894	.684	.516	.388	.292	.220	.166
6	1.074	.999	.974	.855	.602	.418	.290	.202	.140	.0973
10	1.078	.998	.968	.829	.558	.371	.246	.164	.109	.0724
20	1.090	.997	.960	.804	.518	.331	.212	.135	.0857	.0558
50	1.098	.996	.954	.785	.492	.306	.192	.119	.0738	.0459
∞	1.103	.996	.949	.773	.475	.290	1.77	.108	.0660	.0404

TABLE 2 VALUES OF T_x FOR $x/w = 0.25$

$a\theta/w^2$ / bw	0	.05	.1	.2	.4	.6	.8	1.0	1.2	1.4
.2	1.000	1.000	.998	.984	.952	.916	.883	.851	.819	.790
.5	.997	1.000	.994	.964	.892	.819	.752	.691	.634	.582
1	.995	1.000	.986	.935	.814	.702	.606	.523	.450	.389
3	.976	.994	.964	.861	.655	.493	.371	.279	.210	.159
6	.961	.992	.946	.812	.568	.394	.274	.191	.132	.0918
10	.948	.989	.935	.781	.523	.348	.231	.154	.102	.0678
20	.931	.986	.920	.752	.482	.308	.197	.126	.0798	.0520
50	.920	.983	.910	.732	.456	.284	.178	.110	.0674	.0425
∞	.916	.982	.900	.717	.439	.268	.164	.100	.0611	.0373

TABLE 3 VALUES OF T_x FOR $x/w = 0.50$

$a\theta/w^2$ / bw	0	.05	.1	.2	.4	.6	.8	1.0	1.2	1.4
.2	.996	.997	.989	.970	.935	.901	.868	.836	.805	.776
.5	.998	.994	.975	.933	.856	.786	.722	.663	.608	.559
1	.995	.986	.950	.880	.757	.654	.564	.486	.419	.362
3	.980	.966	.888	.757	.566	.427	.321	.241	.182	.137
6	.975	.946	.842	.682	.470	.326	.226	.158	.109	.0760
10	.983	.932	.810	.640	.421	.280	.186	.124	.0823	.0547
20	.995	.913	.776	.598	.378	.243	.155	.0990	.0628	.0409
50	1.010	.898	.753	.572	.353	.220	.138	.0854	.0530	.0330
∞	1.020	.886	.735	.553	.336	.205	.125	.0764	.0467	.0314

TABLE 4 VALUES OF T_x FOR $x/w = 0.75$

$a\theta/w^2$ / bw	0	.05	.1	.2	.4	.6	.8	1.0	1.2	1.4
.2	1.0054	.984	.961	.946	.909	.874	.842	.812	.781	.753
.5	1.015	.966	.930	.877	.799	.733	.673	.618	.567	.521
1	1.026	.933	.870	.784	.667	.574	.495	.428	.368	.318
3	1.072	.844	.722	.585	.430	.323	.243	.183	.138	.104
6	1.107	.767	.619	.471	.319	.221	.154	.107	.0742	.0516
10	1.105	.686	.533	.393	.255	.169	.112	.0749	.0498	.0331
20	1.142	.650	.495	.359	.225	.144	.0919	.0586	.0372	.0242
50	1.136	.604	.453	.325	.198	.123	.0774	.0480	.0298	.0185
∞	1.114	.570	.424	.303	.182	.111	.0677	.0413	.0253	.0155

TABLE 5 VALUES OF T_x FOR $x/w = 1.0$

$a\theta/w^2$ / bw	0	.05	.1	.2	.4	.6	.8	1.0	1.2	1.4
.2	.987	.952	.933	.907	.870	.838	.806	.778	.748	.721
.5	.963	.886	.846	.792	.719	.659	.605	.556	.510	.469
1	.923	.791	.724	.655	.545	.470	.405	.349	.301	.260
3	.780	.538	.442	.349	.254	.191	.143	.108	.0814	.0614
6	.610	.350	.265	.197	.132	.0916	.0635	.0440	.0313	.0219
10	.457	.232	.171	.123	.0789	.0525	.0348	.0232	.0154	.0102
20	.272	.124	.0881	.0622	.0387	.0248	.0159	.0101	.0064	.0042
50	.116	.050	.0363	.0252	.0152	.0094	.0059	.0037	.0023	.0014
∞	0	0	0	0	0	0	0	0	0	0

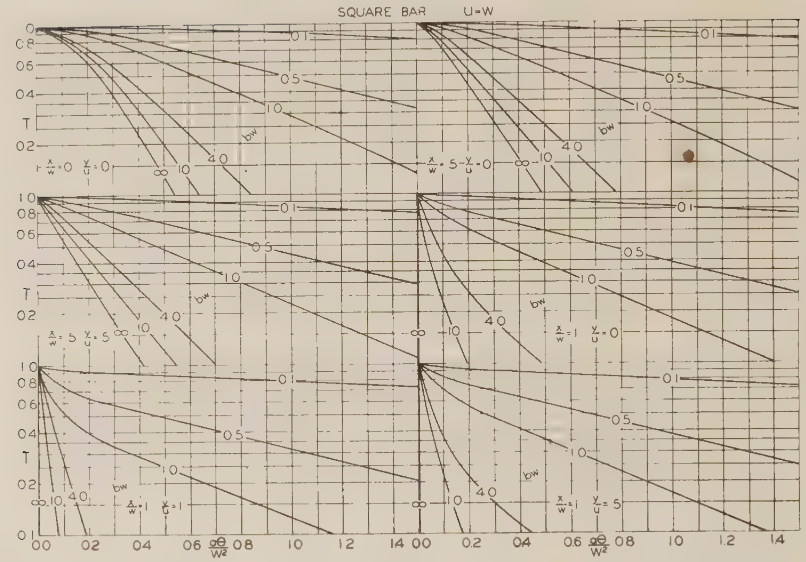

FIG. 11 COMBINED SOLUTION T, FOR POINTS IN INTERIOR OF AN INFINITE SQUARE BAR; ($u = w$)

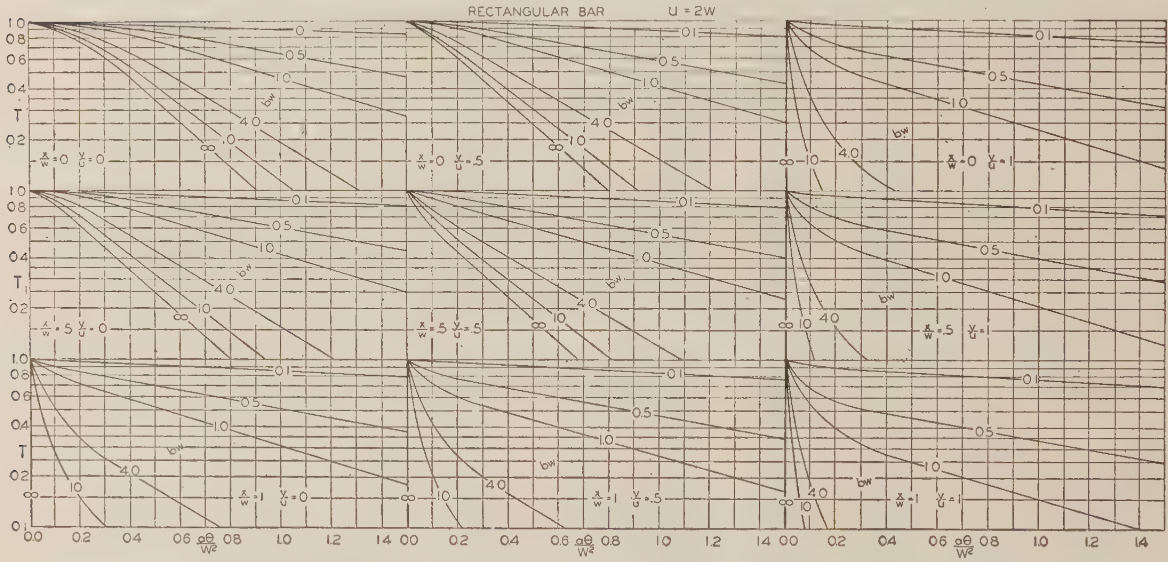

FIG. 12 COMBINED SOLUTION T, FOR POINTS IN INTERIOR OF AN INFINITE RECTANGULAR BAR; ($u = 2w$)

and for

$a\theta/u^2 = 0.2$ and $bu = 2.0$, $T_y = 0.80$ (estimated by interpolation).

Then

$$T = T_x \cdot T_y = 0.564 \times 0.80 = 0.451$$

This same result is obtained directly from curves for $x/w = 0.5$ and $y/u = 0.5$ in Fig. 12. For $a\theta/w^2 = 0.8$ and $bw = 1.0$, the value of T is seen to be 0.45, which is a close check.

CONCLUSIONS

Solutions to two-place accuracy may be obtained by interpolating in the curve sheets Figs. 4 to 8, inclusive, and Figs. 11 and 12, which were constructed from the values of T_x in Tables 1 to 5, inclusive.

More accurate solutions are obtained by solving Equation [3] directly. The first term of the series in Equation [3] is sufficient for three-place accuracy when $a\theta/w^2$ is greater than 0.3. The first two terms should be used for three-place accuracy when $0.1 < a\theta/w^2 < 0.3$.

ACKNOWLEDGMENT

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(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Logographs

By R. S. Tour[1]

A different type of chart[2] is presented for graphical solution of equations of the form $T^pU^qV^rW^s = K$. The charts are called "logographs" as distinct from nomographs and depend basically on the addition of logarithms. Logograph charts develop from the fact that a straight line drawn on semilog paper represents a function of the form: $\log y = a + bx$, or $x = c + n \log y$. The construction of logographs is explained and exemplified in the development of a chart for the calculation of heat-transfer coefficients in turbulent flow. As examples of more elaborate logographs, charts are also offered for the turbulent flow of fluids in pipes and for the flow of fluids through sharp-edged orifices. The equations underlying the last two charts represent new forms of the standard expressions and are briefly discussed and their limitations indicated.

To those who have constructed or used alignment charts involving four or more variables, the attendant difficulties are evident. The reference or blank scales that become part of such nomographs, the length of the scales for wide ranges in the values of the variables, the obliqueness of the intersecting lines that must sometimes be used with resultant inaccuracy, serve to multiply the difficulties. The construction of such alignment charts is also no simple matter since each of the lines requires a special scale laboriously laid out by hand.

These difficulties are particularly evident in nomographs for evaluating heat-transfer expressions wherein the variables may number seven, eight, or more, covering ranges for some of perhaps 10^6. In addition to special scales for each variable there will also be "dummy" scales (or their equivalent) numbering three less than the number of variables. The progressive series of alignments that must be made between the scales, taken in proper order, may make the use of the nomograph so tedious and confusing as to neutralize its value for rapid computations. Especially is this the case if a solution is desired for one of the variables other than that for which the nomograph was originally designed.

A fundamentally different type of chart eliminating most of the objectionable features of nomographs is described in this paper. It is applicable in general to functions expressible in the form

$$T^pU^qV^rW^s = K \qquad [1a]$$

$$p \log T + q \log U + r \log V + s \log W = \log K \qquad [1b]$$

The T, U, V, W terms represent variables, with p, q, r, s as constant exponents which may be either positive or negative, integers or fractions. The K is a constant, dependent on the function and on the dimensions ascribed to the individual variables. Expressions such as Equation [1a] arise frequently in engineering design and practice, especially in the fields of heat transfer and fluid flow. They lend themselves neatly to a graphical solution based on the addition of logarithms.

The principle involved in the computation charts offered here is based on the fact that the equation of any straight line drawn on semilog co-ordinate paper is of the form

$$\log y = a + bx$$

or

$$x = c + n \log y$$

The location of the line determines the constants c (or a) and the slope of the line defines n (or b). If each of the variables in Equation [1b] be plotted as a straight line on semilog paper with the c and n properly chosen, the linear (x) co-ordinates for any true set of values of the variables may be numerically added to accord with Equation [1b]. It is to be noted that a numerical addition is here substituted for the progressive series of graphical alignments used in nomographs. The process of addition is the quicker for such functions and is not subject to the manipulative errors of a series of alignments. But the chart used in this way is obviously not an alignment chart (i.e., nomograph); it is here descriptively titled a "logograph."

Logograph for Heat Transfer in Turbulent Flow

Recently a nomograph was published (1)[3] for the Dittus-Boelter equation for determining heat-transfer film coefficients with turbulent flow in smooth pipes. The equation used to determine these coefficients is of the Nusselt type and is generally given in any consistent units as

$$h = 0.0243 \frac{k}{D}\left(\frac{Du\rho}{Z}\right)^{0.8}\left(\frac{ZC}{k}\right)^{0.4} \qquad [2a]$$

This may be rewritten in an alternative form to eliminate fractional exponents

$$\frac{C^2k^3\rho^4u^4}{DZ^2h^5} = 1.18 \times 10^8 \qquad [2b]$$

In the usual mixed engineering units (specified) this becomes

$$\frac{C^2k^3\rho^4u^4}{DZ^2h^5} = 3.43 \times 10^{-3} \qquad [2c]$$

C = specific heat, Btu/(lb)(deg F)
k = thermal conductivity, Btu/(sq ft)(hr)(deg F/ft)
ρ = density, pcf
u = linear fluid velocity, fps
D = pipe diameter, in.
Z = viscosity, poises
h = heat-transfer coefficient, Btu/(sq ft)(hr)(deg F)

This equation with its seven variables, six independent, is an example of an expression for which a nomograph is not entirely satisfactory.

[1] Department of Chemical Engineering, University of Cincinnati.

[2] The charts (Figs. 1, 2, 3, 4) offered in this paper are simplified tracings from originals made on standard 11-in. × 17-in. three-cycle semilog paper. For convenience in size reduction and in printing, the tracings for this publication carry a minimum of co-ordinate lines. They are thus not satisfactory for computation purposes, but serve here merely to show the construction methods and use of logograph charts. If desired, the reader may transfer the lines shown to standard semi-log paper for more accurate computations. Full-size (11-in. × 17-in.) lithoprinted copies of the original charts are in use at the University of Cincinnati, and a limited number of these lithoprints will be available for distribution on request.

Contributed by the Heat Transfer Division and presented at the Cincinnati Section Meeting, Cincinnati, Ohio, Oct. 2–3, 1945, and at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

A logograph for making computations from Equation [2c] is shown in Fig. 1. The illustrative example given on the chart shows the general method of using logographs. It is the same example used by Ryant (1) for his alignment chart, wherein it is desired to find the heat-transfer coefficient (h) for hot water flowing at a velocity (u) of 8 fps through 1-in-diameter (D) tubes, the specific heat (C), conductivity (k), density (ρ), and viscosity (Z) of the water being specified. Each of the known variables is in turn located on the logarithmic scale (ordinates) and from the appropriate line for that variable, a reading R (abscissa) is determined. The six R's are numerically added and the sum subtracted from the chart constant A. The result is the R (abscissa) for the remaining variable, in this case the heat-transfer coefficient (h), which is then determined from its line on the chart. Six R's are read, an addition and subtraction made, and the unknown variable (which can be any one of the seven) is determined.

Construction of Logographs

To construct a logograph for some specified function, it is first necessary to rewrite the function in the form of Equation [1a]. Having done this, the following procedure is suggested:

1 Choose semilog paper with N logarithmic cycles to cover some selected range in the magnitude of the variables. This choice is flexible since the range for any variable can be increased from 10^N to 10^{2N} to 10^{3N} by the addition of successive constants to the reading for that variable (see Fig. 1). In general, three-cycle semilog paper is quite satisfactory.

2 Select dimensions of suitable magnitude for each of the variables in Equation [1] so that the same logarithmic scale may serve for all variables. This avoids confusion in the use of the scales. Then modify the value of the constant K in original Equation [1a] to agree with the dimensions which are selected.

3 Transform the modified Equation [1a] into the logarithmic form of Equation [1b]

$$(p \log T) + (q \log U) + (r \log V) + (s \log W) = \log K \quad [3]$$

Although as written here the signs are positive, it should be remembered that some of the exponents (p, q, r, s) may be inherently negative. In the general case, a negative sign can be applied throughout in Equation [3]. The use of the + or — is a matter of convenience; a negative sign throughout leads to a logograph which is a mirror image of that with a positive sign.

4 A scale factor or modulus M for the linear co-ordinates (abcissas) is selected so that the linear divisions will cover some desirable range in the value of the logarithms of the variables. The factor M is used as a multiplier in Equation [3].

$$(Mp \log T) + (Mq \log U) + (Mr \log V) + (Ms \log W) = M \log K \quad [4]$$

The modulus M may be positive or negative and its magnitude is a matter of judgment. Its value controls the slope of all the lines and the relative accuracy of the readings of ordinates versus abscissas. The size of M is limited by the ranges to be covered by the various lines on the chart.

5 Added convenience in using the chart is obtained if all the linear co-ordinates (abscissas) are positive. Accordingly, add some specific positive constant of sufficient magnitude (e, f, g, h) to each of the variable terms of Equation [4]

$$(e + Mp \log T) + (f + Mq \log U) + (g + Mr \log V) + (h + Ms \log W) = e + f + g + h + (M \log K) = A \quad [5]$$

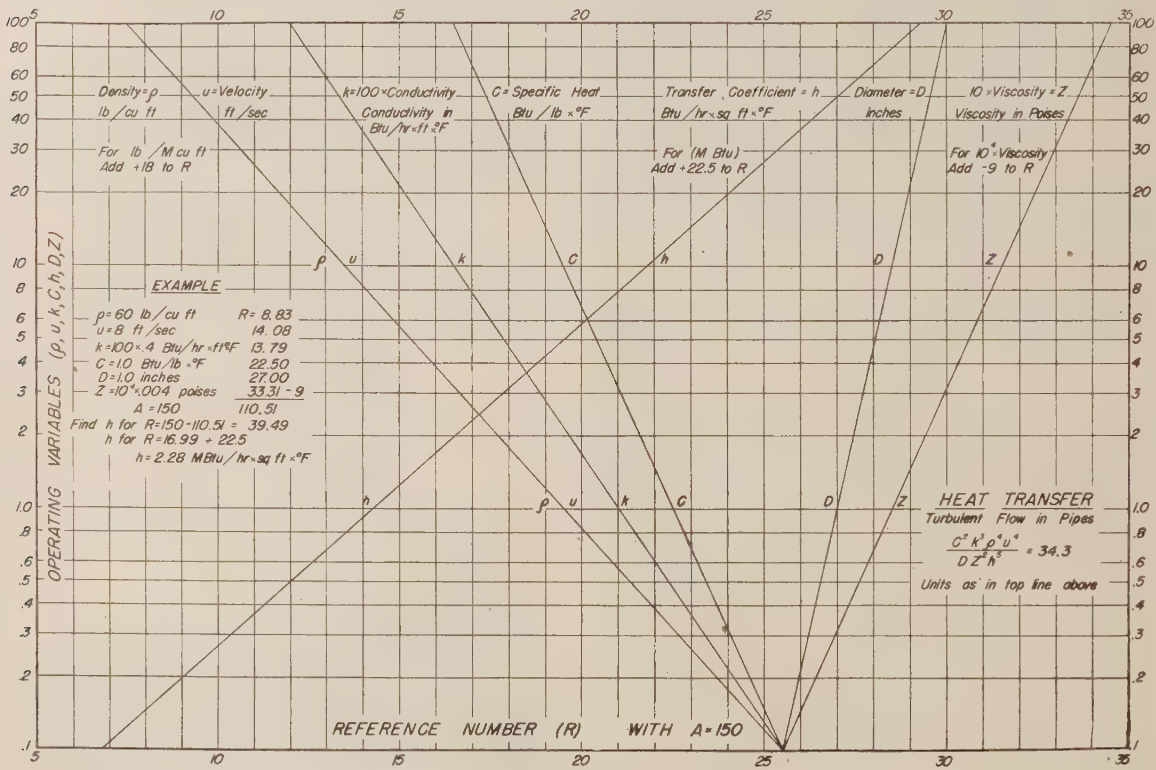

Fig. 1 Logograph for Heat Transfer

A wide choice exists for these addition constants. Their values control the relative locations of the lines on the chart. The constants should be selected so that the lines cross each other as seldom as possible and then not at too small an angle. The sum total A of the constants in Equation [5] should be some round number easily kept in mind, e.g., 150 (as in Fig. 1), and the value of A must be specified on the chart.

6 Each of the separated terms in parentheses in Equation [5] may now be plotted against the linear scale R, each plot resulting in a straight line on the semilog paper, all represented by the generalized expression

$$R_y = C + Mn \log Y$$

Here R_y refers to the abscissa readings for any one of the variables (T, U, V, or W) represented by Y, with n indicating its power (p, q, r, or s), and C the corresponding addition constant (e, f, g, or h). The lines for R_y may be located on the chart as follows: If the logarithmic scale ranges from 10^b at the bottom to 10^t at the top, then the intercepts at the bottom and top of the logograph for each of the variables Y are, respectively

$$R_y = C + (Mn)b \text{ (at bottom)}$$
$$= C + (Mn)t \text{ (at top)}$$

A line joining the bottom and top intercepts for each variable completes the chart.

7 If it is desired to increase the size of the units (or dimensions) of any one of the variables by a factor k, it is only necessary to add to reading R for that variable a quantity $+Mn \log k$. For example, a quantity expressed in inches on the chart may be read as feet by adding $+Mn \log 12 = +1.079Mn$ to the R_y obtained. Conversely, feet may be read as inches if the addition is $+Mn \log (1/12) = —1.079\, Mn$. The signs of M and n must not be neglected. A similar device may be used to change the range for any of the variables, which is equivalent to changing the size of the units for that variable by some power of 10.

8 Some expressions for which a logograph is desired may have one or more variables stated as a function of some more specific variable, e.g., $Y = F(y)$. In such cases the logograph is first designed for Y as described, but the straight line is replaced by a curve, the points for which are calculated so that a reading of y on the logarithmic scale calls for an R_y as determined by $Y = F(y)$, i.e., $R_y = M \log F(y)$.

Construction of Fig. 1. The foregoing directions, as applied specifically to the construction of the logograph in Fig. 1, may serve to clarify the procedure. The function for which the chart is to be designed has been given and the units specified under Equation [2c] as follows

$$\frac{C^2k^3\rho^4u^4}{DZ^2h^5} = 3.43 \times 10^{-3} \ldots\ldots\ldots\ldots\ldots [6]$$

The working range desired for each of the variables is

C: 10^{-1} to 10^1 = (10^{-1} to 10^1)
k: 10^{-3} to 10^0 = (10^{-1} to 10^2) $\times$ (10^{-2})
ρ: 10^{-4} to 10^2 = (10^{-1} to 10^2) $\times$ (10^{-3}) and (10^{-1} to 10^2)
u: 10^{-1} to 10^2 = (10^{-1} to 10^2)
D: 10^{-1} to 10^2 = (10^{-1} to 10^2)
Z: 10^{-5} to 10^1 = (10^{-1} to 10^2) $\times$ (10^{-4}) and (10^{-1} to 10^2) $\times$ (10^{-1})
h: 10^{-1} to 10^5 = (10^{-1} to 10^2) and (10^{-1} to 10^2) $\times$ (10^3)

Three-cycle ($N = 3$) semilog paper is suitable for these ranges, with double ranges required for ρ, Z, and h. For the log scale it seems best to take $b = —1$, $t = 2$, $N = t — b = 3$. To use the same ordinate readings (log scale) for all the variables it is necessary to express k as 100 times the conductivity (standard units) and Z as 10 times the viscosity (in poises), i.e., viscosity in decipoises. Further modifying Equation [6] to allow for this change in the units of k and Z

$$\frac{C^2k^3\rho^4u^4}{DZ^2h^5} = 34.3 \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

Expressing Equation [7] in logarithmic form and applying a modulus $M = —1.5$

$$—3 \log C — 4.5 \log k — 6 \log \rho — 6 \log u$$
$$+1.5 \log D + 3 \log Z + 7.5 \log h = —2.30$$

To hold the number of intersections of the lines of the logograph to a minimum (see Fig. 1), the following addition constants have been chosen

Variable	Term	Bottom intercept	Top intercept
C	$22.5 — 3 \log C$	$22.5 + 3.0 = 25.5$	$22.5 — 6.0 = 16.5$
k	$21.0 — 4.5 \log k$	$21.0 + 4.5 = 25.5$	$21.0 — 9.0 = 12.0$
ρ	$19.5 — 6 \log \rho$	$19.5 + 6.0 = 25.5$	$19.5 — 12.0 = 7.5$
u	$19.5 — 6 \log u$	$19.5 + 6.0 = 25.5$	$19.5 — 12.0 = 7.5$
D	$27.0 + 1.5 \log D$	$27.0 — 1.5 = 25.5$	$27.0 + 3.0 = 30.0$
Z	$28.5 + 3 \log Z$	$28.5 — 3.0 = 25.5$	$28.5 + 6.0 = 34.5$
h	$14.3 + 7.5 \log h$	$14.3 — 7.5 = 6.8$	$14.3 + 15.0 = 29.3$
Add	152.3	Chart constant, A, = 152.3 — 2.3 = 150	

It remains only to construct the logograph and indicate the units applicable to each line (see Fig. 1). The reason for choosing the apparently haphazard constants, such as 22.5, 21.0, etc., is that crossing of lines has been largely avoided with six of the lines having a common bottom intercept at 25.5. The sum, A, of the constants is 150; hence to evaluate any one variable in Equation [7] in terms of the other six

$$R_1 = 150 — (R_2 + R_3 + R_4 + R_5 + R_6 + R_7)$$

The extended (doubled) scales with the range increased by 10^3 in each case for ρ, Z, and h are calculated by adding $+Mn \log 10^3 = —1.5 \times 3n = —4.5n$ for enlarging the unit 10^3 times and adding $+4.5n$ for reducing the unit 10^3 times. Applying the appropriate values of n

For ρ: add $+4.5 \times (+4) = +18$ (smaller unit, factor = 10^{-3})
For Z: add $+4.4 \times (—2) = —9$ (smaller unit, factor = (10^{-3})
For h: add $—4.5 \times (—5) = +22.5$ (larger unit, factor = 10^3)

An example illustrating how the lines on a logograph may be moved relative to each other by changing the addition constants is shown in Fig. 2, a logograph for the same Equation [2c]. Here, $b = —1$ and $t = 2$ both as before, $M = —2$, and the sum of the constants, A, totals 100. The distribution of the constants is such as to give a chart entirely different in appearance and probably less desirable than that in Fig. 1.

Application of Logographs

Graphical solutions become desirable for application in engineering design in so far as they simultaneously offer the qualities of simplicity, speed, and accuracy. Charts should be versatile in the matter of range of the variables included and if possible in the units in which those variables are measured, and they should visually reveal the comparative effects of changes in magnitude of the individual variables. They should require a minimum of instruments or special facilities for their use. Some of these criteria of good charts may be incompatible and must therefore be foregone in special applications. Thus there are conditions under which the standard nomograph is most desirable, others for which the logograph is definitely called for, and still others where specialized diagrams are particularly suited.

The simplicity of the logograph chart for functions of the form

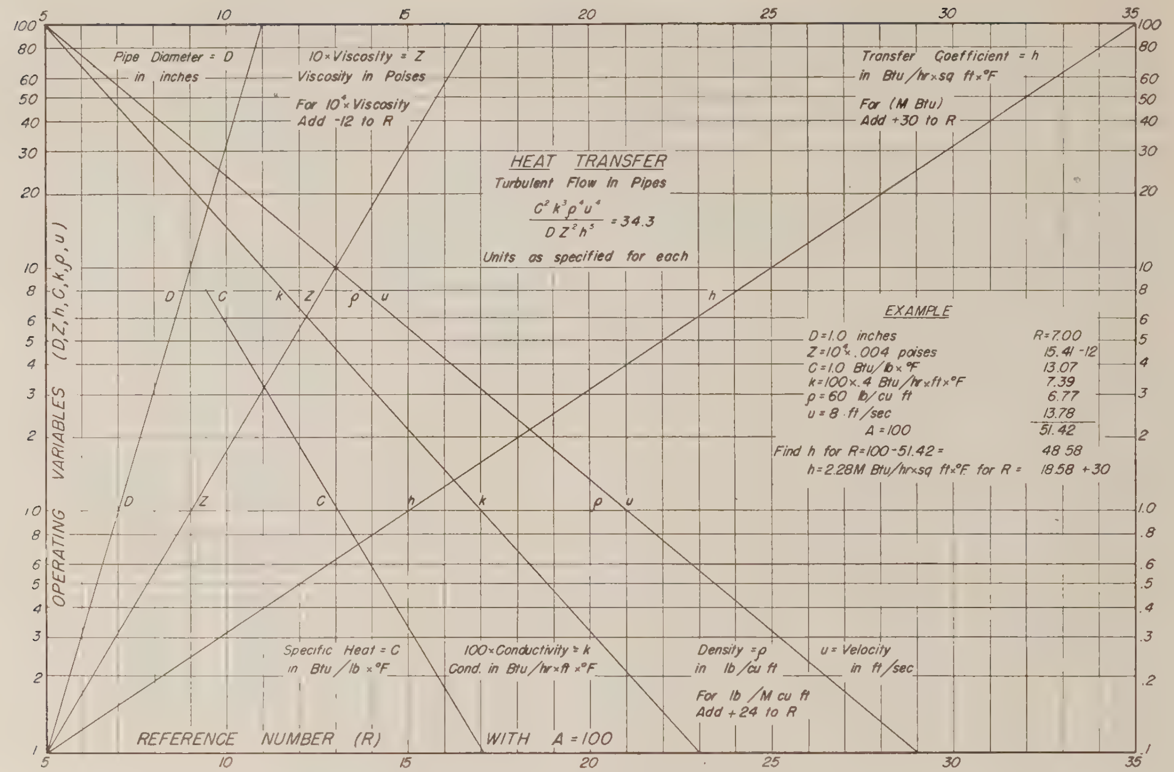

Fig. 2 Modified Logograph for Heat Transfer

of Equation [1*a*] is evident. Practically unlimited ranges (10^6 in Fig. 1) are possible for any of the variables, a change in the units for any one of the quantities may be obtained simply by the addition of a constant, and uniform accuracy is available for all the variables in any combination of ranges. The order in which the readings for the variables are taken from the chart is of no importance, and any one of the variables may be the unknown. In constructing the ordinary logograph, the plot for each variable is a straight line and no special scales need be drawn on or applied to any of the lines.

Various departures from the construction and use of logograph charts as here described may be found desirable or necessary in special applications. For example, instead of using logarithmic scaled paper, the lines of the chart may have their scales plotted directly on them. The location of the lines may become unimportant if they are considered merely as logarithmic vectors, with lengths taken off each of the lines from some point of reference, the various lengths then being added vectorially. Such a procedure may be accomplished wholly graphically by using a pair of triangles to lay off vectors parallel to those specified on the chart, and transposing the proper lengths with a pair of dividers. This method of addition is practical only when not too many vectors are to be so added and when the ranges for the individual vectors are comparatively small. It may be used to accomplish results not possible by numerical addition, e.g., in the Eiffel diagrams for airplane-propeller design (2). Eiffel charts have their scaled vectors so sloped that the addition of abscissa components develops one function of the variables while the ordinate addition determines another, the interrelation of the two functions being defined by some charted line intersected by the graphical vectorial addition.

Special purpose diagrams notwithstanding, numerical addition logographs plotted on semilog paper, as here described, will best serve for repetitive computations of functions of the form of $T^p U^q V^r W^s = K$, which so often arise in engineering design. Logographs cannot, however, replace nomographs where the equation to be solved involves an addition operation as well as a multiplication.

Two more logograph charts are offered in this paper not only as examples of somewhat more complicated logograph construction with appropriate explanation, but also for their intrinsic value in the calculation of industrial problems. For each of the two subsequent logographs, "turbulent flow of fluids in pipes," Fig. 3, and "fluid flow through orifices," Fig. 4, the basic equations are given and their source or development explained.

Turbulent Flow in Pipes

Although Equation [2] involves seven variables, it represents in form a simple type of logograph. One involving a curve develops from Fanning's equation for turbulent flow of fluids in pipes. Before presenting the logograph, a brief explanation is given of the underlying equation since it has not appeared in the literature in the form used here.

The variables involved in turbulent flow in pipes with units modified where necessary to suit the range of a three-cycle logograph are as follows:

Q = mass flow rate, M lb per min

P_1, P_2 = initial and final pressures, psia

ρ = density of the fluid, pcf

D = diameter of pipe, in.

f = friction coefficient $\approx 0.00759(DZ/Q)^{0.2}$

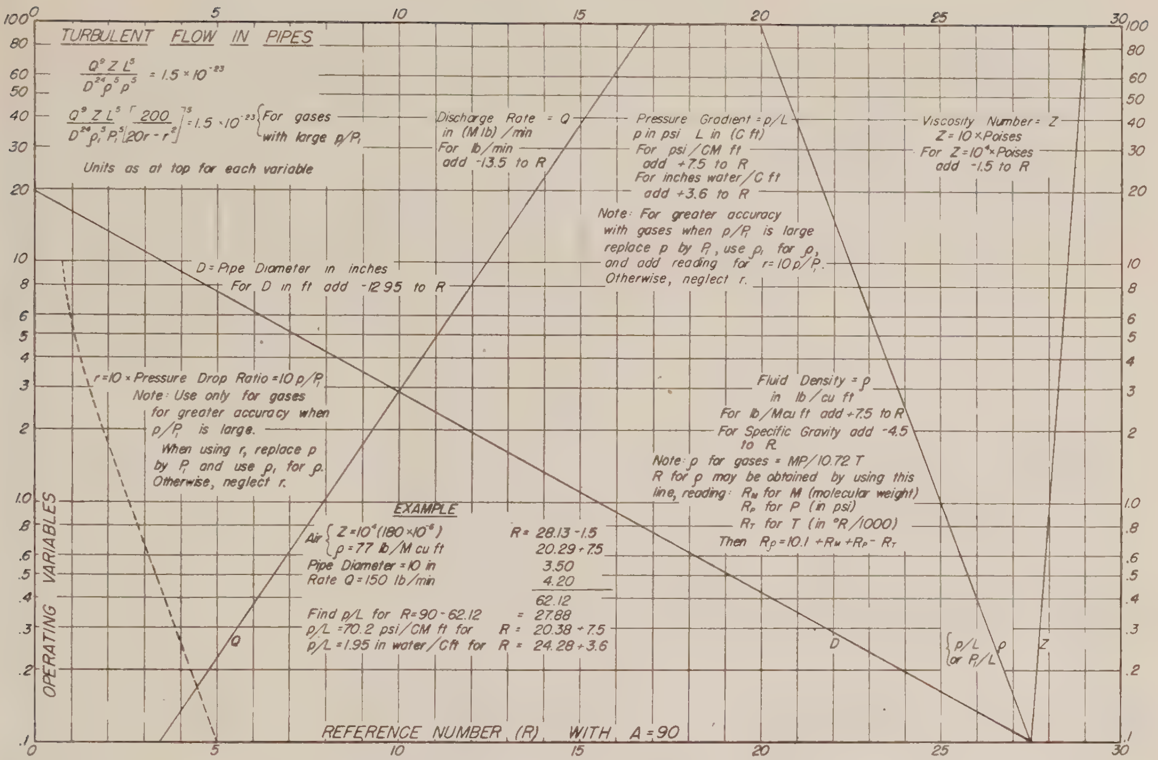

FIG. 3 LOGOGRAPH FOR TURBULENT FLOW IN PIPES

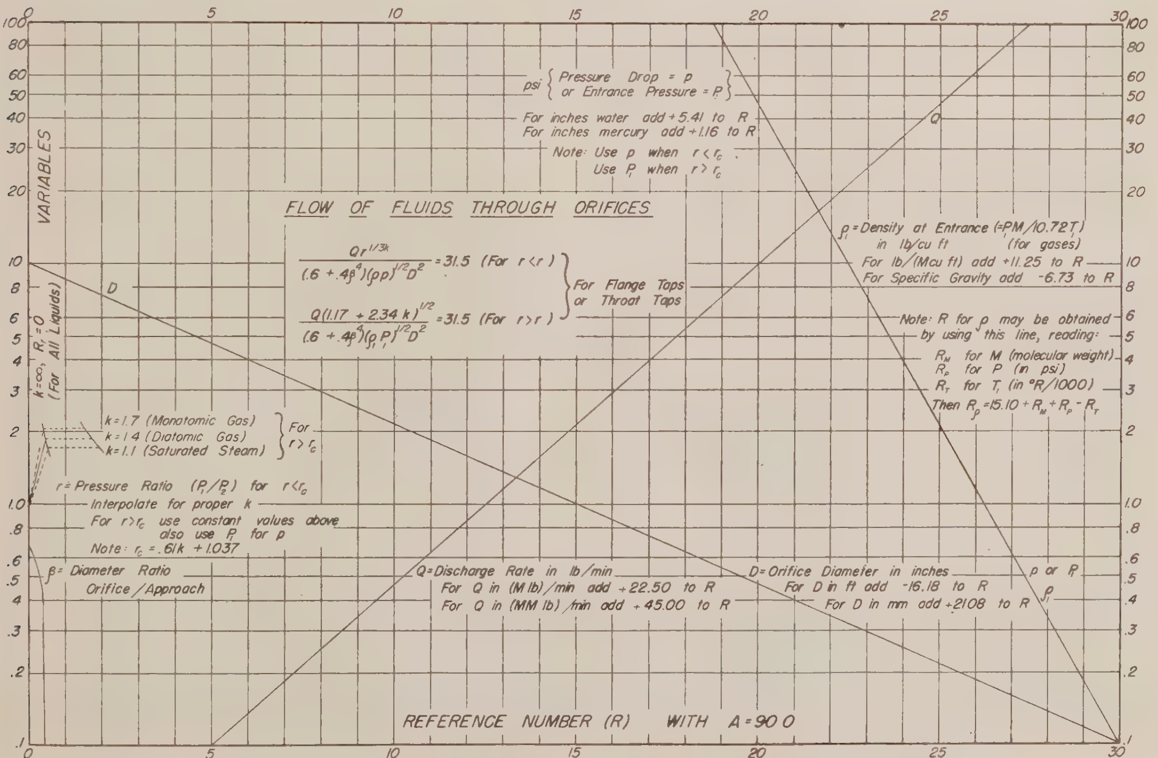

FIG. 4 LOGOGRAPH FOR FLOW OF FLUIDS THROUGH ORIFICES

L = length of pipe, ft in units of 100
p = pressure drop, psi
p/L = pressure gradient in the pipe line, psi/Cft
r = pressure drop ratio = $10p/P_1$
Z = viscosity of the fluid, decipoises
N = Reynolds number = 37,900 Q/DZ

In the foregoing units, assuming isothermal expansion and neglecting a logarithmic term which arises, Fanning's equation for turbulent flow leads to

$$Q = 3.215 \times 10^{-4} \sqrt{\frac{(P_1^2 - P_2^2)\rho_1 D^5}{fLP_1}}$$

This may be expressed by

$$\frac{fQ^2}{\rho D^5}\left(\frac{L}{p}\right) = 2.067 \times 10^{-7} \left\{\begin{matrix}\text{For all liquid flow or for} \\ \text{gases with small } p/L\end{matrix}\right\}$$

$$\frac{fQ^2}{\rho_1 D^5}\left(\frac{L}{P_1}\right)\left(\frac{200}{20r - r^2}\right) = 2.067 \times 10^{-7} \left\{\begin{matrix}\text{for gas flow} \\ \text{with large } p/L\end{matrix}\right\}$$

The coefficient of friction f is now expressed according to $1/f = 16(N)^{0.2} = 131.8\,(Q/DZ)^{0.2}$. The use of a constant exponent on N in any expression for f is not acceptable for smooth pipe or for pipe with specified roughness, but is nevertheless supported experimentally for industrial piping with its relative roughness decreasing (though its absolute roughness increases) for the larger pipe diameters. Inserting this expression for $1/f$ leads to

$$\frac{Q^9 Z}{\rho^5 D^{24}}\left(\frac{L}{p}\right)^5 = 1.5 \times 10^{-23} \left\{\begin{matrix}\text{for all liquid flow or for} \\ \text{gases with small } p/L\end{matrix}\right\} \quad \text{[8a]}$$

$$\frac{Q^9 Z}{\rho_1^5 D^{24}}\left(\frac{L}{P_1}\right)^5\left(\frac{200}{20r - r^2}\right)^5 = 1.5 \times 10^{-23} \left\{\begin{matrix}\text{gas flow with} \\ \text{large } p/L\end{matrix}\right\} \quad \text{[8b]}$$

Equations [8] are quite satisfactory for industrial piping up to Reynolds numbers of several million and offer an accuracy within a few per cent for Q. Greater precision is not required in ordinary practice since corrections for variations in original roughness of the pipe, for the effect of scaling or corrosion, and for the equivalent length of installed fittings are at best but approximate.

It must be remembered that Fanning's equation is in itself inaccurate in that it assumes isothermal gas expansion and also omits a logarithmic term. The equation may show considerable error for gas flow with large proportionate pressure drops and small frictional resistance. The inaccuracy increases nearly directly with pD/fLP_m where P_m is the logarithmic mean of P_1 and P_2; it becomes objectionable (depending on the accuracy desired) when the pressure gradient p/L exceeds some 100 times fP_m/D for gases, the error in Q then being 2 per cent (with units for p, L, P, D as previously specified). Fortunately this situation seldom arises in industrial practice.

A logograph for Equations [8] (a or b) is given in Fig. 3. The [8a] portion of the equation is a simple logograph covering five variables. The [8b] portion, used when greater accuracy is desired for gases and vapors that expand appreciably during the flow, substitutes P_1 for p in p/L with ρ taken as ρ_1 (compare Equation [8a] with [8b]) and adds a curve for the function of $r(=10p/P_1)$. This chart illustrates the use of a curved line.

An innovation, included in Fig. 3, further indicates the versatility of the logograph: R for ρ of the gas can be determined graphically from its molecular weight, pressure, and temperature if the gas is considered to be ideal. This auxiliary use of any straight line of a logograph is a direct consequence of the fact that for each variable, the line on the chart reads a value of $R_y = C + Mn \log Y$. The line can thus be used as a slide rule for calculation by logarithms.

Some sample computations from Fig. 3 may not be amiss. Air, for which the viscosity is 180×10^{-6} poises, is to be transmitted through a 10-in-diam flue at nearly atmospheric conditions (60 F, 30 in. Hg) with ρ = 77 lb per 1000 cu ft. For a flow of 150 lb per min (1950 cfm), the pressure gradient is desired. The solution for this problem is outlined in Fig. 3 and need not be repeated here. The p/L in this case is small and no consideration need be given to the expansion ($r = 10p/P_1$) during the flow.

Another example will show the use of r. Here high-pressure air (60 F, P_1 = 100 psia) is being delivered through a 2-in. pipe at a rate of 150 lb per min. The pressure gradient is desired. The solution is shown both neglecting r and including r. If desired, the $\rho_1(=0.524$ pcf) can be obtained from the chart with $R\rho = 10.07 + 21.34 + 20.0 - 25.71 = 25.70$.

	Neglecting r	*Including r*
Air { $Z = 10^4(180 \times 10^{-6})$ R =	28.13 — 1.5	28.13 — 1.5
Air { $\rho_1 = 0.524$ pcf	25.70	25.70
Pipe diameter = 2 in.	11.89	11.89
Rate = 0.15 M lb per min	4.20	4.20
P_1/L = 100 (for 100 ft pipe)		20.0
	68.42	88.42
Find p/L		
for R = 90 — 68.42 =	21.58	
p/L without r correction =	23.3 psi/Cft	
Find $r = 10p/P_1$		
for R = 90 — 88.42 =		1.58
r = 2.70 or p/L corrected =		27.0 psi/Cft

Fluid Flow Through Orifices

An example of a more elaborate logograph is shown in Fig. 4. Again the equation applying is first discussed since it has not been given previously. The basic equation for the flow of fluids through a standard sharp-edged thin-plate orifice is given in hydraulic form by

$$Q = 31.5D^2KY\sqrt{\rho_1 p} \quad \text{[9]}$$

where

Q = mass discharge rate, lb per min
D = diameter of orifice, in.
K = orifice coefficient (approach factor included)
Y = expansion factor for the gas
ρ_1 = entrance density, pcf
p = pressure drop, psi
P_1, P_2 = initial and final pressures, psia
β = diameter ratio; orifice/approach
r = pressure ratio: $P_1/P_2 > 1.0$
k = specific-heat ratio

A series of experiments conducted by the U. S. Bureau of Standards (3) has determined KY in the hydraulic form of flow, Equation [9], for β up to 0.6, and for expansion ratios r from 1.0 to critical r_c (r_c corresponds to sonic velocities in the fluid stream). The experimental data are very well correlated (to 1 per cent) by $KY = (0.6 + 0.4\beta^4)r^{-1/3k}$ for flange-tap or throat-tap pressure readings. This expression is sufficiently accurate for the purposes of this paper and will serve for liquids by taking $k = \infty$.

More recent and elaborate experimentation (4) has indicated that the coefficient is also a function of the actual pipe diameter and of Reynolds number, but this refinement is unnecessary here.

The simple expression for KY is substituted in Equation [9] for flows at pressure drops below critical r_c. For flows with drop greater than indicated by r_c, the effective r equals r_c, the KY for the orifice becomes constant, and the effective pressure drop p is $P_1(r_c - 1)/r_c$. Thus

$$\frac{Qr^{1/3k}}{(0.6 + 0.4\beta^4)D^2(\rho_1 p)^{1/2}} = 31.5 \text{ (for gases with } r < r_c) \ldots\ldots \quad [10a]$$

Take $k = \infty$ and $r_c = \infty$ in Equation [10a] $\left\{\begin{array}{l}\text{for liquids at}\\ \text{all values of } r\end{array}\right\}$

$$\frac{Qr_c^{1/3k}}{(0.6 + 0.4\beta^4)D^2(\rho_1 P_1)^{1/2}}\left(\frac{r_c}{r_c - 1}\right)^{1/2} = 31.5 \left\{\begin{array}{l}\text{for gases with}\\ r > r_c\end{array}\right\} \ldots\ldots\ldots \quad [10b]$$

The development of the logograph for Equation [10a] gives the usual straight lines for Q, D, ρ_1, and p. A curve is calculated for the term $(0.6 + 0.4\beta^4)$ up to $\beta = 2/3$; measurements are undependable beyond that value.

The principal limitation of the logograph arises in plotting the term $r^{1/3k}$. As in nomographs, functions with mixed variables cannot be represented except by a family of scales or lines. In Fig. 4 lines for r (up to r_c) are given, respectively, for $k = \infty$ (all liquids), and for $k = 1.7, 1.4, 1.1$, allowing for interpolation of k between these values. It is evident from the chart that the effect of the value of k for gases is small and a rough interpolation is satisfactory. No addition constant is included in the logarithm of the r function, hence for liquids with $k = \infty$, the R_r is zero, i.e., r and k are both neglected in using the chart for liquids.

The logograph for Equation [10b] is the same as for [10a] except that p is replaced by P_1 and the function of r_c becomes a constant depending only on k for the particular gas involved. It can be shown that

$$r_c = \left(\frac{k+1}{2}\right)^{\frac{k}{k-1}}$$

Although calculations for the logograph may be made directly from the foregoing expression for r_c, simpler empirical relations are suggested for extended calculations as follows

$$r_c \approx 0.61k + 1.037 \text{ (to 0.1 per cent)}$$

$$r_c^{1/3k}\left(\frac{r_c}{r_c - 1}\right)^{1/2} \approx \sqrt{1.17 + \frac{2.34}{k}} \text{ (to 0.1 per cent)}$$

In using Equation [10b] or the chart for gas flow with r far into the supercritical range, it must be remembered that the orifice calibration has not been experimentally determined in that region. The assumption that KY is constant for $r > r_c$ seems logical but may lead to some inaccuracy if carried too far. Orifices should not be used under conditions where r is much larger than r_c.

The logographs for Equation [10a] and [10b] may be combined into one if proper directions are given for its use. In this way Fig. 4 covers fluid flow through orifices for all cases and conditions: Liquids or gases for their complete range from $r = 1.0$ to $r = r_c$, and for $r > r_c$.

No example appears in Fig. 4, but two are given as follows: The simplest case is for liquid flow where $r (= P_1/P_2)$ is not involved. A sharp-edged orifice of 1 in. diam is being used in a 2-in. pipe line ($\beta = 0.5$) with water ($\rho = 62.4$ pcf). The pressure drop (p) is 10 in. Hg. Find rate of discharge (Q).

Orifice	$D = 1$ in.	$R = 15.00$
	$\beta = 0.5$	0.29
Water:	$\rho = 62.4$ pcf	19.41
Drop:	$p = 10$ in. Hg	22.50 + 1.16
		58.36
Find Q for $R = 90 - 58.36 =$		31.64
$Q = 360$ lb per min for $R =$		9.14 + 22.50

Using the same orifice as before for air ($k = 1.4$) with $P_1 = 100$ psia at 60 F ($\rho_1 = 0.524$ pcf) and $P_2 = 80$ psia in Case 1 (for $r < r_c$) or $P_2 < 53$ psia in Case 2 (for $r > r_c$), find the rate of discharge (Q).

		$r < r_c$	$r > r_c$
Orifice	$D = 1$ in.	$R = 15.00$	15.00
	$\beta = 0.5$	0.29	0.29
Air:	$\rho_1 = 0.524$ pcf	27.30	27.30
	$p = 20$ psi	21.37	
	$r = P_1/P_2 = 1.25$	0.17	
	$P_1 = 100$ psia		18.75
	$r > r_c$ for $k = 1.4$		1.70
		64.13	63.04
Find Q for $R = 90 - 64.13 =$		25.87	
	Q for P_2 at 80 psia =	60.6 lb per min	
Find Q for $R = 90 - 63.04 =$			26.96
	Q for $P_2 < 53$ psia =		84.7 lb per min

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(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Some Effects of Work Positioning When Face-Milling Steel

By FRED W. LUCHT,[1] DETROIT, MICH.

All of the work that has been done and all of the observations which have been made to date on the subject of face-milling steel with carbide cutters have indicated clearly that the longest cutter life is obtained when (1) the cutters have double negative rake angles, and (2) the full depth of cut is taken along some definite corner angle. Until recently, not much thought had been given to the positioning of the work in relation to the cutter, and only in limited cases has any consideration been given to the fact that the radial engagement angle between the face of the tooth and the edge of the work where the cutting edge enters it should be negative. Milling operations are frequently found where the diameter of the face mill is too small to permit the proper positioning of the work to a setting which gives longest cutter life. Under such a condition, there seems to be no definite means available which will assist in determining the diameter of a face mill capable of producing the maximum number of pieces per grind. This problem of determining optimum cutter diameter for a given face-milling job, and the proper positioning of the workpiece in relation to the cutter, led to a series of studies, the results of which are reported in this paper.

Introduction

All of the work that has been done and all of the observations which have been made to date on the subject of face-milling steel with carbide cutters have indicated clearly that the longest cutter life is obtained when (1) the cutters have double negative rake angles, and (2) the full depth of cut is taken along some definite corner angle.

There are certain operating conditions in every milling-machine setup which should at all times be maintained as nearly ideal as possible. These objectives are design, manufacture, and sharpening of the cutter; power capacity, feed and speed of the machine; and a flywheel to smooth out the roughness obtained from the necessarily coarser-tooth cutters removing stock at increased table travel. But even after all this has been done, the cutter performance and the number of pieces obtained per cutter grind still will show noticeable variations as the work position is shifted at right angles to the direction of feed of the work past the cutter.

Until recently, not much thought had been given to the positioning of the work in relation to the cutter, and only in limited cases has any consideration been given to the fact that the radial engagement angle D, Fig. 1, between the face of the tooth and the edge of the work where the cutting edge enters it, should be negative. It is interesting to note that when this angle has been negative, the number of pieces per grind has at least been more consistent than when no attention at all was paid to it, even though the cutter may not have produced its maximum number of workpieces per grind.

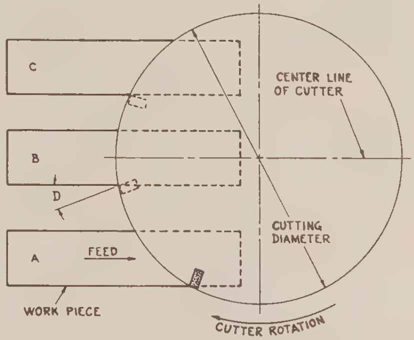


Fig. 1 Work Position Influences Face-Mill Life

It has also been found that a face-milling cutter usually gives a smooth cutting action together with the maximum number of pieces per grind when the workpiece is positioned about as shown at B. Position C generally rates second as to the number of pieces per grind, with quite consistent results. Position A rates third, with somewhat erratic results. This difference in results has been noticed for years while observing the face-milling of steel with high-speed steel cutters, but it has become more pronounced when milling steel with carbides.

Milling operations are frequently found where the diameter of the face mill is too small to permit the proper positioning of the work to a setting which gives longest cutter life. Under such a condition, there seems to be no definite means available which will assist in determining the diameter of a face mill capable of producing the maximum number of pieces per grind.

This problem of determining optimum cutter diameter for a given face-milling job, and the proper positioning of the workpiece in relation to the cutter, resulted in the investigation to be discussed.

Details of Test Procedure

Milling Machine. A No. 5-HM vertical milling machine in the engineering department of the author's company was used.

Power available:	
Main spindle, hp	20
Feed and rapid traverse, hp	5
Wattmeter:	
Main spindle, kw	30
Feed and rapid traverse, kw	12

Face-Milling Cutter. An 8-in-diam single-tooth face-milling cutter, enshrouded with an 18-in-diam × 4-in-thick steel flywheel, was used, Fig. 2. The tool-bit slot is parallel to the axis and the tool bits are clamped in place by setscrews. The combined weight of the cutter head flywheel tool unit was 255 lb.

Tool Bit. A total of twelve tool bits, as shown in Fig. 3, was used for this test. These tools were tipped with Carboloy Grade 78B, a general steel-milling grade.

[1] Development Engineer, Carboloy Company, Inc. Mem. A.S.M.E.

Contributed by the Research Committee on Metal Cutting and Bibliography and presented at the Fall Meeting of the Cincinnati Section, Cincinnati, Ohio, Oct. 2–3, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

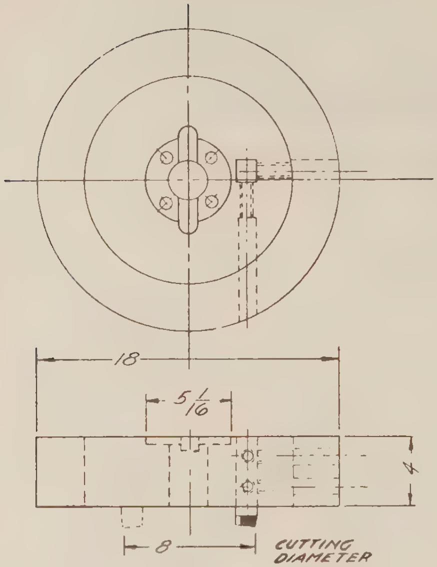


Fig. 2 To Eliminate Vibration, Cutter Was Enshrouded in a Flywheel

Both carbide and steel on all tools were rough-ground on a single-point tool grinder using a 60-grit silicon-carbide wheel. A 220-grit resinoid-bond diamond wheel was used for the final sharpening of the carbide only.

The cutting edges on all tools were inspected at 20 magnifications. Each edge along the corner angle and the chamfer was

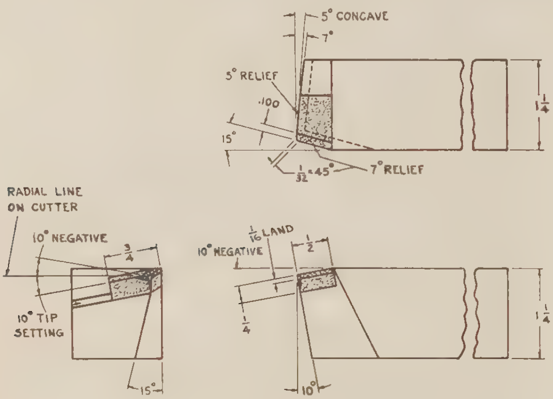


Fig. 3 All Tool Bits Used Were Ground to Same Cutting Angles

brushed lightly with a 320-grit silicon-carbide stone at an angle of about 45 deg to the face of the tool. This hand-honing removed any slight irregularities before the tool was assembled in the cutter head.

Positioning Tool in Cutter Head. Fig. 4 shows the method used to position the tool bit accurately in the tool-bit slot in the cutter head. Two sets of shims were required; one set, 0.100 to 0.200 in. thick, was used to maintain the 4-in. radius which gave the 8-in. cutting diameter; the second set, 0.180 to 0.310 in. thick, was used to maintain the 10-deg negative radial rake. This arrangement eliminated holding the corner location on the tool to close limits and facilitated correction for the stock which was

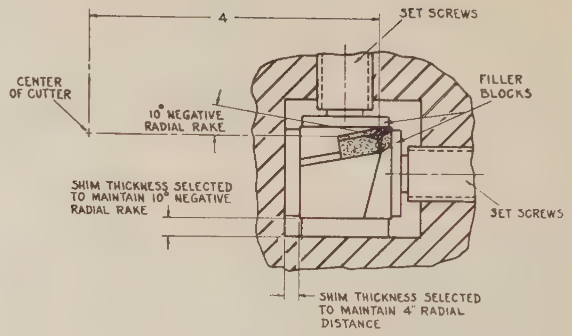


Fig. 4 All Tool Bits Were Accurately Positioned in Cutter Head by Aid of Shims

ground from the face of the tooth, the corner angle (side cutting-edge angle on the tool), and the face of the cutter (end cutting-edge angle on the tool) every time the tool bit was sharpened.

Work. For all face-milling runs, S.A.E.-1045 forged-steel billets were used. These billets were $1^1/_2 \times 3 \times 12$ in. in size. They were heat-treated and drawn to a Brinell hardness of 190 to 200.

Machine Setup. Fig. 5 shows the machine setup that was used while making all milling runs. In this setup the steel billet, or workpiece, was held rigidly in a plain vise in relation to the cutter, as shown.

Speed. The 8-in. face mill was operated at 238 rpm (498 sfpm) because previous runs had indicated that this speed gave the longest cutter life when milling S.A.E. 1045 steel having a Brinell hardness of 190 to 200.

Feed. The following feeds were used to permit exploration of a wide range of face-milling activity:

Table travel, ipm	Feed per tooth, in.
2	0.0084
$2^1/_2$	0.0105
3	0.0126
$3^1/_2$	0.0147

Depth of Cut. A depth of cut of 0.150 in. was taken on all runs. This is an average depth of cut which shows wear similar to that caused by deeper cuts, yet which does not involve the machining of excessive amounts of steel as would the deeper cuts.

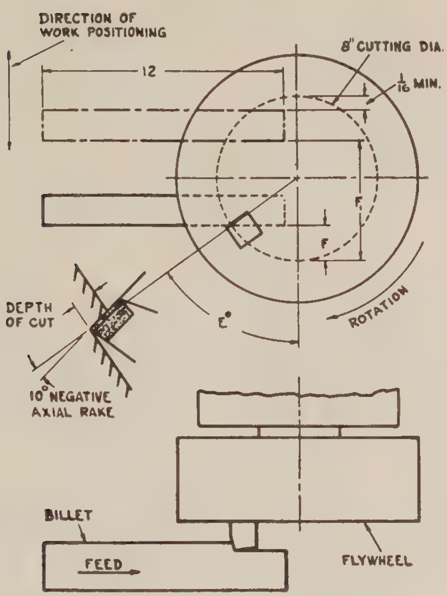
Operating Detail and Observations. In order to determine the effect of work positioning in relation to the face mill while all other conditions were kept constant, as has been outlined, the following runs were made with the work:

1 Shifted horizontally.
2 Shifted in a direction perpendicular to the table travel.
3 Shifted to each one of the positions listed in Table 1.

The "angular position E" represents the position where the cutting edge on the 8-in-diam tool first contacts the work.

The dimension F gives the distance from the point at which the cutting edge on the 8-in-diam tool first contacts the work, to the 8 in. diam measured perpendicularly to the direction of feed. During the milling runs, the work was shifted through a series of positions as listed from one extreme position where E equaled 10 deg and F was 0.061, to the other extreme position where E equaled 127 deg 33 min and F was 6.437. At each of these two extreme positions, the outside diameter of the face mill overlapped the workpiece $^1/_{16}$ in.

Each tool was run until the cutting edge showed the first sign of breakout. This was detected either from the marking on the inner side of the chip which came from the cut; from the lines left on the work surface by the corner angle; or from the change



WORK POSITION	
ANGULAR POSITION E (Degrees)	DISTANCE F (Inches)
10	.061
20	.241
30	.536
35	.723
40	.936
45	1.169
50	1.429
55	1.706
60	2.000
65	2.310
70	2.632
75	2.965
80	3.305
90	4.000
95	4.349
100	4.695
105	5.035
110	5.368
120	6.000
127°-33′	6.437

Fig. 5 Cutter Setup and Table Showing Work Positions Used

TABLE 1 DIMENSIONS USED FOR POSITIONING WORK IN RELATION TO FACE MILL

Angular position E, deg	Distance F, in.	Angular position E, deg	Distance F, in.
10	0.061	70	2.632
20	0.241	75	2.965
30	0.536	80	3.305
35	0.723	90	4.000
40	0.936	95	4.349
45	1.169	100	4.695
50	1.429	105	5.035
55	1.706	110	5.368
60	2.000	120	6.000
65	2.310	127–33[a]	6.437

[a] Reported in deg-min.

in sound made by the milling operation. The distance traveled by the cutter, and the power readings, were recorded for analysis.

The cutting edge was inspected under a 20-power microscope and a record made in the form of an enlarged (to scale) freehand sketch showing the development of the cutting edge along the corner angle, the chamfer, and the face of the cutter. This sketch showed the width of the wear land behind the cutting edge along each of these three edges. It also showed the distance the cutting edge had worn back from the original corner-angle outline, and the width of the crater on the face of the tooth behind the cutting edge. All unusual signs of breakdown of the cutting edge were recorded on this sketch to serve as an aid in determining why the tool failed.

Using the twenty different work positions and the four different feed rates, a total of 190 runs was distributed as evenly as possible over the 12 tools. This gave 16 runs for each of 10 tools and 15 runs for each of the remaining two tools.

An average of the distances traveled up until the time of tool failure was plotted, Fig. 6, with its work position for each feed used. An average of all the runs made, irrespective of the feeds used for each work position, was also plotted.

The runs were made first at 2 ipm feed, i.e., 0.0084 in. feed per tooth. It was soon found that the various points arranged themselves in an irregular outline and also that the length of runs seemed to be somewhat erratic. Later, many additional runs were made at 0.0105, 0.0126, and 0.0147 in. feed per tooth, and it was found that not only did each set of runs follow the same general pattern but also that the lengths of runs were much more consistent than were those for the 0.0084-in. feed per tooth.

Analysis of How Tooth Contact Affects Cutter Life. To assist in determining why all the curves followed a definite general pattern, an enlarged wooden model, as shown in Fig. 7, was made of the cutter-work setup.

The model is shown in outline in Fig. 8 and is 10 times actual size. The base of the model is divided into angular spacing about a pivot point C. The block D, representing an exact 10-times enlargement of the cutting portion of the tool, is mounted on the end of the arm so that the intersection of the 15-deg corner angle and the 5-deg concavity angle is at a 40-in. radius.

Ten work blocks were made, as shown in Fig. 9, to the forms given in Table 2. This represents the exact contour cut by the cutter tooth when the entrance portion of the work is located at each one of the respective angular work positions.

To determine the type of contact for any given angular position of the cutter tooth, the work block for that angular position was placed against the guide block and slid past the 40-in. radius for a distance comparable to the feed per tooth, i.e., 0.105 in. for a 0.0105-in. feed per tooth. The cutter tooth was then swung in the direction shown until it contacted the workpiece. The type of tooth contact was observed. This procedure was repeated for each angular position. The results were compiled and the proper sequence of tooth contact for each position was recorded.

Figs. 10 to 18, inclusive, are close-up views of the various possible types of initial contacts between the cutter tooth and the

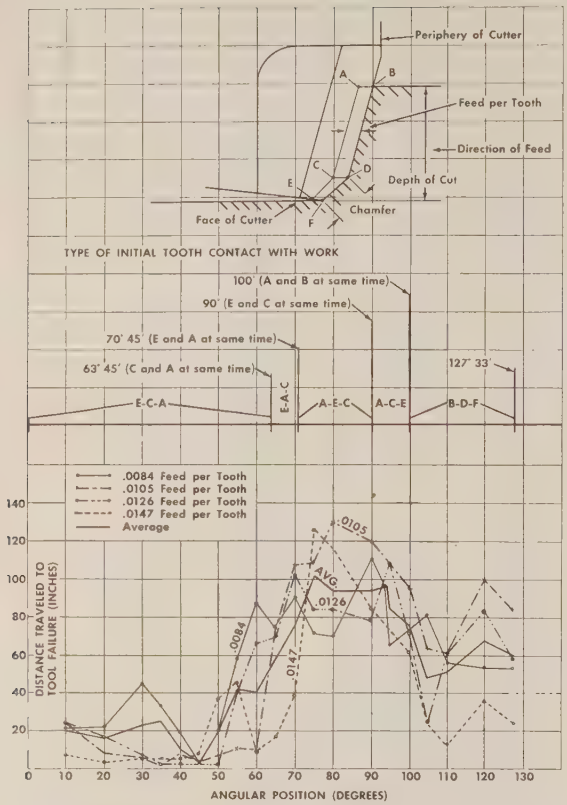


Fig. 6 Cutter-Life Curves for Various Work Positions and Feeds per Tooth

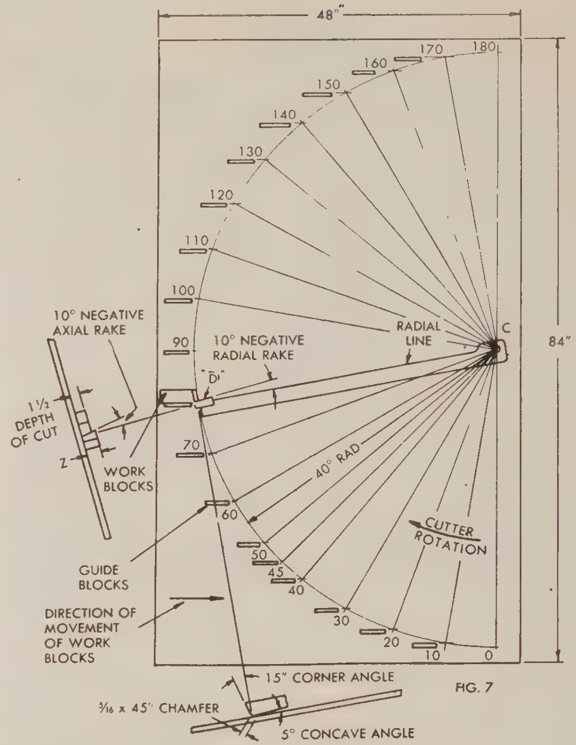


Fig. 8 Detail Dimensions of Wooden Model

Fig. 7 Enlarged Wooden Model Showing Tooth Approaching a Work Block

work. All of the original photographs were taken looking outward from the center of the cutter in a direction along the 10-deg negative radial rake angle on the face of the tooth.

Fig. 10 covers the conditions which exist when the point of initial tooth contact with the work is at E on the face of the tooth where it intersects the face of the cutter and is at a distance from the cutting edge on the chamfer equal to the feed per tooth. This is followed by point C, and finally by point A, contacting the work. This occurs when the tooth enters the cut anywhere from the zero position up to the 63-deg 45-min position.

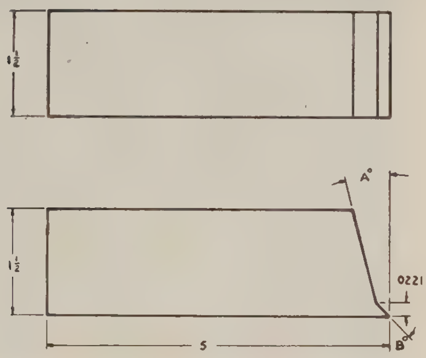


Fig. 9 Work Blocks Used While Determining Type of Tooth Contact

TABLE II

ANGULAR POSITION	ANGLE ON EDGE OF WORK A	B
10-170	53°-32′	80°-36′
20-160	37°-06′	71°-18′
30-150	27°-50′	63°-32′
40-140	22°-30′	57°-19′
45-135	20°-39′	54°-46′
50-130	19°-13′	52°-35′
60-120	17°-09′	49°-07′
70-110	15°-55′	46°-47′
80-100	15°-13′	45°-27′
90	15°-00′	45°-00′

TABLE 2 OUTLINE OF FORM MILLED ON ENTRANCE EDGE OF WORK FOR EACH WORK POSITION USED

Angular position, deg	—Angle on edge of work, deg-min— A	B
10–170	53–32	80–36
20–160	37–06	71–18
30–150	27–50	63–32
40–140	22–30	57–19
45–135	20–39	54–46
50–130	19–13	52–35
60–120	17–09	49–07
70–110	15–55	46–47
80–100	15–13	45–27
90	15–00	45–00

At the 63-deg 45-min position, the initial contact point of the face of the tooth with the work still remains at E, as is shown in Fig. 11. However, as the cut progresses, the entire face of the tooth along the corner angle gives a line contact from A to C. This line contact is at a distance from the cutting edge equal to the feed per tooth.

In Fig. 12 we can see that between the 63-deg 45-min position and the 70-deg 45-min position, the face of the tooth makes contact with the work in the order of points E at the face of the cutter; A at the top of the cut; and C at the intersection of the corner angle and the chamfer.

Fig. 13 shows the set of conditions which obtain at the 70-deg 45-min position where both A and E on the face of the tooth contact the work at the same time. This is followed by point C, also on the face of the tooth, and making the final contact. The point A is located at a 0.150-in. distance, which is the depth of cut from E at the face of the cutter. This marks the beginning of the true double-negative cutting action, which extends up to the 100-deg position.

Between the 70-deg 45-min position and the 90-deg position, the initial point of contact is at A. This is followed by contact at E, and finally by contact at C. Fig. 14 clearly illustrates this set of conditions.

When the cutter tooth contacts the work at the 90-deg position, the initial contact is still at A. This is on the face of the tooth, as illustrated in Fig. 14. This is, however, followed by a line contact which is along the face of the tooth extending from C to E. This is parallel to the chamfer and is shown in Fig. 15.

Fig. 16 shows the initial tooth contact being made on the face

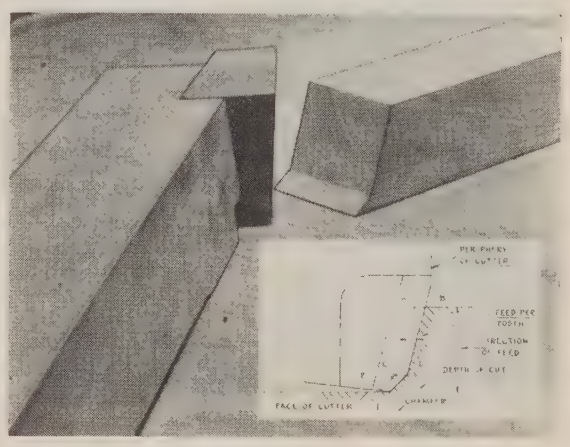


Fig. 10 Sequence of Initial Contact, E-C-A; Poor Operating Condition

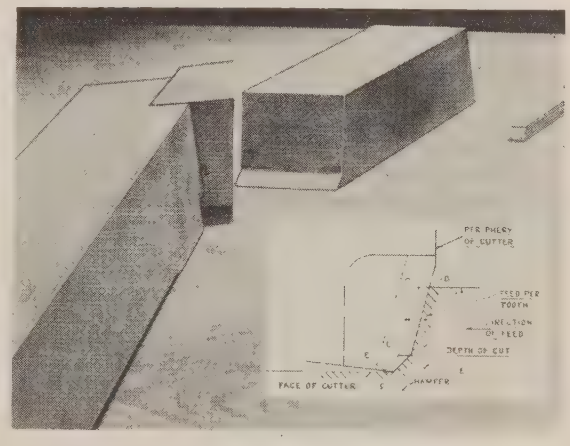


Fig. 11 Sequence of Initial Contact, E-(CA); Poor Operating Condition

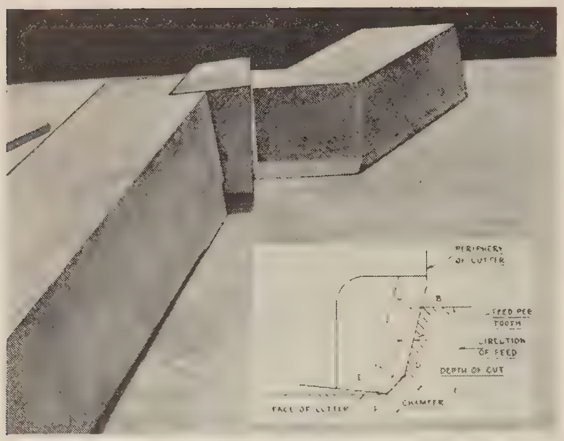


Fig. 12 Sequence of Initial Contact, *E-A-C;* Poor Operating Condition, Slight Improvement

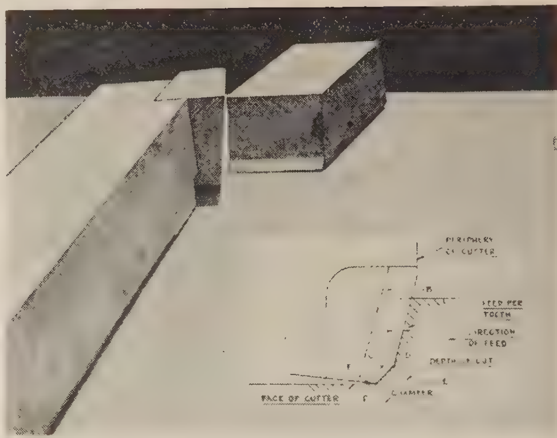


Fig. 14 Sequence of Initial Contact, *A-E-C;* Good Operating Condition

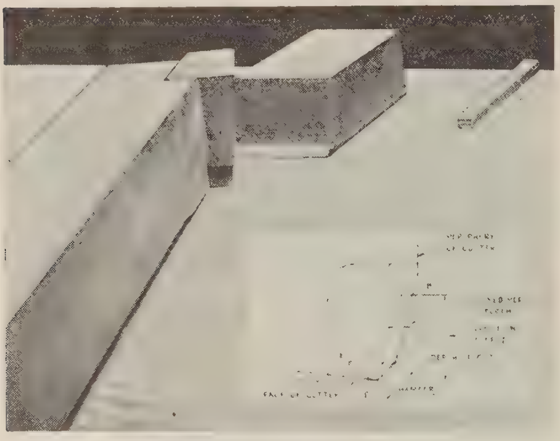


Fig. 13 Sequence of Initial Contact, (*EA*)-*C;* Start of Double-Negative Action and Good Performance

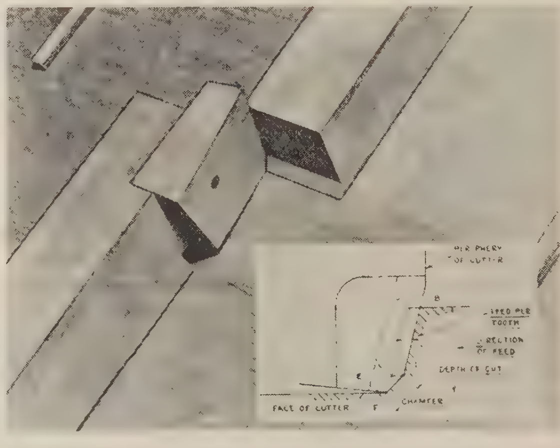


Fig. 15 Sequence of Initial Contact, *A*-(*EC*); Good Operating Condition, Slightly Retarded

of the tooth at *A*, as illustrated in both Figs. 14 and 15. This is followed by contact at *C* and finally at *E*. This set of conditions prevails from the 90-deg position up to the 100-deg position.

At the 100-deg position, the initial contact between the face of the tooth and the work is a line connecting point *A* with *B*. The length of this line represents the feed per tooth. Fig. 17 shows this condition. At this position the engagement angle between the face of the tooth and the entering side of the work changes from negative to positive. In other words, at this position the initial contact point moves from a point *A* on the face of the tooth, which is at a distance equal to the feed per tooth away from the cutting edge, up to a point *B* on the edge itself.

Finally, Fig. 18 shows the condition which exists at all angular positions beyond the 100-deg position. The initial contact between the cutting tooth and the work is at a point *B*, which is located at the cutting edge itself. This is followed by cutting-edge contact at *D* and then at *F*.

Analysis of Results

The data, compiled in Fig. 6, quickly indicate that a face-milling cutter having a 10-deg negative axial rake; a 10-deg nega-

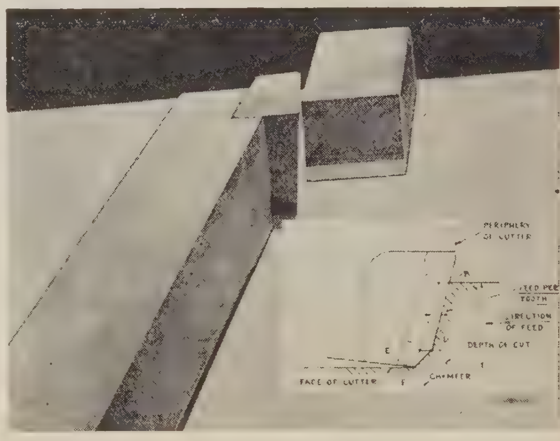


Fig. 16 Sequence of Initial Contact, *A-C-E;* Excellent Operating Condition

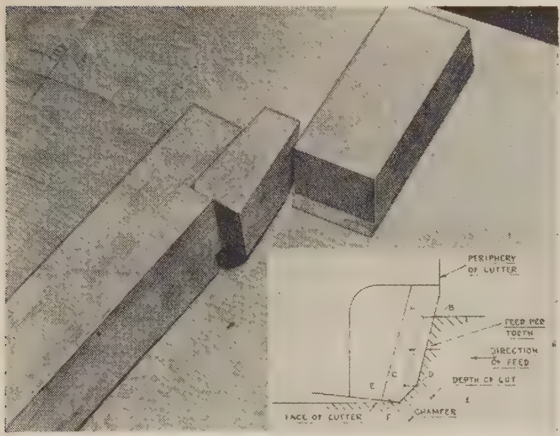


FIG. 17 SEQUENCE OF INITIAL CONTACT, (*AB*)-*D*-*F*; END OF DOUBLE-NEGATIVE ACTION AND EXCELLENT OPERATING CONDITION

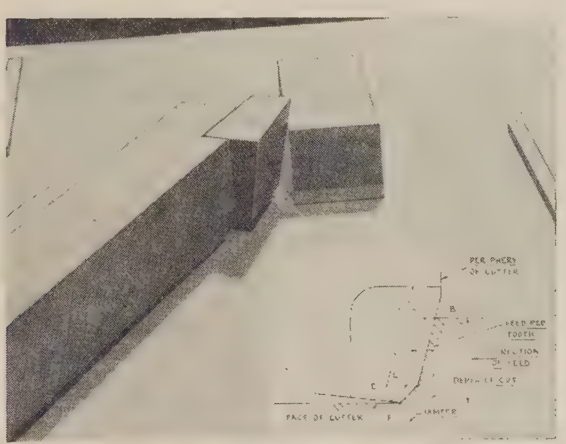


FIG. 18 SEQUENCE OF INITIAL CONTACT, *B*-*D*-*F*; FAIR TO POOR OPERATING CONDITION

tive radial rake; a 15-deg corner angle, and a $^1/_{32}$-in. × 45-deg chamfer:

1 Gives the longest cutter life when the nose or corner on the cutter enters the cut anywhere between the 70-deg 45-min and 100-deg work positions. The reason for this is the point *A* on the face of the tooth which contacts the work first.

(*a*) A 0.0105-in. feed per tooth gives the longest as well as the most uniformly consistent cutter life between the 70-deg 45-min and 100-deg work positions.

(*b*) A 0.0084-in. feed per tooth gives the shortest cutter life but has a tendency to cover a wider range of work positioning between 63 deg 45 min and 100 deg. Evidently the reason for this is the second point of contact with the work, which happens to be at *A* even though its initial point of contact is at *E*.

(*c*) The longest cutter life for a 0.0147-in. feed per tooth is confined within the narrowest range of work positioning for the four feeds used.

(*d*) At about the 90-deg tooth-entry position, the cutting edge along the 45-deg chamfer contacts the entry edge of the work broadside. This actually tends to decrease the length of run by a slight amount. An analysis of all tools run at this position shows that the initial breakdown occurs along the chamfer.

2 The life of the cutter is short and erratic when the cutter nose enters the work first anywhere between zero position and the 63-deg 45-min angular work position. This is due to the fact that the point *E* on the face of the tooth, which is below the point where the chamfer intersects the face of the cutter, contacts the work first.

This is substantiated by an analysis of the wear record of all the tools run at these positions, which indicates that a chip out of the cutting edge at *E* is followed by a complete collapse of the chamfer.

(*a*) It will be noted that an 0.0084-in. feed per tooth shows the longest cutter life, and that the 0.0126, 0.0106, and 0.0147-in. feeds per tooth, in the order given, show the shortest cutter life between the zero-deg and 55-deg work positions.

(*b*) At the 45-deg angular position, all of the feeds per tooth gave practically a zero cutter life. Plenty of runs were made at this work position to verify the results. There is no indication that this is the result of a change in the type of tooth contact because the same type of contact exists all the way from the zero-deg to the 63-deg 45-min angular position.

An analysis of the wear record of all the tools which were run at the 45-deg position showed that the cutting edge in the vicinity of the intersection of the corner angle and the chamfer consistently failed by spalling off in the same manner. It was assumed that this was due to the direction of, and also the intensity of, the impact load at the time the tooth hit the work. To verify this, an analysis was made of the major cutting forces at the time of the initial tooth impact, using the assumption that the longitudinal force (in the direction of feed) is 30 per cent of the tangential force, as outlined in Fig. 19.

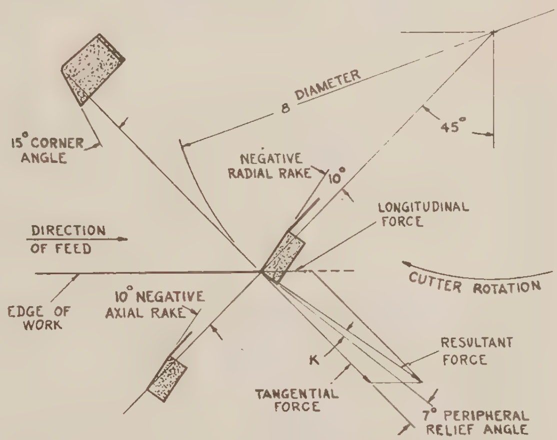


FIG. 19 GRAPHICAL ANALYSIS OF DIRECTION OF RESULTANT TOOTH-IMPACT FORCE, AT 45-DEG POSITION

This type of analysis was applied to each work position from the zero position to the 90-deg position. The angle *K*, which gives the relationship of the resultant force to the peripheral relief angle behind the cutting edge for each one of the various work positions, was determined graphically and then plotted as shown in Fig. 20.

It will be noticed that for the 45-deg position, which is shown in Fig. 19, the resultant-force line lies 3 deg 30 min within the peripheral-relief-angle line. It will also be noted that the angle *K* is zero for a work position of 32 deg. This means that the direction of the resultant force coincides with the peripheral relief angle behind the cutting edge. This also means that for all work-position angles which are less than 32 deg, the direction of

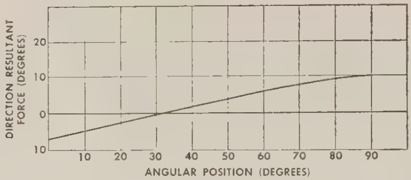


Fig. 20 Graph Showing How Direction K of Resultant Impact Force Varies With Work Position

the resultant force lies outside the peripheral relief angle, and shows why there is an inclination to spall off behind the cutting edge.

(*c*) There is a tendency for cutter life to decrease slightly at the 63-deg 45-min work position. This is apparently owing to the fact that at this position the entire cutting edge along the corner angle enters the work at the same time. This causes a momentary increased shock, which tends to break down the cutting edge.

(*d*) At the 70-deg 45-min work position, the face of the cutter tooth contacts the work at both A and E at exactly the same time. This momentarily increased contact also has a tendency to reduce cutter life by a slight amount.

Rating Work Position by Total Distance Cutter Is in Contact With Work

It is a known fact that when work is so positioned in relation to a face mill that the center line of the work and the center line of the cutter, measured in the direction of the feed, actually coincide, the arc of contact B between the cutter tooth and the work is at its minimum, Fig. 21. This arc of contact increases grad-

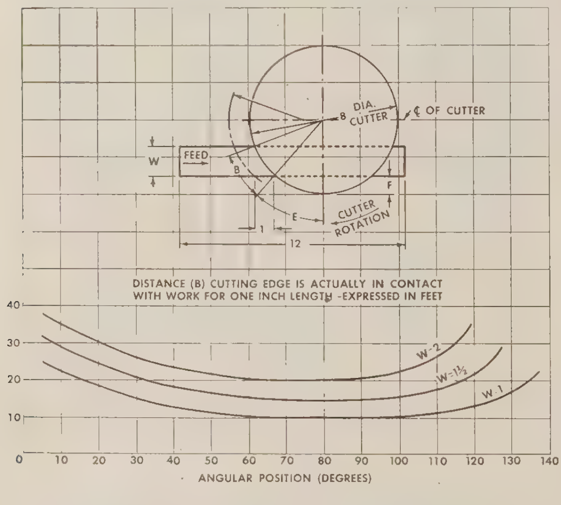


Fig. 21 Analysis of Length of Tooth Path Through Work When Considering Work Width and Work Position

ually as the work position is shifted to either side of this central position. It is at its maximum under the present set of conditions, either where E equals 10 deg and F equals 0.061, or where E equals 127 deg 33 min and F equals 6.437, as we saw listed in Table 1.

Table 3 shows the length of this arc of contact B, expressed in inches, for the various work positions for the $1^1/_2$-in. width of cut which was used in the test runs. This arc of contact (tooth path) has been called by the mathematical term "an arc of a looped trochoid." Since the width of cut is narrow, however, it will be treated here as an arc of the cutter circumference. This will give a close approximation to the theoretical curve already mentioned.

Table 3 also shows the total length of these arcs of contact expressed in feet for a 1-in. length of cut for the various work positions when using an 0.0084-in. feed per tooth. This is obtained by multiplying the length of the arc of contact for one tooth pass by the number of tooth engagements, with the work in a 1-in. length of cut for the given feed per tooth. (This is expressed in feet to reduce the number of digits in the figures.) The following example shows the method used to arrive at the total length of arc of tooth contact with the work for each inch of length of cut when using the 80-deg work position and the $1^1/_2$-in. width of cut

$$\frac{1.509\,(\text{Length of arc of contact}) \times 238\ (\text{rpm})}{2\,(\text{Inches per minute}) \times 12\ (\text{inches to feet})} = 14.96 \text{ ft}$$

The lengths, given in Table 3, were compiled for the several work positions, and the graph shown in Fig. 21 was plotted from them. This graph shows that there is a marked difference in the distance which a cutter tooth has to travel as the work position changes from either side of the central position where the center line of cutter and center line of work coincide.

This information is also expressed on a percentage basis, using the position where the arc of contact is shortest, as at 100 per cent.

In order to determine the effect of changing the width of the work on the length of arc of tooth contact with the work, similar data were compiled for both 1-in. and 2-in. cuts.

Using the data compiled for the $1^1/_2$-in-width cut in Table 3,

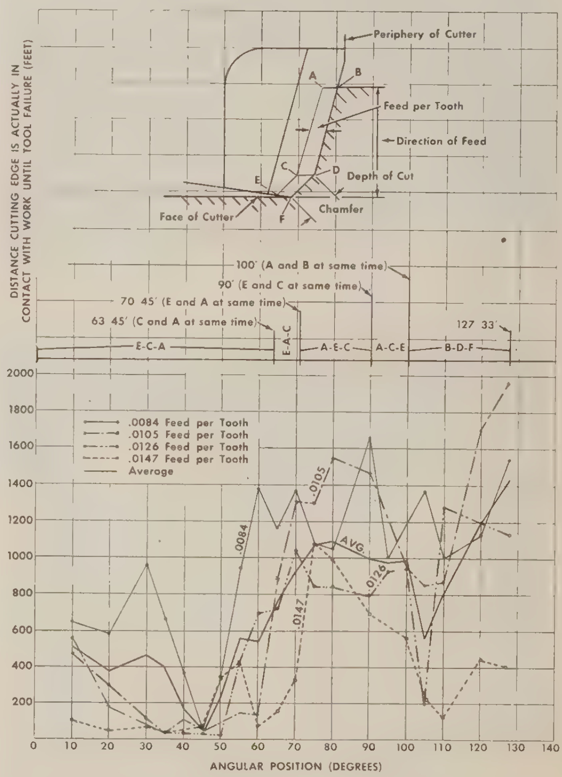


Fig. 22 Effect of Work Position on Cutter Life Based on Length of Tooth Path Through the Work

TABLE 3 EFFECT OF WORK POSITIONING AND WIDTH OF CUT ON LENGTH OF TOOTH

Angular position of entrance point on work, E	Length arc-of-contact tooth makes with work, B, in.			Total length arc-of-contact tooth makes with work for each inch length of cut, ft			Per cent total length of arc-of-contact with work using position of minimum contact as 100 per cent		
deg	W = 1 in.	W = $1^1/_2$ in.	W = 2 in.	W = 1 in.	W = $1^1/_2$ in.	W = 2 in.	W = 1 in.	W = $1^1/_2$ in.	W = 2 in.
5	2.565	3.253	3.857	25.43	32.20	38.25	256	215	191
10	2.278	2.960	3.560	22.59	29.40	35.31	227	197	177
20	1.843	2.487	3.066	18.27	24.66	30.40	184	165	152
30	1.534	2.136	2.690	15.21	21.15	26.67	153	142	133
40	1.322	1.884	2.414	13.11	18.65	23.93	132	125	119
50	1.178	1.708	2.219	11.68	16.90	22.01	117	113	110
60	1.065	1.593	2.094	10.56	15.83	20.77	106	106	104
70	1.028	1.528	2.031	10.19	15.16	20.14	102	101	100
75–31–22	...	...	2.021	...	...	20.04	...	...	100
79–11–35	...	1.509	...	...	14.96	...	...	100	...
80	1.004	1.509	2.028	9.95	14.96	20.11	100	100	100
82–49–09	1.002	...	...	9.94	...	...	100	...	...
90	1.011	1.538	2.094	10.02	15.24	20.77	101	102	104
100	1.052	1.625	2.258	10.43	16.06	22.40	105	107	112
110	1.138	1.802	2.608	11.28	17.83	25.86	113	119	129
118–58–17	...	...	3.552	...	...	35.23	...	...	176
120	1.298	2.167	...	12.87	21.50	...	130	144	...
127–32–39	...	2.954	...	...	29.25	...	...	195	...
130	1.621	...	...	16.08	...	...	162	...	...
137–15–15	2.276	...	...	22.57	...	...	227	...	...

and shown in the graph in Fig. 21, a new set of tool curves as shown in Fig. 22 was developed to take the place of those shown in Fig. 6. This was done by multiplying "the distance traveled to tool failure," expressed in inches, by the "total length of arc of contact with the work for each inch length of cut," expressed in feet, and plotting the result against the various work positions.

This set of curves follows the same general pattern as those shown before in Fig. 6, and these curves:

1 Amplify the point that at both the low and the high angular work positions, the rapid failure of the cutting edge on the cutter tooth is accentuated by the increased distance that it is actually in contact with the cut.

2 Clearly show that there are critical work positions at 45 and 105 deg.

3 Also show the effect of the tooth engagement with the work at the 63-deg 45-min, 70-deg 45-min, 90-deg, and the 100-deg work positions in about the same manner as when rated on the actual distance traveled to point of tooth failure.

Effect of Changing Corner Angle on Work Positioning

It frequently happens that the outline of the workpiece, where the cutting edge first contacts it, is of such a shape that a corner angle other than 15 deg is required to keep the corner of the cutter from hitting the work first.

Fig. 23 (a) shows a condition where a cutter tooth having a 15-deg corner angle and a small chamfer contacts the work first next to the face of the cutter. This naturally results in short cutter life because the intersection M of the chamfer at the face of the cutter usually chips out first.

Fig. 23 (b) shows the same work outline. However, the cutter tooth has been reground to an increased corner angle of 45 deg, which takes the initial contact with the work at some point N at a distance from the intersection of the corner angle and the chamfer. This slight change of conditions prolongs the life of the cutter.

When all the data from the test runs made with a face mill having a 15-deg corner angle used on work with a square corner (90-deg included angle) had been compiled, it was learned that the best work-position angles were between 70 deg 45 min and 100 deg. The analysis of the $^1/_{32}$-in. × 45-deg chamfer angle showed that the best work-position angles for a 45-deg angle were between 90 and 100 deg.

The same analysis also showed that the high limit of this range of best work-position angles was always a constant for a given negative radial rake angle on a cutter, and was equal to 90 deg plus the negative radial rake angle on the cutter, irrespective of the corner angle.

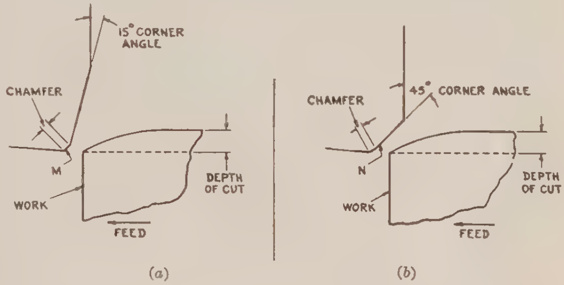


Fig. 23 Effect of Changing Corner Angle to Suit Work Contour

The analyses showed that the low limit of this range of the best angular work positions is at the work-position angle where the initial point of contact of the face of the cutter tooth and the work begins at M, in Fig. 23. A change in the corner angle on the cutter changes the angular work position where the initial contact at the point A, which was shown previously in Fig. 6, actually begins. For a 15-deg corner angle, the work position is 70 deg 45 min. For a 45-deg corner angle, the angular work position is 90 deg.

The curve X shown in Fig. 24 was developed as an aid in obtaining the best angular work positionings for other corner angles. This curve covers the work positions shown previously in Fig. 11, where points A and C contact the work at the same time. This curve shows that as the corner angle is increased with all other conditions identical, the range of operation of the best angular work position decreases, and vice versa.

Curve Y shows a similar line plotted to indicate where the face of the tooth makes initial contact at A and E at the same time, which is followed by contact at C. This set of conditions was shown in Fig. 13. The location is important because it tends to reduce cutter life. It also tends to decrease the width of range of longest cutter life as the corner angle increases.

All tests made while using the cutting angles of 15 deg corner angle, 10 deg negative radial rake, and 10 deg negative axial rake clearly indicated that the cutter life would tend to be at its lowest whenever the initial tooth contact with work would be at an angular position of 70 deg 45 min or less. This forces the work positions which gave the longest cutter life into the narrow operating band between 70-deg 45-min and 100-deg angular positions, as indicated by the curve marked 0.0105, which was previously shown in Fig. 6. This same curve also shows practically no cutter life for all angular positions of 60 deg or less.

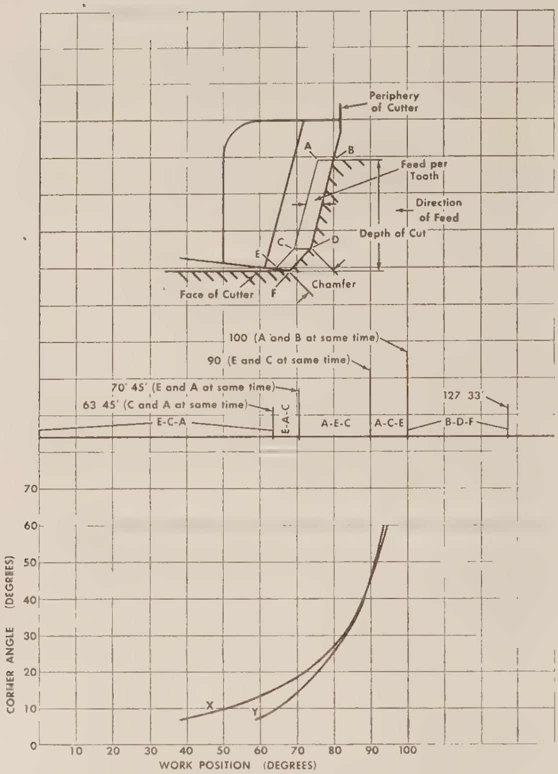


Fig. 24 Graphs Showing How Critical-Point Positions Vary With Corner Angle

The last-mentioned conditions offer a real challenge, and so a series of 60 runs was made with 40 tools while using increased corner angles to determine some means whereby the cutter life in general could be improved. The tools used were ground as follows: 14 tools to a 25-deg corner angle, 13 to a 35-deg, and 13 to a 45-deg corner angle. All other tool angles were kept the same as were ground on the tools which had the 15-deg corner angle, except for those tools with the 45-deg corner angle. In this case no chamfer was ground between the corner angle and the face of the cutter. Instead, it was broken lightly with a 320-grit silicon-carbide stone. A total of 20 runs were made for each set of tools which had the same corner angle, while using the work positions ranging between the 10 and 90 deg.

All the operating conditions were duplicated while using a table travel of $2^1/_2$ ipm (0.0105 in. feed per tooth). The distance traveled in inches per minute up to the time of tool failure was plotted for each one of the various work positions and corner angles, as shown in Fig. 25. The tool-life curve, previously shown as Fig. 6, for the same feed per tooth when using a 15-deg corner angle was also duplicated.

General Conclusions

A comparison of the four curves shows that for a face mill having a 10-deg negative radial rake and a 10-deg negative axial rake and operated under the conditions as already outlined:

1 Cutter life reaches maximum when a 15-deg corner angle is used. The only way this maximum cutter life can be obtained is by using special care to see that the initial point where the tooth enters the work lies between the 70-deg 45-min and the 100-deg work positions.

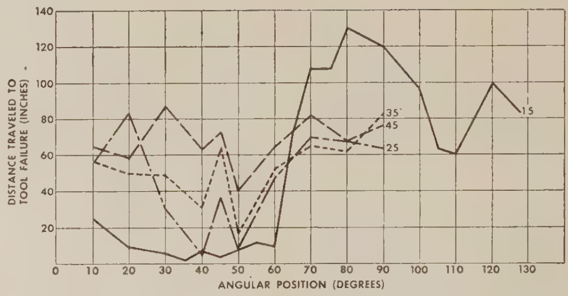


Fig. 25 Effect of Corner Angle on Cutter Life

2 The limited long-life working range between the 70-deg 45-min and the 100-deg positions, when using a 15-deg corner angle, makes it necessary for the face mill to overhang the entrance side of the work by an increased amount. This results in the use of face mills slightly larger in diameter than have been ordinarily used on most face-milling jobs.

3 Cutter life is at its minimum with a 15-deg corner angle when the initial point where the tooth enters the work lies between the zero-deg and the 70-deg 45-min work positions.

4 Cutter life can be improved by using corner angles greater than 15 deg when it becomes necessary to operate a face mill within the zero-deg to 70-deg 45-min work positions.

5 When a 25, 35, or 45-deg corner angle is used, the cutter life for the 70-deg 45-min to 100-deg work position is only from 50 per cent to 60 per cent of that obtained from a face mill which has a 15-deg corner angle.

6 Any face mill with a 45-deg corner angle will produce the maximum cutter life when practically no attention at all is given to work positioning. The 35-deg and then the 25-deg corner angles rate next in the order given.

7 Mediocre but somewhat consistent results can be obtained with a face mill having a diameter slightly larger than the work width, provided that it has a 45-deg corner angle.

8 The face mill can be adapted to various face-milling operations by grinding an appropriate corner angle on it and then properly positioning the cutter entry edge of the work to the cutter. This indicates the possibility of standardizing carbide-tipped face mills and adapting them to a large number of jobs.

9 Cutter life was shortest for the 45-deg work position when operating a face mill with a 15-deg corner angle at all feeds per tooth. When the corner angle was changed to either 25, or 35, or 45 deg, the cutter life was considerably longer for the same position. However, the one work position which gave the reduced cutter life was transferred to two positions, one at 40 deg and the other at 50 deg, on either side of the 45-deg position. This set of conditions is worth further analysis. It must be kept in mind that all conclusions were reached as a result of using a face mill on work having a $1^1/_2$-in-width cut.

While these tests were being run, it was observed that when runs were made at the critical 45-deg work-entry position the cutter life could be increased many times for some cuts wider than $1^1/_2$ in., even though the stock removal was much greater. This seems to indicate that the position where a cutter tooth leaves the work or a possible variation in cutter-flywheel-spindle momentum may affect cutter life. This is another phase of work positioning which will be given extensive study in the future.

(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Circular Bulging of Aluminum-Alloy Sheet at Room and Elevated Temperatures[1]

By GEORGE SACHS,[2] GEORGE ESPEY,[3] and G. B. KASIK[3]

In this paper preliminary results are reported on the strength characteristics and forming limits of various aluminum-alloy sheet, when subjected to hydraulic pressure over a circular area at temperatures up to 500 F. The strain distributions were analyzed by means of a photogrid, after these bulges failed. The bulges exhibited a practically balanced biaxial strain state over most of the contour with the exception of a very narrow range of local deformation in the vicinity of the fractures. The maximum meridional strain outside the necked area observed at the pole of a bulge, in spite of its biaxial nature, generally exceeded the uniform elongation known from tensile tests. The forming limits for all annealed conditions increased considerably with increasing temperature if the temperature exceeded a certain critical value, between 250 and 400 F, depending upon the alloy. The heat-treated conditions of the alloys 24S and 61S were only slightly affected by the testing temperature, while the forming limits (uniform stretch) of 75S-T, R301W, and R301T were materially increased at elevated temperatures. The reduction of thickness, which is a measure of the ductility of an alloy, followed approximately the same trend as the forming limit.

THE use of a fluid as a pressure medium has been restricted in the past primarily to testing purposes and to such minor cold-forming operations as the bulging of recesses in tubular parts. However, various shapes can be formed by bulging or hydraulic application of uniform pressure to one side of a sheet over a confined area. This process has found a restricted commercial application for the forming of letters and other shapes on signs.

Temperatures in the vicinity of 400 F (200 C) are, to a small extent, being used in the bending, joggling, and dimpling of high-strength aluminum-alloy sheet. The application of temperatures in this range presumably does not impair the properties of high-strength alloys.

Hydraulic bulging at elevated temperatures has been used successfully for the fabrication of magnesium-alloy parts. On the contrary, this method has previously not been found useful for the hot-forming of aluminum alloys, as an excessive temperature was considered necessary, 850 F (450 C), approximately.

The test results obtained on this project, however, indicate that temperatures between 250 and 400 F (120 to 200 C) substantially improve some forming characteristics of a number of high-strength aluminum alloys.[4] This fact, if substantiated by further experimentation, may render forming by means of hydraulic pressure at elevated temperatures a valuable addition to the various commercial methods of aluminum-alloy-sheet forming.

Examples of specimens bulged for this investigation are shown in Fig. 1.

The metal at the crown or pole of a circular bulge is subjected to a practically uniform tensile stress, and tensile strain in all directions parallel to the metal surface. This applies throughout the test until the "forming limit" is reached when failure by local necking or fracturing occurs. Such a "balanced biaxiality" constitutes a fundamental stress-and-strain state, for it is a borderline case which many types of actual forming operations approach and which is present in aircraft and other structures in service. Therefore, rather than measure the depth of the cup or bulge at the moment of failure, the unit strains (in the vicinity of the break) which the metal exhibits after failure, were determined. These are considered as important metal characteristics.

No attempt will be made, at present, to evaluate the fundamental significance of the data discussed here. Further reports will cover various other phases of cold- and hot-forming aluminum alloys, and their fundamental analysis.

MATERIAL, EQUIPMENT, AND PROCEDURE

Materials Tested

Twenty-one different conditions of seven aluminum alloys (3S, 52S, 61S, 24S, 75S, R301, and R303) were tested. The alloys 3S, 52S, and 61S are supplied in bare (unclad) sheets, and only bare R303 was tested. 75S and R301 were available only as clad sheet, and alloy 24S was tested both bare and clad.

For the general tests on circular bulges, sheet of the same thickness for all alloys, 0.040 $^{+0.002}_{-0.0015}$ in., was selected. For any particular alloy condition, specimens were cut from a single sheet for which the thickness variations were within ±0.001 in.

The yield strength, tensile strength, and elongation (2 in. gage length) in both the transverse and longitudinal (rolling) directions are assembled in Table 1.

The various alloy conditions are assembled into two main groups and four subgroups as follows:

Group I Annealed Sheet	Group II Heat-Treated Sheet
(A) Nonaging alloys	(A) Room-temperature-aged alloys
(B) Heat treatable alloys	(B) Artificially aged alloys

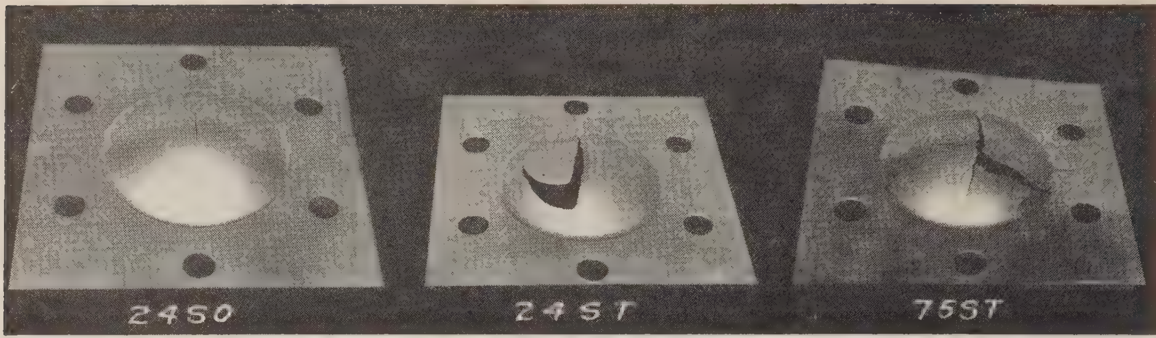
[1] This paper is the first published report on the research program on hot-forming of aluminum alloys conducted at Case School of Applied Science under contract with the Office of Production Research and Development of the War Production Board. This research, which was supervised by the War Metallurgy Committee under the "restricted" Project NRC-547, is the basis of this paper which has been released for publication by the O.P.R.D.

[2] Professor of Physical Metallurgy, Case School of Applied Science, Cleveland, Ohio. Mem. A.S.M.E.

[3] Department of Metallurgical Engineering, Case School of Applied Science.

Contributed by the Metals Engineering Division of The American Society of Mechanical Engineers.

[4] A particularly valuable application of hot-forming, at these temperatures, has been found to be the roll-forming of sections in the rather brittle high-strength alloys, suggested recently by an unpublished report on another phase of this W.P.B. research investigation.



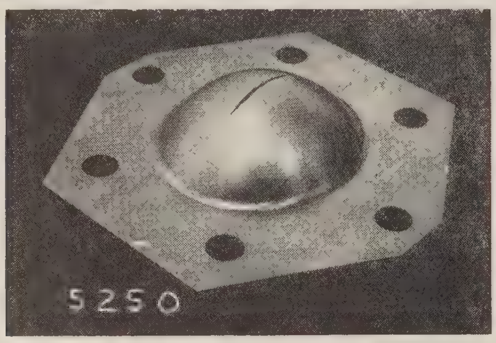


Fig. 1 Typical Hydraulic Circular Bulges

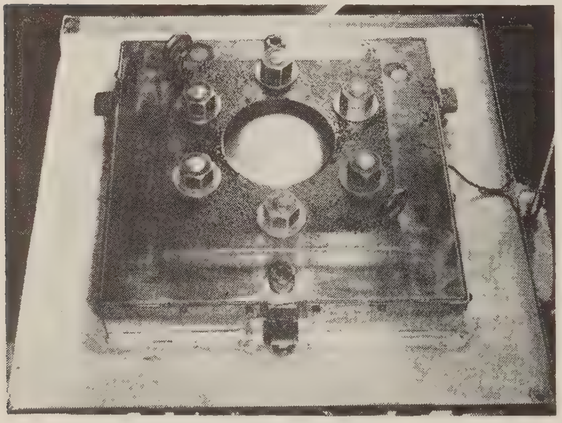


Fig. 2 Head for Hot Hydraulic Circular Bulging With Specimen in Place

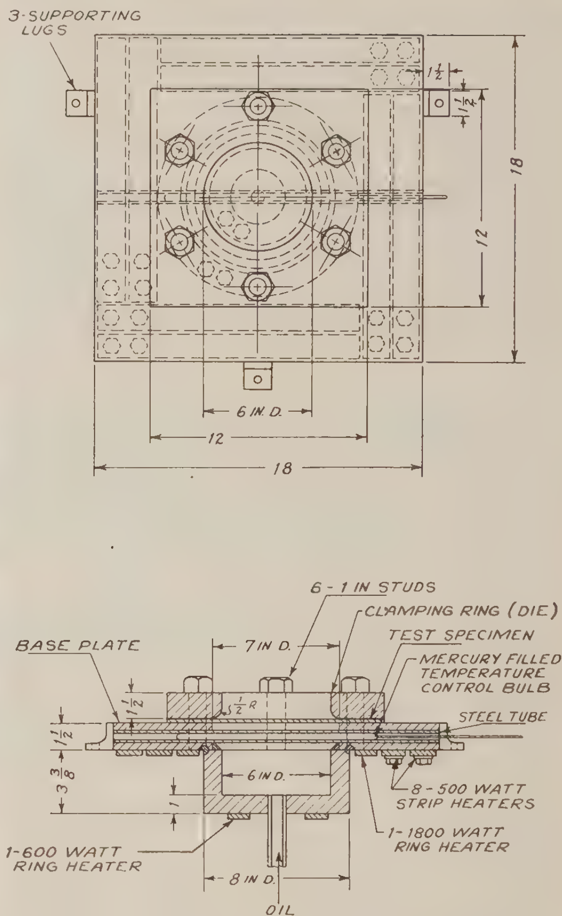


Fig. 3 Detail of Hot Hydraulic Bulging Head

The alloys investigated represent all conditions for the structural alloys of high strength available at present for production and a few additional alloys which are also used in aircraft fabrication, because of their superior forming properties.

Details of Equipment

The equipment[5] consisted of a bulging head and a hot-oil displacement cylinder capable of furnishing oil under pressure up to 3000 psi to the head on which the specimen was clamped, Figs. 2 and 3. The clamping ring (die) had a $^1/_2$ in. radius to prevent tearing of the specimen at the edge of the die contour. Originally, an as-machined surface of both the base plate and clamping-ring interfaces provided satisfactory gripping surfaces. As the die became nicked it was necessary to serrate the surface with four buttress teeth per inch ($^1/_8$ in. deep $\times$ $^1/_4$ in. long).

Oil pressure for the bulging was supplied from a displacement

[5] The equipment for these tests was developed by E. J. R. Hudec and C. L. Bennett, Instructors under the supervision of G. B. Carson, Associate Professor of Mechanical Engineering, Case School of Applied Science.

TABLE 1 TENSILE PROPERTIES OF SHEET USED IN HOT HYDRAULIC CIRCULAR BULGING

(Each Value represents average from two tests)

SHEET ALLOY	TRANSVERSE SHEET THICK. IN.	YIELD STR. PSI	TENSILE STR. PSI	ELONG. IN 2" %	LONGITUDINAL YIELD STR. PSI	TENSILE STR. PSI	ELONG. IN 2" %
GROUP I-A							
3SO	.041	7520	17100	31.5	7400	17200	32.0
52SO	.039	13800	26300	25.0	13700	27900	24.0
GROUP I-B							
24SO Bare	.0415	14500	33400	17.0	14400	34500	17.5
24SO Clad	.039	12500	28200	17.5	12800	39100	16.0
R3010 (Clad)	.0395	10400	23700	19.5	10600	25500	23.0
75SO (Clad)	.040	16100	32300	14.0	16700	33100	14.0
61SO	.040	7800	17300	24.5	8220	17800	26.0
GROUP II-A							
24ST Bare	.0395	43700	67400	17.5	48400	69500	19.5
24ST Clad	.0395	41300	64700	17.0	47200	65700	17.0
24SRT Bare	.0395	55000	71000	13.5	60200	72400	13.5
24SRT Clad	.0385	51600	67500	13.0	59200	69300	11.5
R301W (Clad)	.038	36200	61300	19.5	40700	63600	21.5
61SW	.038	29700	41300	15.5	33000	42000	17.0
GROUP II-B							
24ST81 Bare	.040	60400	68900	5.0	62600	70800	7.0
24ST81 Clad	.040	59500	66500	5.5	62300	68000	6.5
24ST86 Bare	.041	65900*	72800	4.5	66400*	72800*	5.5
24ST86 Clad	.040	63400	69400	5.0	65100	70200	6.0
R301T (Clad)	.0375	57600	69100	10.5	60900	70100	10.5
75ST (Clad)	.041	64400	78000	12.0	70700	80500	12.0
61ST	.039	38100	46000	13.0	39800	46600	13.5
R303T315 (Bare)	.041	67100	75500	9.5	70000	76200	9.0

*Does not meet minimum specified by the Army Air Force Specification, No. 11354.

cylinder, the piston of which was actuated by the lead screw of a broaching machine that was driven from a variable-speed drive, Figs. 4 and 5. The total volume-displacement capacity of the cylinder was 95 cu in.

The bulging pressure was recorded by a helical-type 3000-psi recording pressure gage tapped into the oil line connecting the bulging head and displacement cylinder. A synchronous-motor drive for the recording chart permitted the recording of a pressure-time (stroke) curve for any desired test.

The elevated temperatures were obtained by heating the lower plate, the reservoir, and the oil-displacement cylinder by means of resistance heaters. A mercury-tube temperature-control switch, sensitive within ±5 deg F, inserted into the base plate, controlled the temperature of the bulging head and the oil in the head. Two manually adjusted thermoswitches (50–600 F) were used for controlling the temperature of the oil in the displacement cylinder. The actual temperature was indicated by a thermometer set in a well in the exhaust end of the cylinder, Fig. 5.

The temperature of the specimens and various parts of the bulging fixture was determined (within ±3 deg F) with a Cambridge surface pyrometer, mold type, 50–600 F range, having a minimum accuracy of 2 per cent of full scale.

To protect the operators from the danger of hot oil at high pressure, a safety-glass cover was constructed to fit over the bulging head, Fig. 4. The cover was built of $^{1}/_{4}$-in. heat-tempered plate glass set in a $^{1}/_{8} \times$ 1-in. angle-iron frame.

For the low-temperature (30–50 F) tests, a small 3-gpm rotary gear pump was employed by which refrigerated oil was pumped from a 5-gal sump, containing dry ice as a coolant, to the reservoir of the bulging fixture, and thence circulated through the displacement cylinder. Dry ice was placed around the bulging head to maintain it at testing temperature.

Testing Procedure

A 20-line-to-the-inch orthogonal grid was applied to one side of each test specimen by a modification of the photogrid process

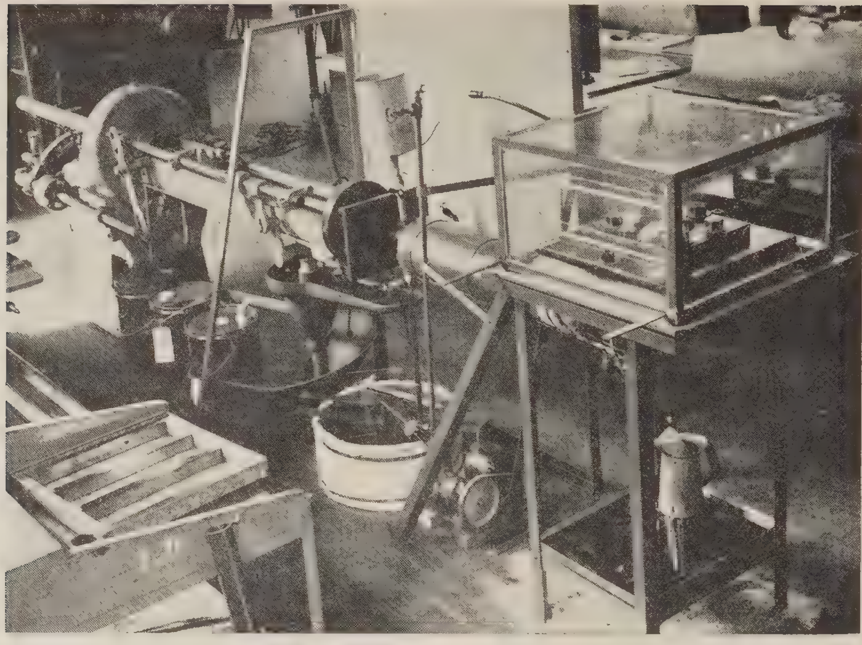


Fig. 4 Equipment for Experimental Hydraulic Bulging

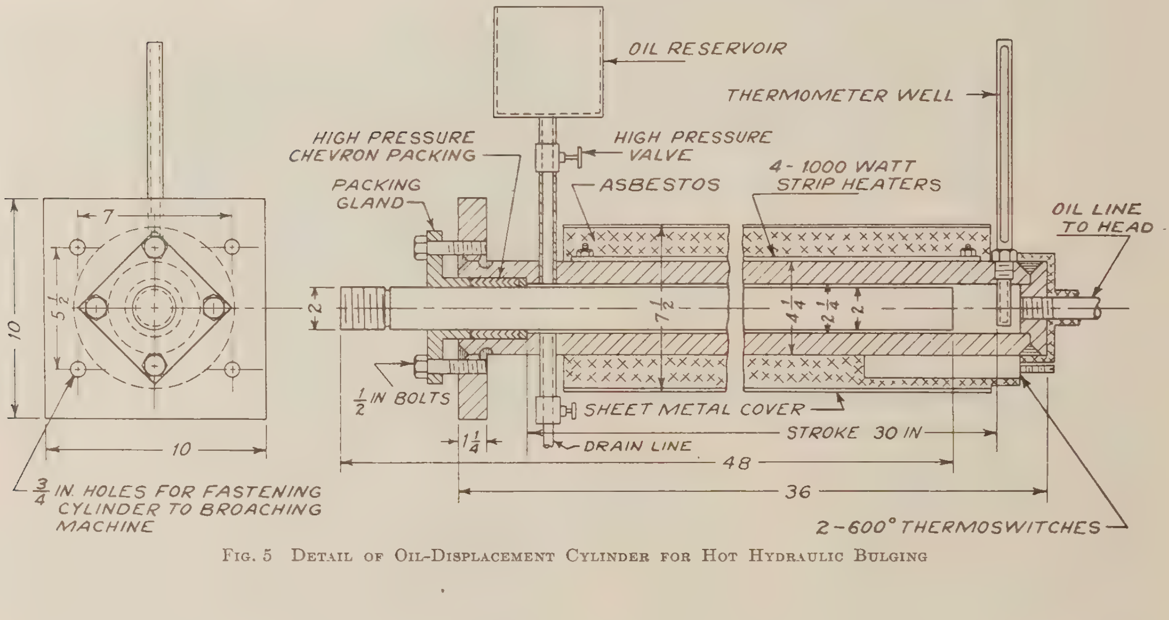


Fig. 5 Detail of Oil-Displacement Cylinder for Hot Hydraulic Bulging

(1, 2).[6] The grid lines were applied parallel to the transverse and longitudinal directions of the sheet. The maximum variation of the grid for $^1/_2$ in. gage lengths was 0.5 per cent.

The 12-in-sq test specimens were set in a jig and drilled with six $1^1/_{16}$-in-diam holes to accommodate the 1-in. studs in the base plate.

The bulging head, with the clamping ring in place, and the displacement cylinder were held at the predetermined testing temperature for at least 15 min prior to the start of a given run. To minimize the temperature gradient across the bulge and to insure the absence of air in the head, care was exercised to have the oil level in the reservoir flush with the surface of the base plate before placing a test specimen on this fixture.

The general testing procedure was to place the test specimen on the bulging head, set the preheated die loosely in place and allow 10 min time in which to take several temperature determinations around the bulge, and then tighten the clamping ring onto the specimen. It was found that the sheet reached a constant temperature within 2 min after being placed on the bulging head. The circular edge of the sheet next to the die was generally above the temperature of the center of the sheet by 5 to 10 deg F in the vicinity of 250 F and 10 to 20 deg F in the vicinity of 500 F.

The tightening of the die, which took a few minutes, was deferred until near the end of the 10-min period (after 7 min of preheating), for it was found that premature tightening favored heat-warping of the restrained sheet specimen. After the 10-min heating period the safety cover was placed over the head and the actual bulging was started and proceeded without interruption until failure. The forward speed of the piston was held constant throughout these tests at $5^3/_8$ ipm; this gave a displacement volume of 17.7 cu in. per min. Each test required approximately 2 min for the actual bulging operation. The time required to remove the clamping plate brought the total time for each test, during which the sheet specimens were exposed to the elevated temperature, to between 14 and 16 min.

For the purpose of comparing the properties of the various alloys at different testing temperatures, the maximum bulging pressure or "bursting pressure" has been used as a measure of the metal strength. In addition, the strain distributions along the "principal axes" after failure of the specimens were determined, and the ductility characteristics were obtained, as discussed in the following section.

Measurement of Strains

The over-all investigation, of which this paper is a part, entails the consideration of the effects of various degrees of stress biaxiality. For a better understanding of the forming limits, this necessitates the theoretical consideration of various geometrical shapes, which include the border cases of a flat sheet, a tube, and a sphere. It has been found necessary to deviate from general practice and adopt a different and comprehensive system of co-ordinates and directions which will permit an unambiguous description of any stress or strain in any given part.

A system of co-ordinates which will fit those cases having two perpendicular planes of symmetry for which the strains are measured along the "principal axes" has been selected as follows: The intersection of the two planes of symmetry and surface will be designated as the "pole." For any point on the surface, the "normal direction" N is readily recognized as the direction perpendicular to the surface at that point, Fig. 6. A "meridian line" is taken to be the intersection between the part surface and a plane passing through the pole and the line normal to the surface at the point under consideration. At the given point, the "meridian direction" M is determined by the line of tangency to

[6] Numbers in parentheses refer to the Bibliography at the end of the paper.

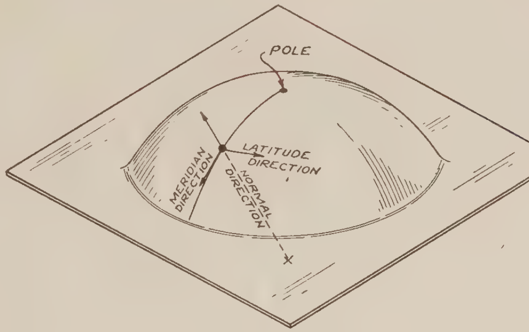


Fig. 6 Diagrammatic Representation of Co-Ordinate Designations

the meridian line. The "latitude direction" L is defined as the perpendicular to both the normal and meridian directions at the point under consideration. The relation of these terms to those used for a plane, a tube, and a sphere are readily recognized; they are assembled in Table 2.

TABLE 2 TERMS USED IN STUDIES

Adopted terms	Common terms: Plane	Tube	Sphere
Normal N	Normal	Normal or radial	Normal or radial
Meridian M	(In the surface, longitudinal or transverse)	Axial or longitudinal	Circumferential or tangential
Latitude L		Circumferential or tangential	Circumferential or tangential

In the case of an extended part, the "principal axes" will be defined by the intersections between the part surface and the two symmetry planes. The axis in the direction of the larger dimension will be designated by the letter a, and that of the smaller dimension by the letter b. Even in the case of a circular bulge such axes are also necessary to designate the grain direction.

For the circular bulge the a-axis has been chosen as parallel and the b-axis as perpendicular to the rolling direction of the sheet. The strains have been designated as "meridional" e_{aM} or e_{bM}, if measured along the axes a and b, respectively, and "latitudinal" e_{aL} or e_{bL} if measured perpendicular to the axes, where the subscripts a and b refer to points along the a- and b-axes, respectively. Strains not measured at points on one of the principal axes will be referred to a special set of co-ordinates for the particular case in question.

It was observed in the tests, as discussed later, that the strain distributions were usually characterized by a rather flat maximum at the pole of the bulge, disregarding a very narrow range containing the failure. These maxima in the principal meridional directions of the sheet were considered as the "forming limits" for the bulging processes, Figs. 9 to 12, inclusive, and represent the maximum amount of strain which the metal permits without failure by local necking or fracture. The strain at the forming limit for a bulged part could not be obtained in the same manner as for a regular tensile test specimen, i.e., by measuring a value of strain some distance away from the fracture. The measurements of the strain along the two major axes of the specimen, Fig. 7, with w, and cross-grain x, however, revealed that the strain values both parallel to the direction of each axis or "meridional," and perpendicular to the direction of each axis or "latitudinal," increased continuously from a zero value somewhere at the radius of the die to a maximum value at the crown of the bulge; a few representative plots are illustrated in Figs. 7 and 8.

For a rapid determination of the forming limits, the strains occurring in an area approximately 3 in. sq and in the proximity

of the pole of the bulge were measured along and in the direction of the principal axes by means of a strip of orthogonal co-ordinate paper to within $^1/_2$ per cent (stretch). A gage length of $^1/_2$ in. was generally used, although in some cases a gage length of $^1/_4$ in. was necessary for measurements near the break and in directions intersecting the failure on both sides (these values of $^1/_4$ in. gage length are distinguished by parentheses in the curves). No such measurements were made closer than $^1/_8$ in. to the fracture, nor across the fracture, i.e., including the fracture. A number of curves obtained in this manner are represented in Figs. 9 to 12, inclusive. That these measurements do not include the necked regions is shown by the thickness distribution perpendicular to the locus of failure, Fig. 13. From the figure it can be seen that for the very ductile alloy 52S-O, the necked region does not extend more than 0.01 in. on either side of the fracture.

The "reduction in thickness," at the point of failure constitutes another metal characteristic, corresponding to the contraction in area (reduction of area) of a regular tensile test presumably measuring the ductility of the metal. This value was determined by measuring the thickness of the sheet at the fracture by means of a measuring microscope at $\times 10$ magnification. Three readings approximately evenly spaced over a total distance of $^1/_2$ to 1 in. at the center of the primary fracture yielded values which could be readily reproduced.

In order to obtain the local (normal) strain distribution in the circular bulge in the vicinity of failure, thickness measurements were made with a measuring microscope at appropriate intervals on specimens sectioned in a plane perpendicular to the locus of failure. Reduction in thickness values thus obtained for two alloys yielded quite smooth strain-distribution curves, Fig. 13.

General Characteristics of Strain Distribution

The agreement between the meridional strains along the two principal axes was rather close. The maximum was flat and quite definite for all specimens yielding uniform stretch values of less than 25 per cent, Figs. 7 to 12, inclusive. The maximum was sharper for more ductile materials and differed slightly in position and value along the two axes, a difference of up to 3 per cent for temperatures below 375 F, and of up to 6 per cent for the tests at higher temperatures being observed. Less regular curves were occasionally obtained at the highest testing temperatures, probably because of slight variations in temperature. It appeared that if the temperature differences over the surface of the bulge exceeded 10 deg F during the test, such irregularities resulted.

The values of both the meridional and latitudinal strains measured along any one meridional axis, showed a variation up to $\pm 2^1/_2$ per cent (stretch) for a meridian distance within 2 in. from the crown or pole of the bulges; this change was greatest for the soft alloys and/or tests at high testing temperatures, Figs. 7 to 12, inclusive.

The depth of a bulge within an accuracy of ± 10 per cent was primarily a function of the forming limit. The depth corresponding to various values of uniform stretch was approximately as follows:

Stretch, per cent	10	20	30	40	50	60
Depth, In.	1	$1^1/_2$	2	$2^3/_8$	$2^5/_8$	$2^7/_8$

The measured values for reduction in thickness generally exceeded those calculated from the meridional strains. The meaning of these values which should measure the local ductility of the metal is rather obscure, as a radical change of stress state from balanced biaxiality to plane strain occurs as soon as local necking begins. No attempts have been made, as yet, to evaluate the significance of the reduction-in-thickness measurements.

Some thickness measurements in a plane perpendicular to the

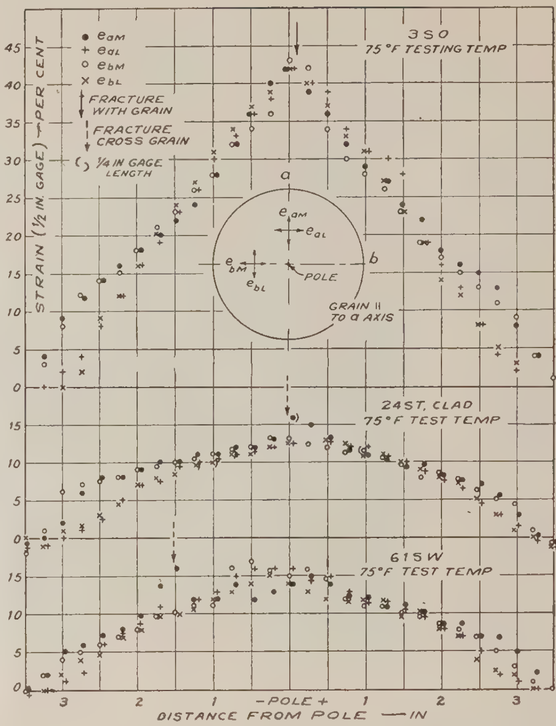


Fig. 7 Latitudinal and Meridional Stretch Distribution for Hydraulic Circular Bulges of Various Alloys Tested at Room Temperature

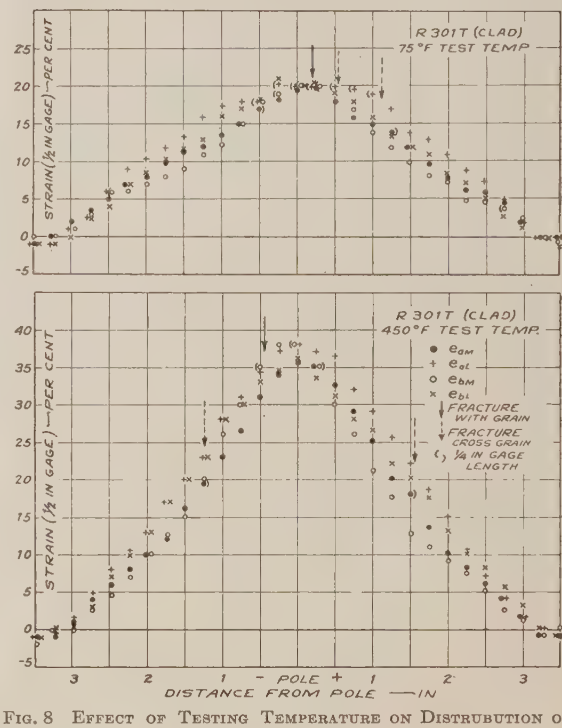


Fig. 8 Effect of Testing Temperature on Distrubution of Latiudinal and Meridional Stretch for R301T Clad, Hydraulically Bulged

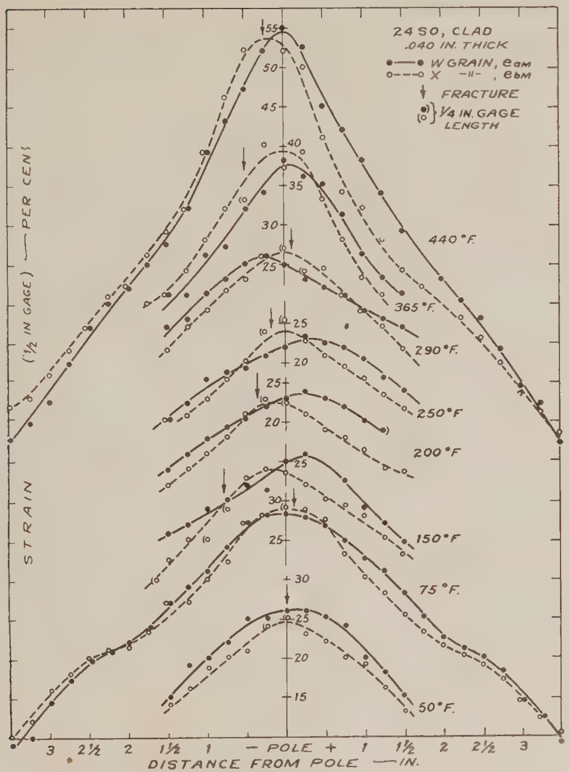


Fig. 9 Meridional Stretch Distribution for Circular Bulges of 24S-O Clad, Hydraulically Bulged at Various Temperatures

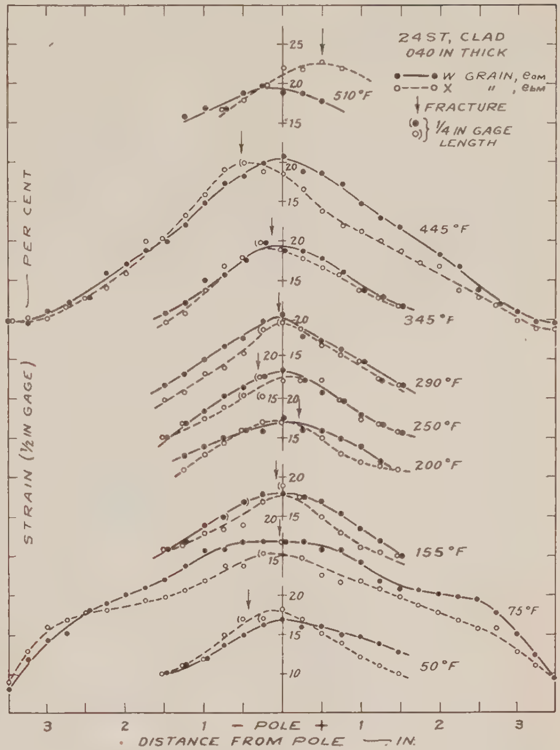


Fig. 10 Meridional Stretch Distribution for Circular Bulges of 24S-T Clad, Hydraulically Bulged at Various Temperatures

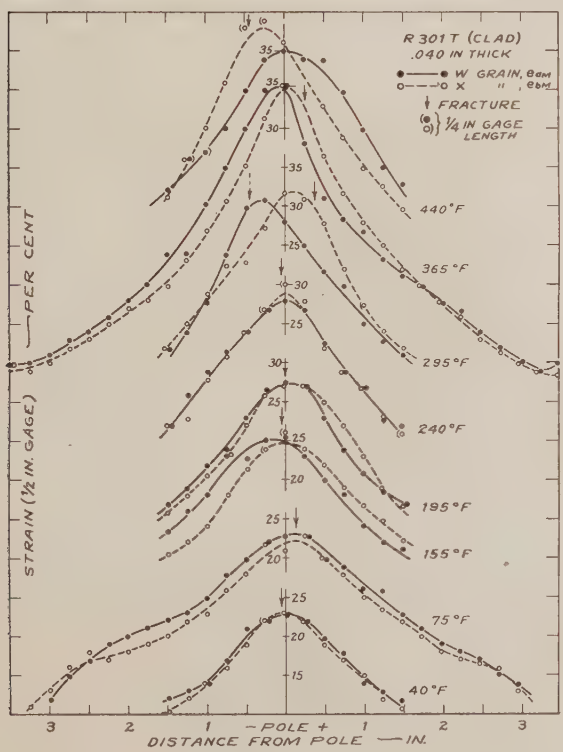


Fig. 11 Meridional Stretch Distribution for Circular Bulges of R301T Clad, Hydraulically Bulged at Various Temperatures

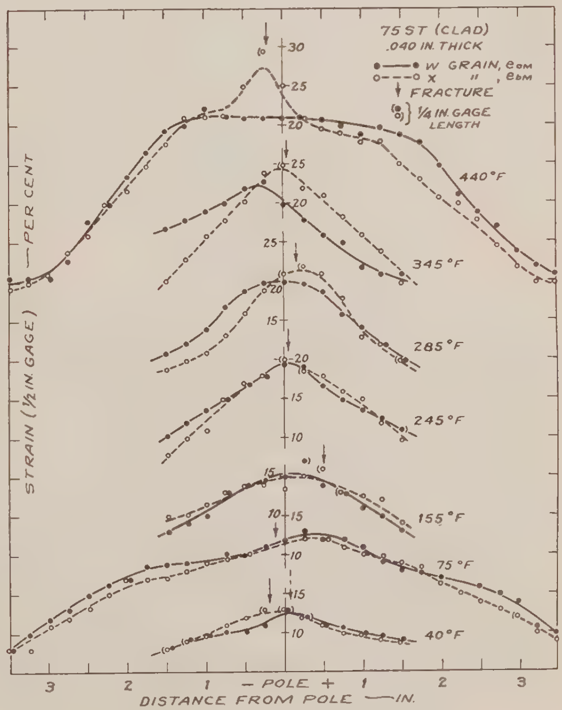


Fig. 12 Meridional Stretch Distribution for Circular Bulges of 75S-T Clad, Hydraulically Bulged at Various Temperatures

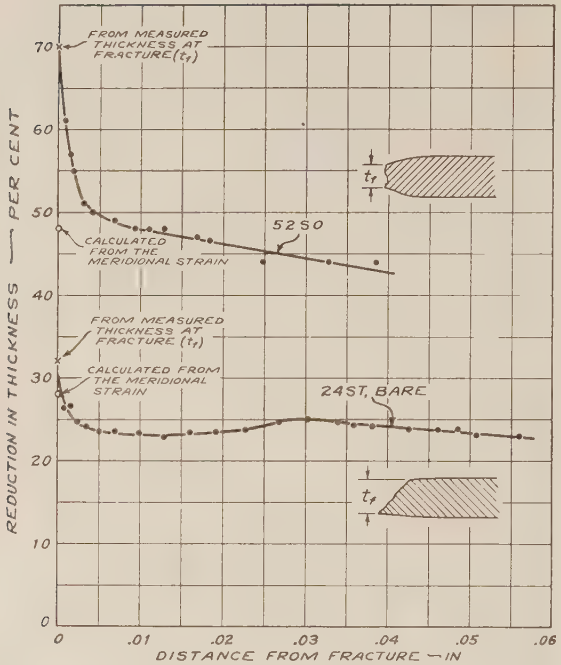


Fig. 13 Reduction in Thickness in Vicinity of Fracture for Two Representative Alloys Tested at Room Temperature

locus of failure show that the strain gradually increases toward the locus of failure and that the larger fracture strain only affects a very narrow region,[7] Fig. 13. A very ductile alloy, 52S-O, yielded practically the same forming limits from such measurements extrapolated to the fracture, as from the grid measurements. The more brittle alloys, such as 24S-T, exhibited some irregularities in the immediate vicinity of the failure. However, disregarding this range of irregularity the curve of thickness change extrapolates to practically an identical value, as calculated from the average of two meridional-strain values at the pole, as shown in Fig. 13.

The more tedious thickness measurements, therefore, would yield practically the same average value of forming limit as the rather simple grid measurements of the strains.

RESULTS OF TESTS

Bursting Pressure

The bursting pressure can be considered as a semiquantitative measure of the ultimate strength of the metal, under conditions of biaxial tension. While the ratio of bursting pressure to strength is affected by the amount of strain, or rather the curvature of the sheet, at the point of failure, this effect is apparently considerably smaller than that of the testing temperature.

As a general rule, the bursting pressure decreased continuously with increasing temperature above room temperature. Such a behavior is normal for any metal or alloy in a stable condition.

Of the alloys investigated, Figs. 14 to 24, inclusive, those of Group II-A, or the room-temperature-aged conditions, might have been expected to deviate from the general rule if artificial aging had taken place to such an extent that this change exceeded the reverse temperature effect. Obviously, this aging effect was not very pronounced, but a slight hump in the curves of all room-temperature-aged alloys, Figs. 18 to 20, inclusive, can be taken as such an effect.

A comparison of the two investigated nonaging alloys of Group I-A, showed 3S-O to be somewhat more respondent to the effect of temperature on the bursting pressure up to 200 F, while the per cent loss in this property was the same for both alloys at 400 F.

Of the alloys of Group I-B, or the annealed conditions of the age-hardenable alloys, Figs. 15 to 17, inclusive, 24S-O, both bare and clad, was least affected (10 per cent approximately) of all alloys investigated by testing temperatures up to 300 F. On the contrary, the strength of R301-0, 75S-O, and 61S-O, decreased noticeably at rather low testing temperatures, and to a considerable extent, 25 to 30 per cent, at 300 F. Alloy 75S-O suffered the largest decrease of the bursting pressure at still higher temperatures.

In the Groups II-A and II-B, Figs. 18 to 24, inclusive, i.e., room-temperature-aged and artificially aged alloys, respectively, again all 24S conditions exhibited considerably less reduction in strength on increasing the testing temperature than the other alloys, R301, 61S, and R303. At the highest temperatures investigated, 400 F and higher, the reduction of the bursting strength was largest for the alloys 75S and R303.

The absolute values of bursting pressure, of course, varied within wide limits for the different alloys and conditions, ranging between 350 and 1400 psi at room temperature, and between approximately 200 and 800 psi at 400 F. Thus, at 400 F the strength and forming resistance of the high-strength alloys are reduced almost to the values of the softest alloy investigated, 3S-O, at room temperature.

Of the alloys in Group II-A, 24S-T clad and R301-W clad, possessed approximately the same bursting strength at room temperature, while the bursting strength of 61S-W at the same temperature was only 60 per cent of 24S-T clad; 24S-T bare, was the strongest alloy of this group. At 400 F, however, the alloy R301-W required only approximately 70 per cent of the bursting pressure for 24S-T clad and at 450 F was not appreciably stronger than 61SW bare.

Of the heat-treated alloys, 75S-T clad, and R303-T315 bare, were the strongest up to 250 F, and 24S-T81 clad, and 24S-T86 clad, were the weakest at temperatures up to 300 F, while 24S-T bare and clad, R301-W and R301-T, were intermediate, Figs. 20 to 24, inclusive. At temperatures between 300 and 450 F, however, 24S-T required considerably larger bursting pressures than either 75S-T or R301-W, R301-T, and R303-T315.

At the lowest temperature investigated, approximately 35 F, the bursting pressure was usually equal to or slightly higher than at room temperature. Exceptions were 24S-T86 bare and, less pronounced, 24S-T86 clad. The decrease of the bursting strength of these alloys with decreasing temperatures, in the range below room temperature, is explained by the corresponding loss in ductility, which is apparent in both the reduction in thickness and the uniform stretch. The alloys 24S-RT both bare and clad, also exhibited irregular and low strength, and low ductility properties in a temperature range close to room temperature, and up to 250 F. Alloy 24S-T bare and clad yielded rather uniform results on specimens taken from a single sheet.

Artificial aging had very little effect on the alloy R301, while both cold-rolling (RT) and, even more, artificial aging (81 and 86), considerably reduced the bursting strength of 24S, up to a temperature of 400 F. At still higher temperatures, the different aged alloys of either 24S clad, or 24S bare, appeared to possess practically the same bursting strength.

Forming Limits

Regarding the effect of testing temperature on the forming

[7] This region is extremely small as compared with the region of necking in tension tests on 0.50-in-wide specimens.

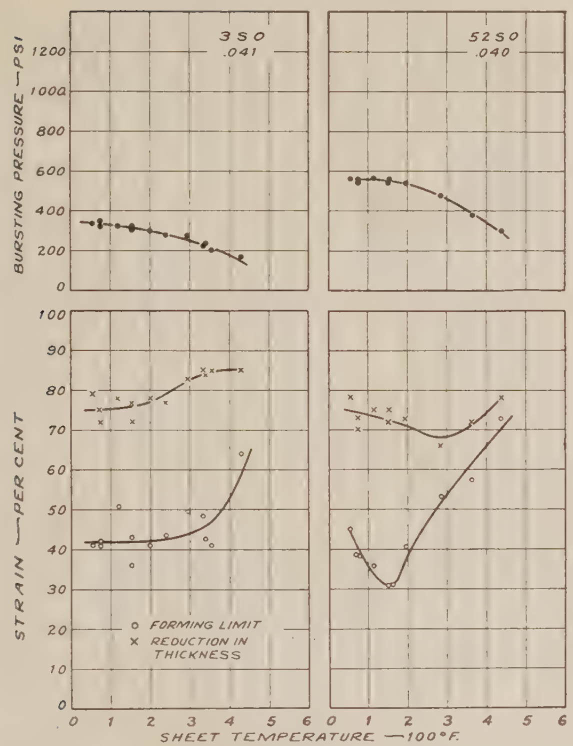


Fig. 14 Results of Hot Hydraulic Circular Bulging for 3S-O and 52S-O

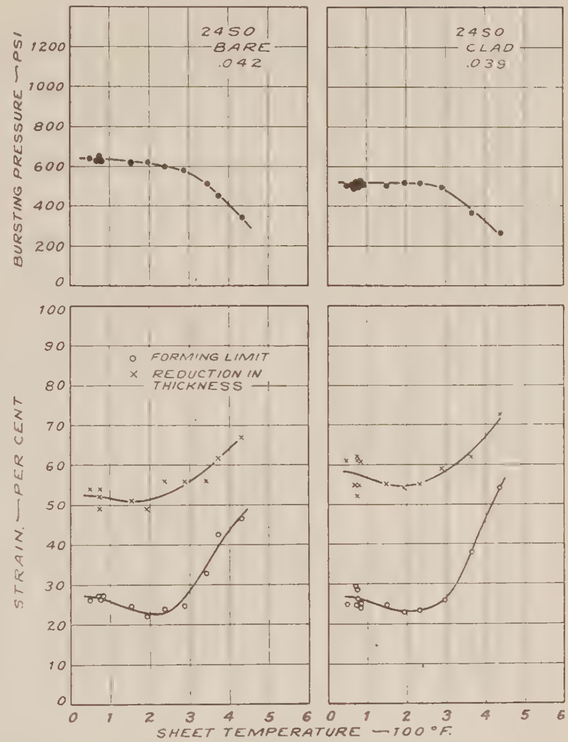


Fig. 15 Results of Hot Hydraulic Circular Bulging for 24S-O, Bare and Clad

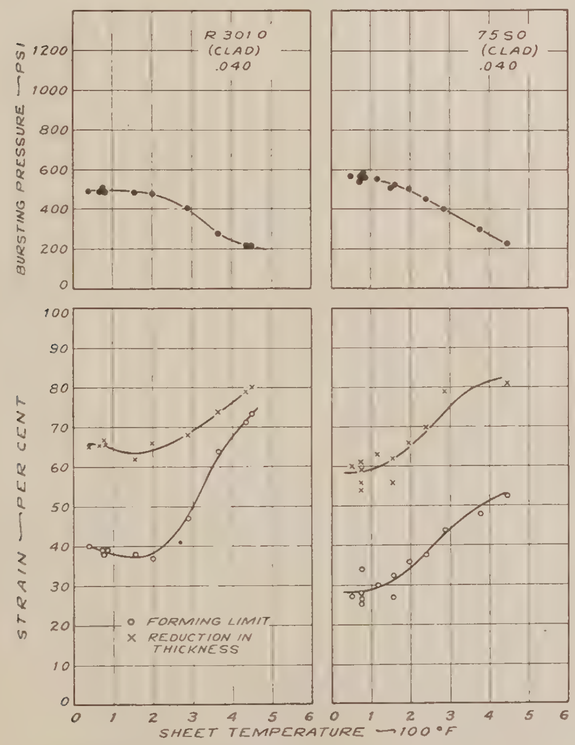


Fig. 16 Results of Hot Hydraulic Circular Bulging for R3010 Clad and 75S-O Clad

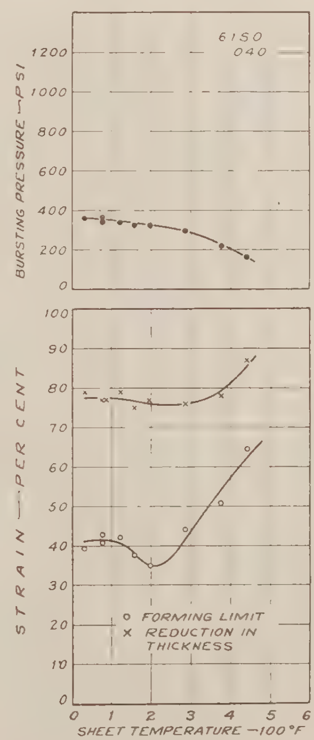


Fig. 17 Results of Hot Hydraulic Circular Bulging for 61S-O

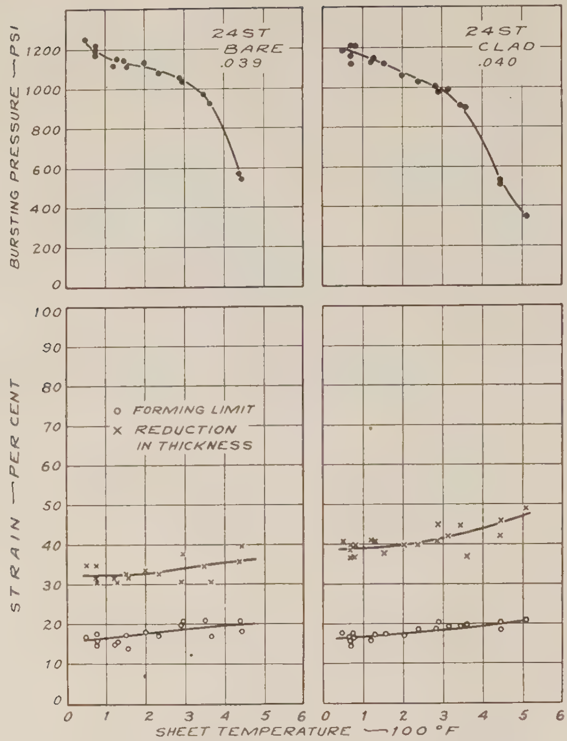


FIG. 18 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR 24S-T, BARE AND CLAD

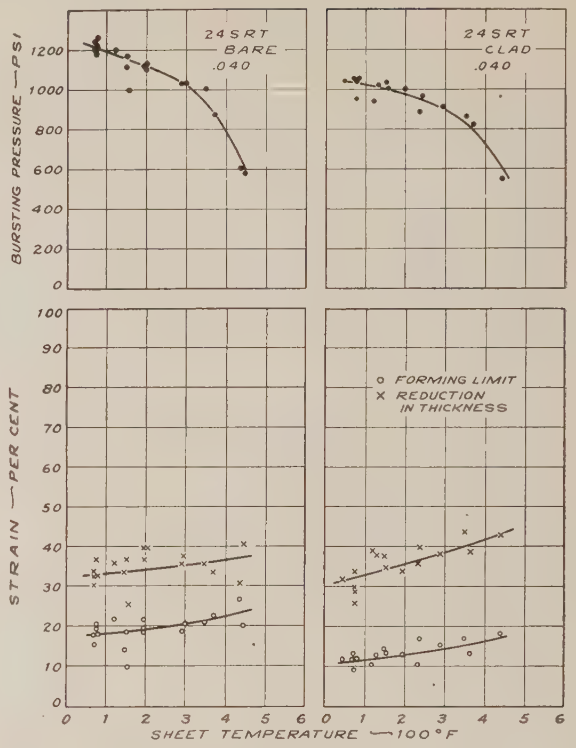


FIG. 19 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR 24S-RT, BARE AND CLAD

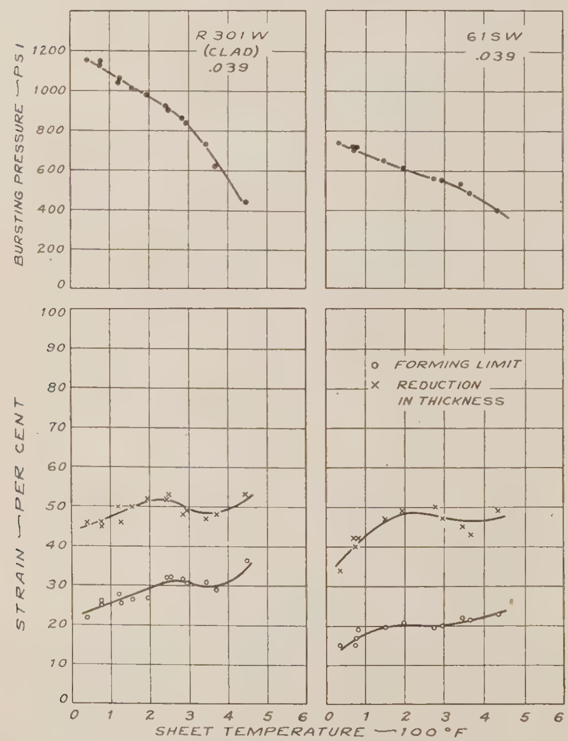


FIG. 20 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR R301-W CLAD AND 61S-W

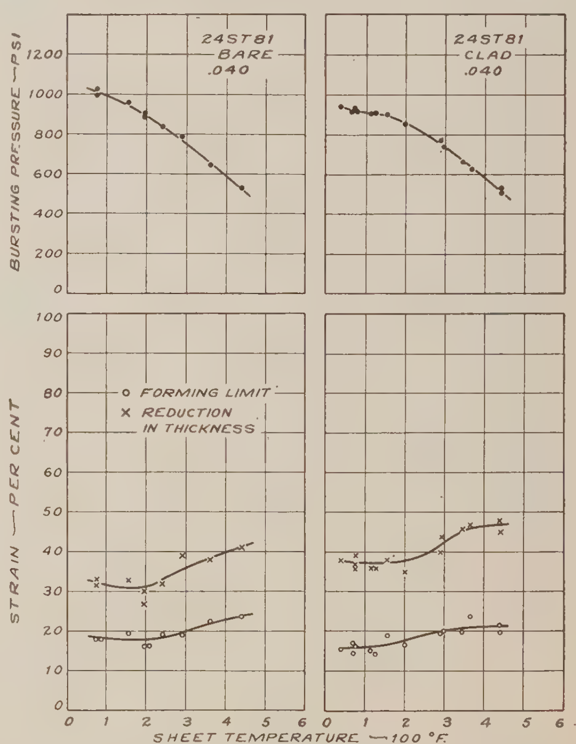


FIG. 21 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR 24S-T81, BARE AND CLAD

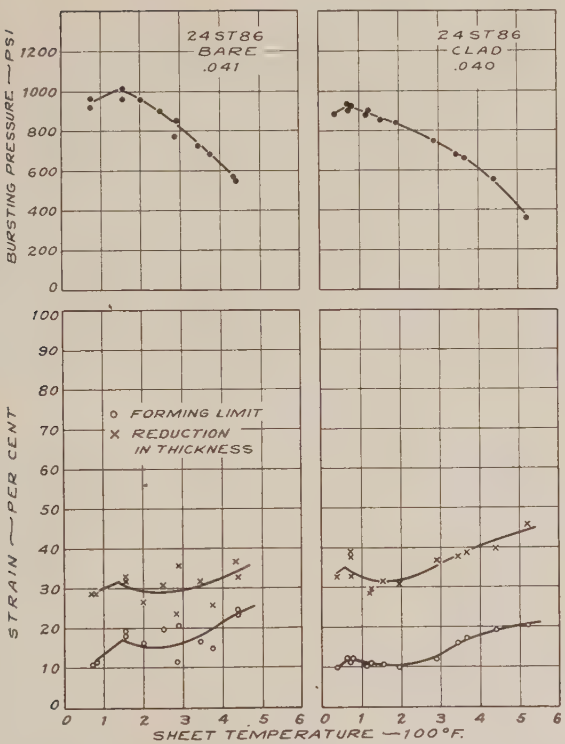


FIG. 22 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR 24S-T86, BARE AND CLAD

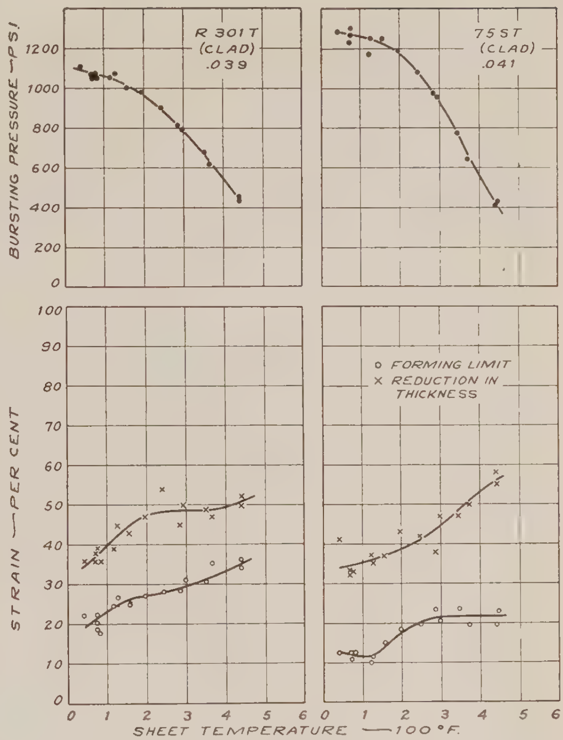


FIG. 23 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR R301-T CLAD AND 75S-T CLAD

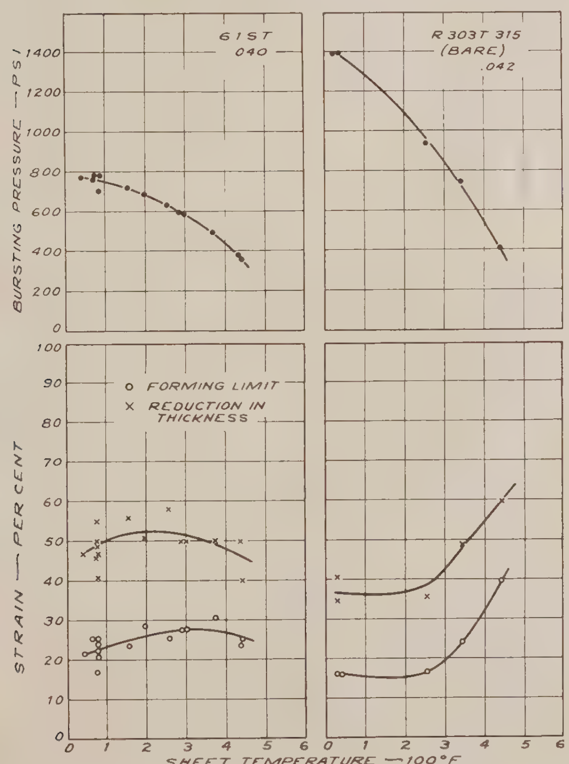


FIG. 24 RESULTS OF HOT HYDRAULIC CIRCULAR BULGING FOR 61S-T AND R303-T315 BARE

limits, the annealed alloys, Group I, may be considered together, whereas it is advantageous to discuss individually the room-temperature-aged conditions, Group II-A, the various conditions of 24S, and the artificially aged alloys Group II-B.

It appears that the forming limits of all the annealed alloys except 3S-O and 75S-O were slightly reduced by raising the temperature up to a certain limit which varied for the different alloys between 100 and 250 F (24S-O), Figs. 14 to 17, inclusive. With further increasing temperature, the forming limit of all these alloys increased continuously by as much as 35 to 40 per cent (stretch) at the testing temperature of 450 F. The magnitude of this effect probably depended considerably upon the rate of loading and was rather large for these tests because of the slow speed.

The forming limits (for the annealed conditions) at room temperature were highest for the alloys of Group I-A, 3S-O and 52S-O, and for two alloys of Group I-B, 61S-O and R3010, while 75S-O and 24S-O, both bare and clad, exhibited only approximately 70 per cent of the forming capacity of the other alloys. With increasing temperature the difference between 24S-O and the other alloys was retained, while 75S-O showed almost as high strain values as the more ductile alloys, and markedly higher strain values than 24S-O, at temperatures between 200 and 350 F. The alloys of Group II-A, Figs. 18 to 20, inclusive, aged at room temperature, were in general more responsive to the effect of temperature in the lower temperature range, but at the higher testing temperatures did not increase to definitely higher strain values than the respective artificially aged conditions.

The forming limits for the aged conditions were highest for the alloys R301-W, 61S-T, and surprisingly also R301-T, for practically the entire range of temperatures investigated.

The alloy 61S-W showed rather low strain values in comparison to the other alloys of this group, and were not appreciably higher than 24S-T.

Testing temperatures up to 250 F caused practically no change in the forming limits for all aged conditions of the alloy 24S, but slight improvement in formability was observed for all these alloys at still higher temperatures.

The lowest strain values occurred in 24S-T86 and 24S-RT, while 24S-T and 24S-T81 were intermediate.

Artificial aging of 24S-T and 24S-RT to 24S-T81 and 24S-T86, respectively, caused a surprisingly small reduction in the forming limits by not more than 1 to 2 per cent, when tested at room temperature, Figs. 18, 19, 21, and 22. However, the results at testing temperatures in excess of 200 to 300 F revealed that the artificially aged alloys have the higher forming limits after passing through a minimum at 200 F.

The cold-rolling of 24S-T clad, resulting in 24S-RT clad, Figs. 18 and 20, caused the expected reduction in strains by approximately 5 per cent, but this effect could not be recognized for 24S-RT bare, the strain values of which scattered within wide limits; with increasing temperatures this difference between the forming limits for 24S-T clad and 24S-RT clad, gradually decreased.

The alloys of Group II-B, the artificially aged conditions, Figs. 21 to 24, inclusive, were markedly affected by testing at higher temperatures, and between 300 and 400 F their forming limits exceeded those of the room-temperature-aged conditions.

The forming limit for R301-T, and to a smaller extent for R301-W, 61S-W, and 61S-T, materially increased with increasing temperature, particularly at slightly elevated temperatures. The alloys 75S-T and R303-T, possessed very low forming limits at room temperature; however, the strain values for these alloys at temperatures above 200 and 300 F, respectively, exceeded those of 24S-T. Alloy R301-T values were not appreciably different from R301-W, except that the latter showed a slight minimum in the forming-limit curve for the tests in the range 300 to 400 F. Both alloys had identical forming limits at 450 F.

Regarding the absolute strain values, some recent results show these values to be appreciably increased by increasing the sheet thickness using the same die diameter.

Reduction in Thickness at Failure

The rather large variations of the values of reduction in thickness do not permit a clear evaluation of this metal property, which measures the (local) ductility, Figs. 14 to 24, inclusive.

Of the two nonaging alloys of Group I-A, 3S-O was slightly superior and more affected by elevated temperature than was 52SO.

Of the annealed conditions of the age-hardenable alloys, Group I-B, 61S-O possessed by far the highest ductility at lower temperatures, even better than 52S-O. However, the other alloys approached such a high ductility with increasing temperatures: 75S-O above 250 F, R3010 above 400 F, and 24SO, both bare and clad, apparently above 450 to 500 F, Figs. 15 to 17, inclusive.

The forming temperature up to 450 F had only a comparatively slight effect on the ductility of the room-temperature-aged alloys, Group II-A, Figs. 18 to 20, inclusive. The reduction in thickness of the various conditions of 24ST increased continuously with increasing temperature, at a rate of 2 to 3 per cent (ductility) per 100 deg F, up to a value which was approximately $^1/_3$ higher at 450 F than at room temperature. The other two alloys investigated, 61S-W and R301W, showed a peculiar maximum in the range of 200 to 250 F, followed by a minimum at 350 F. Both of these alloys exhibited slightly higher values of reduction in thickness than 24S-T clad, over the whole range of temperatures tested, while 24S-T bare, 24S-RT clad, and 24S-RT bare, were inferior to 24S-T, by approximately 5 to 10 per cent (ductility).

Of the alloys of Group II-B, the artificially aged conditions of 24S-T were only slightly affected regarding their ductility at temperatures up to 250 F, while their ductility increased considerably between 250 and 350 F, and then approached or even exceeded that of the corresponding room-temperature-aged condition, Figs. 21 and 22. The ductility of R301-T improved markedly at temperatures slightly above room temperature, and then continued to increase at a slow rate with further increasing temperature. On the contrary, the ductility of 75S-T and R303-T appeared to increase at a steadily increasing rate, at temperatures exceeding 150 and 200 F, respectively. Both at room temperature and at approximately 400 F, the ductility of the alloys 24S-T81 clad, R301-T, 75S-T and R303-T, was approximately the same and considerably higher than that of 24S-T86 clad, 24S-T81 bare, and 24S-T86 bare. At temperatures between 200 and 300 F, however, the ductility of the alloys R301-T and 75S-T exceeded that of 24S-T81 clad, and also that of 24S-T clad.

Effects of Cladding

A comparison of the respective clad and bare aged conditions of the alloy 24S revealed a rather pronounced effect of the cladding on the forming limits and local ductility (reduction in thickness). These values were considerably more uniform for the clad than for the bare alloys. In particular, quite uniform strain values could be obtained for 24S-T clad, 24S-T81 clad, 24S-T86 clad, and 24S-RT clad, while extreme variations were encountered in 24S-RT bare, and 24S-T86 bare.

The reduction in thickness of the clad alloys was slightly larger than that of the bare conditions, on the average, by 5 per cent (0 to 10 per cent ductility).

On the contrary, the forming limits were frequently higher for the bare alloys, and very high values were occasionally encountered in both 24S-RT bare, and 24S-T86 bare. No explanation can be given for these peculiar findings. The higher stretching ability of bare alloys, however, cannot be utilized because of their larger variations, and the minimum values of the bare and clad conditions usually differed only slightly.

Alloys 24S-O clad, and 24S-O bare possessed almost identical forming limits, while the reduction in thickness was slightly higher for 24S-O clad.

As was to be expected, the bursting pressures were, on the average, approximately 10 per cent higher for the bare alloys than for the clad alloys, other factors being equal.

Types of Fracture and Defects

While most of the blanks tested failed by splitting close to the crown in the longitudinal direction, the final appearance of a bulge after the test appeared to be determined primarily by the ductility of the alloy, Fig. 1. The various types of failures observed are diagrammatically illustrated by plan views in Fig. 25. The ductile alloys generally formed several variations of single cracks, without branches, which were designated by "D-1" and "D-2." The brittle alloys ruptured in various manners, types "B-1" to "B-4" exhibiting a well-defined straight primary crack with several secondary branches which appeared to be caused by rather violent failure of the bulge.

Two variations of ductile failures were observed. The one consisted of a straight crack, running, without exception, in the longitudinal direction, type "D-1," Figs. 1 and 25. The other, type "D-2," apparently started as a short crack but continued circumferentially to one side of the crack, forming a more or less C-shaped flap rather than a straight split. While the straight split was found to occur always in the longitudinal direction, not more than $^1/_2$ in. away from the crown, the center of the flap-type crack ran longitudinally with some alloys, but with others (R301 and 75S) in various directions; however, it also approached

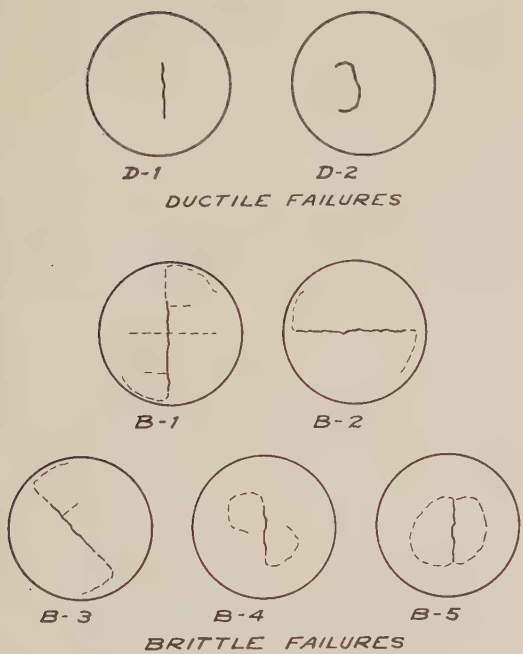


Fig. 25 Diagrammatic Sketch of Representative Types of Failures

the crown within a distance of less than $^1/_2$ in.[8] Only in three instances (24S-O bare, at 345 and 375 F, and 52S-O at 285 F), a branch in the fracture was observed for a ductile alloy.

The type of failure of the heat-treated alloys, Groups II-A and II-B, differed for the various specimens of each alloy. Most of the specimens exhibited a longitudinal primary crack, while a smaller number showed a pronounced transverse crack, and a still smaller number possessed a center crack at different angles to the rolling direction. Apparently, the more brittle the specimen, the more violent was the bursting and the more branched was the fracture.

With increasing temperature, the cracks usually became shorter and more regular, and in some instances, degenerated into a pinhole at the highest temperature used.

Rather pronounced orange peel was also encountered in the alloy 61S-W, and slight orange peel in the alloys R301-0, 75S-O, 24S-RT clad, 24S-T81 clad, 24S-T86 clad, 75S-T (at higher temperatures). The results of the investigation do not appear to be markedly affected by this condition.

The heat-treated alloys, with the exception of 61S-W and 61S-T, exhibited pronounced heat warping ("oil-canning") after the tests.

[8] Only at the highest temperatures some fractures occurred farther apart from the pole of the bulge associated with a rather irregular strain distribution. This is explained by temperature variations as previously mentioned.

Tests in which some draw-in occurred because of insufficient hold-down pressure were discarded, as were the results for a few specimens which yielded pinhole fractures because of slight irregularities in the sheet. The strain values of these specimens were generally lower than those of perfect specimens.

Little difficulty was caused by failure at the radius. A few specimens of the alloy, 24S-T86 bare broke at the radius at low pressures and exhibited a peculiar small fold extending toward the inside of the bulge into the oil. Also, one specimen of 24S-T clad failed at the radius when tested at a temperature of 525 F.

Conclusions

The pressure required to burst a sheet with a circular area exposed to gradually increasing hydraulic pressure on one side is, according to the theory of strength, a function of the tensile strength of the metal, and the thickness and curvature of the part at the moment of failure. However, as far as the effect of increasing temperature is concerned, the bursting pressure follows much the same trend as the tensile strength, according to previous investigation (3). On the contrary, the relation between different alloys is clearly different in the bursting test than in the tensile test. Of the artificially aged alloys of very high strength, only 75S-T exceeded 24S-T in bursting strength at room temperature, while 24S-RT, 24S-T81, 24S-T86, and R301-T exhibited a lower bursting strength than 24S-T. No explanation can be given for this phenomenon, at present.

The general significance of the two strain values, namely, the forming limit and reduction in thickness (area), used as metal characteristics and supplied by the tests on circular bulges, and their relation to the corresponding strain values supplied by the tensile test, is also obscure at present. In a tensile test the forming limit is the uniform strain or "infinite gage length elongation" and is entirely independent of the contraction in area at failure, or "zero gage length elongation." On the contrary, the bursting tests yielded forming limits, which always apparently paralleled the reduction in thickness (or area) at failure.

This also conforms to the fact that the forming limits of the bulges considerably exceeded the uniform tensile strain, and in most instances the conventional 2-in-gage-length elongation (3). No exception to this rule was observed in these tests. However, if the metal should possess an extremely low ductility, i.e., contraction in area, this should be exhausted in a bulge test earlier than in a tensile test, and a forming limit in bulging below that in tension could be expected.

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(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until March 11, 1946)

Discussion

In 1945, when there was a ban on national meetings, some papers originally scheduled for these meetings were presented before local groups. In the case of these papers the Commitee on Publications suspended its rule, which requires simultaneous publication of paper and discussion, and accepted discussion based on the published paper.

Acceleration Damper: Development, Design, and Some Applications[1]

B. S. Cain.[2] This interesting paper develops the mathematical theory of the acceleration damper by analytical methods. The authors wish "to give a clear step-by-step picture of the mechanism of the acceleration damper and thereby facilitate handling of transient cases, effect of rebound, and extension of theory to modified cases." A graphical method, however, seems also to be effective in these cases. If we plot the two limits of motion of the mass particle against time, then, in any steady motion, it can easily be seen that the particle stays at one limit until this passes through a point of inflection, after which the particle moves freely until it again reaches a limit.

Taking the case of sinusoidal motion, the kinetic-energy loss per cycle is easily obtained for any conditions. It can also be seen at once that the maximum damping is obtained when the free path of the particle is π times the semiamplitude of the motion, as given in the paper. This simple graphical method lends itself to a clear picture of the operation, with easy calculation of the results. It is easy to introduce the effects of any wave-shape, of transients, of rebound, and of friction.

The graphical method is particularly adapted to practical design and analysis of behavior, because the whole picture can be seen at once, and it is not necessary to use Dirac functions or even the calculus to obtain practical results. The results of graphical analysis can be used for correlation with viscous damping without difficulty. In view of the advantages of the graphical method, it would be interesting to know if the authors have used it and what disadvantages they have found in comparison with the methods which they describe.

The experimental results and the application to flutter analysis show that a simple and easily understood theory of the acceleration damper is of real value, and the authors are to be congratulated on their contribution.

Harry Frissel.[3] The authors state that theoretical calculations show the acceleration damper to be effective in raising the flutter speed, but this had not been confirmed by experiment. In an earlier paper,[4] H. Voight mentions briefly the results of model experiments using this same device in an aileron.

[1] By Paul Lieber and D. P. Jensen, published in the October, 1945, issue of Trans. A.S.M.E., vol. 67, pp. 523–530.

[2] Assistant Engineer, Locomotive Division, General Electric Company, Erie, Pa. Mem. A.S.M.E.

[3] Research Engineer, Curtiss-Wright Corporation, Airplane Division, Research Laboratory, Buffalo, N. Y.

[4] "Weitere Versuche über Tragflügelschwingungen," by H. Voight, Jahrbuch der Deutschen Luftfahrtforschung, part 1, 1938, pp. 249–258 ("Further Experiments on Wing Vibration," R.T.P. Translation no. 874).

In the latter case it was called a "disturbance damper." A plot of vibration amplitude versus dynamic pressure shows that for an aileron mass, balanced with a disturbance damper, the critical dynamic pressure is 4 times that for an aileron with damper inoperative. Also, if cork plates were used for the bottom of the container holding the balls, the critical dynamic pressure was approximately twice that of an aileron with damper inoperative. When the damper is inoperative, the vibration amplitude increases rapidly as the velocity approaches the critical value. With the damper acting, the vibrations at first commence in the same way. The balls then break away from their support when $1g$ acceleration is exceeded, and the amplitude is constant with increasing velocity. The function of the damper is to upset the phase relations necessary for flutter. When the impact energy of the damper can no longer upset the phase agreement sufficiently, the amplitude again rises sharply.

Authors' Closure

The authors appreciate that Mr. Cain has brought to their attention an interesting graphical method for determining the characteristics of the acceleration damper. It is believed that the graphical method proposed by Mr. Cain will prove especially useful when the effects of viscosity, friction, etc., on the motion of the particle are considered. It is noted that in spite of the involved mathematical treatment contained in the theory, the essential formulas used in the design of an acceleration damper for a specific application are simple algebraic equations and therefore the authors have not given consideration to graphical analysis.

The authors wish to bring to Mr. Frissel's attention that the type of acceleration damper considered in the theoretical investigation of its effect on the flutter characteristics of an airplane embodies a structural feature which is essential to its efficient operation and which is not contained in the disturbance damper described by H. Voight.[4] It is noted that the theory of the acceleration damper assumes that a gravitational or any other constant field does not act on the moving mass. Therefore this theory does not describe the behavior of a disturbance damper in which the motion of the particle perpendicular to the earth is acted upon by an acceleration of g. Since this paper has been submitted for publication, the authors have embodied in their acceleration damping device a structural means for realizing in practice a physical configuration treated by the theory. This feature ascertains the functioning of the damper and its effective operation for all accelerations however small, which is essential in order that the damper operate effectively in controlling flutter. The device described by Mr. Voight is not effective until accelerations of g or greater are attained. It is also noted that the disturbance damper cannot be designed to operate as effectively as the acceleration damper even when accelerations of g are exceeded, since the effect of the gravitational field changes the phase angle between the mass particle and the motion of the configuration which is being damped. As demonstrated by the theory, the selection of a suitable phase angle and its maintenance are especially important for the efficient operation of the acceleration damper. Herein lies one of the essential differences between the acceleration damper and the disturbance damper described.[4]

Another significant consideration embodied in the design of an acceleration damper is that the length of the free path of the mov-

ing mass is definite and is determined by Equation [13] of the paper. This dimension determines the phase angle such that the damper can operate at maximum efficiency. This design criterion is not included in the description of the disturbance damper given by Mr. Voight.

Figs. 4, 5, and 6, in which the effect of the gravitational field is not considered, were intended to describe the structure of the acceleration damper schematically.

The authors regret that this was not given more emphasis in the original paper.

Jet Propulsion and Rockets for Assisted Take-Off[1]

By M. J. ZUCROW,[2] AZUSA, CALIF.

This paper discusses the two kinds of jets which may be used for propelling bodies through the atmosphere: (a) Highly heated compressed atmospheric air admixed with products of combustion formed by burning a fuel in the air; the thermal energy of the fuel being employed to raise the air temperature. This type is known as a "thermal-jet engine." (b) A jet formed by generating large quantities of gases by chemical reaction. The equipment in which such a jet is produced is known as a "rocket motor." Operating principles and estimated performance characteristics of thermal-jet engines are treated, and certain unrestricted information on the use of rocket motors or "Jato" units for assisted take-off of aircraft is presented.

Nomenclature

The following nomenclature is used in the paper:

T = propulsion thrust, lb
G = weight flow, lb per sec
g = acceleration due to gravity, ft per sec per sec
w = relative exit velocity of jet, fps
V = flight speed, fps
c = absolute exit velocity of jet, fps
a = acoustic velocity, fps
$m = \frac{G}{g}$ = mass flow, slug per sec
P_T = thrust power (TV), ft-lb per sec
P_L = exit loss, ft-lb per sec
P = propulsion power, ft-lb per sec
c_p = specific heat, Btu per lb per deg F
D = diameter, ft
$\mathbf{M}$ = Mach number
T_1 = temperature of entering air, deg R
T_2 = temperature at entrance to compressor, deg R
T_3 = temperature at end of isentropic compression, deg R
T_4 = temperature at end of actual compression, deg R
T_5 = temperature of air entering turbine, deg R
T_6 = temperature at end of isentropic expansion from turbine inlet pressure to turbine back pressure, deg R
T_7 = temperature at end of actual expansion in turbine, deg R
T_9 = temperature at end of isentropic expansion from turbine back pressure to atmospheric pressure, deg R
$\phi = T_2/T_1 = 1 + 0.2\mathbf{M}^2$ for air
$\theta = (P_3/P_2)^{\frac{k-1}{k}} = T_3/T_2$
$\alpha = T_5/T_1$
$\nu = V/w$ = velocity ratio
ρ = density, slug per cu ft
η = over-all efficiency = $\eta_i\eta_P$
η_i = internal efficiency
η_P = propulsion efficiency
η_c = compressor efficiency
η_t = turbine efficiency

$$\Sigma = \left(1 - \phi\frac{\theta - 1}{\eta_r\eta_t\alpha} - \frac{1}{\phi\theta}\right)\left[\frac{1 - \phi\dfrac{\theta - 1}{\eta_c\alpha}}{1 - \phi\dfrac{\theta - 1}{\eta_r\eta_t\alpha}}\right]$$

Subscripts:

a = air
f = fuel

Introduction

In its broadest sense, the term jet propulsion refers to the art of propelling a body by the reaction or thrust of a fluid jet. There is no limitation, in general, upon the conceivable means which may be employed to produce the jet. Furthermore, the working fluid may be a liquid, vapor, heated air, the gaseous products of a chemical reaction, or combinations of these. However, when it comes to propelling bodies through the atmosphere by jet propulsion, the jet may be one of two kinds.

1 There is the jet consisting of highly heated, compressed, atmospheric air admixed with the products of the combustion formed by burning a fuel in the air; the thermal energy of the fuel being employed to raise the air temperature. A jet of this type will be termed a "thermal jet," and the equipment utilized for producing it will be termed a "thermal-jet engine."

2 There is also the jet formed by generating large quantities of gases by a chemical reaction. These gases are usually produced by reacting a fuel with a chemical oxidizer, but reducing reactions are not ruled out. The significant thing is that atmospheric oxygen does not enter into the chemical reaction, which is therefore independent of the atmospheric surroundings. A jet produced in this manner will be termed a "rocket jet," and the equipment wherein the chemical reaction takes place, including the discharge nozzle, will be called a "rocket motor."

This paper discusses certain basic features of jet propulsion as a mode of propelling bodies through the air. In this connection, the comments on thermal-jet engines will be restricted to operating principles and estimated performance characteristics. Another purpose of this paper is to present certain unrestricted information on the rocket motors used for the assisted take-off of aircraft, commonly referred to as "Jato" units.

Reaction and Momentum Principles

The fundamental operating principle of any jet-propulsion device is the "action-equals-reaction" principle known as Newton's third law of motion. This states: "To every action there is an equal and opposite reaction." This principle is exceedingly common in Nature, and is so well known to engineers that it merits little discussion here. It should be noted, however, that where motion of a body through a fluid medium is concerned, the

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[1] Based on parts of Chapters 4 and 8 of the book, "Principles of Jet Propulsion," by the author, to be published in 1946, by John Wiley & Sons, Inc., New York, N. Y.

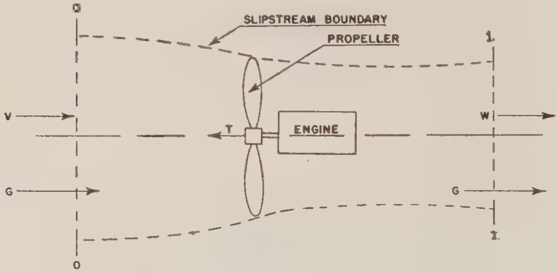
[2] Aerojet Engineering Corporation. Mem. A.S.M.E.

Contributed by the Aviation Division and presented at the Aviation War Conference, Los Angeles, Calif., June 11–14, 1945, of The American Society of Mechanical Engineers.

reaction principle predominates.[3] Thus the screw employed for propelling a ship, the propeller for moving an airplane, and jet propulsion all operate by virtue of this principle. In every one of these cases a fluid is accelerated in the direction opposite to that of the desired motion for the body, and the reaction due to the acceleration produces a propulsion force or thrust in the direction of motion. The magnitude of the propulsion force is determined by applying the momentum principle, "the rate of change of the momentum of a bounded mass system of discrete particles (body of fluid) is equal to the sum of the external forces acting on the system."[4]

Thrust Equations for Propulsion Systems

Fig. 1 illustrates the application of the foregoing principles to the engine-driven propeller. Since the air reactions on a body



(1) MOMENTUM ENTERING SECTION O–O

$$M_0 = \frac{G}{g} V = mV$$

(2) MOMENTUM LEAVING SECTION 1–1

$$M_1 = \frac{G}{g} w$$

(3) CHANGE IN MOMENTUM PER. SEC.

$$T = M_1 - M_0 = \frac{G}{g}(w - V) \text{ LB}$$

AND

$$\frac{T}{G} = \frac{w}{g}\left(1 - \frac{V}{w}\right) = \frac{w}{g}(1 - \nu)$$

(4) THRUST POWER

$$P_T = TV = \frac{G}{g} w^2 (1 - \nu)\nu$$

Fig. 1 Momentum Principle Applied to Propeller

depend only upon the relative velocity between it and the air, the body may be assumed to be stationary and the air to approach it with the actual speed of the body. In the ideal case the propeller creates a slip stream as indicated in the figure. The air enters the slip stream with the linear velocity V feet per second relative to the propeller and leaves it with the exit relative velocity w feet per second. If the air flow through the slip stream is G pounds per second, then the thrust developed is

$$T = \frac{G}{g}(w - V) \qquad [1]$$

It will be seen that the difference between the characteristics of propeller propulsion and jet propulsion is not as great as might be assumed. Thus if the propeller and engine are enclosed in a duct, so that the propeller acts as a blower or compressor, and the air of the slip stream is heated by burning a fuel in it, the resulting thermal-jet engine is that originally ascribed to S. Campini. This arrangement is illustrated in Fig. 2. The essential difference between this system and the propeller is that the relative

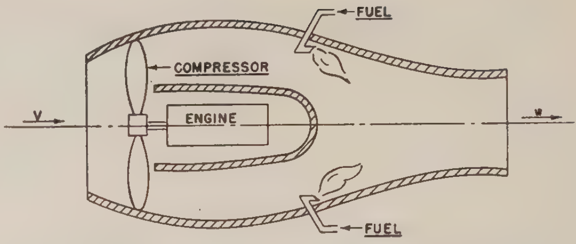
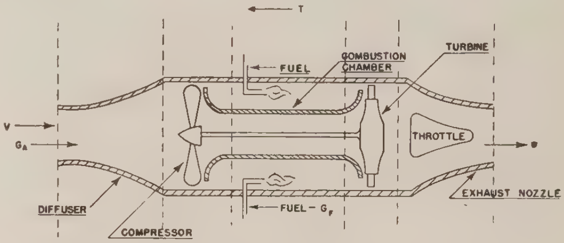


Fig. 2 Momentum Principle Applied to Campini System



$$T = \text{THRUST} = \frac{G_A}{g}(w - V) + \frac{G_F}{g} w \approx \frac{G}{g}(w - V)$$

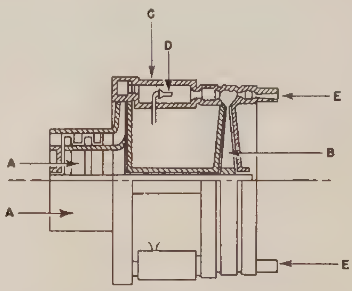
$$\frac{T}{G} = \text{THRUST/LB. OF AIRFLOW} = \frac{w}{g}\left(1 - \frac{V}{w}\right) = \frac{w}{g}(1 - \nu)$$

$$P_T = TV = \frac{G}{g} w^2 (1 - \nu)\nu$$

Fig. 3 Momentum Principle Applied to Thermal-Jet Engine

exit velocity of the gases (mainly air) has been increased by virtue of the addition of thermal energy. The thrust equation has the same form, but the thrust per unit mass of air flow is larger owing to the increased exit velocity. It should be noted, however, that the absolute exit velocity of the gases $c = (w - V)$ has also been increased so that the kinetic energy lost in the exit gases is also larger.

Fig. 3 illustrates the type of thermal-jet engine currently the most popular. The major difference between this and the Campini system is that the air compressor is driven by a gas turbine instead of an internal-combustion engine. The patent literature discloses many different arrangements for this type of thermal-jet engine. Their apparent differences, however, are not basic, but are confined to such features as the type of air compressor, the type of turbine, the combustion-chamber arrangement, and construction details. All of them have the same objective, i.e., to compress air, heat it, drive a turbine, and



A = AIR COMPRESSOR
B = TURBINE
C = COMBUSTION CHAMBER
D = FUEL SPRAY NOZZLE
E = EXHAUST NOZZLE (JET)

Fig. 4 Patent Application of Air Commodore Frank Whittle
(British Patent No. 347,206; filed January 16, 1930.)

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[3] "Raketenflugtechnik," by E. Sänger, R. Oldenbourg, Munich, Germany, 1935.

[4] "Fundamentals of Hydro- and Aeromechanics," by L. Prandtl and O. G. Tietjens, McGraw-Hill Book Company, Inc., New York, N. Y., 1944, p. 233.

discharge heated gases through some form of nozzle. The most successful pioneer in this field has been Air Commodore Frank Whittle of the R.A.F. It is interesting to note that his British Patent Application, No. 347,206, filed January 16, 1930, illustrated in Fig. 4, embraced all of the necessary elements.[5] For a review of the patent literature, the reader is referred to a text by G. Geoffrey Smith.[6]

It is seen by reference to Fig. 3 that the equation for the thrust developed by this type of thermal-jet engine is given by

$$T = \frac{G_a}{g}(w - V) + \frac{G_f}{g}w \qquad [2]$$

Let $G = G_a + G_f$ which is approximately equal to G_a; since G_f can be neglected because it is ordinarily less than 3 per cent of G_a. The equation for the thermal-jet engine then becomes

$$T = \frac{G}{g}(w - V)$$

which is identical with Equation [1].

The thrust equations for the propeller and thermal-jet propulsion systems can be transformed to show the relation between the thrust and the velocity ratio ν; the latter being the ratio of the velocity of the moving body (V) to the exit relative velocity of the fluid (w). Thus for these two propulsion systems

$$T = \frac{G}{g}w(1 - \nu) \qquad [3]$$

or the thrust per pound of air flow per second is

$$\frac{T}{G} = \frac{w}{g}(1 - \nu) \qquad [4]$$

Equation [4] indicates how thrust developed, per pound of fluid flowing through the propulsion system per second, varies with the exit velocity (w) and the velocity ratio (ν).

It is seen that as the velocity ratio approaches unity, the thrust per pound of air flow per second approaches zero. Consequently, to develop thrust under conditions close to unity velocity ratio, the quantity of working fluid required becomes very large; hence a large exit area would be needed for passing the fluid. For a fixed velocity ratio the size of the required exit area increases as the speed of the airplane decreases; hence the propeller, with its ability to handle extremely large quantities of air at velocity ratios close to unity, is well adapted to the propulsion of bodies at moderate and low speeds. This is particularly true since, as will be seen later, operation close to unity velocity ratio gives high propulsion efficiency.

Fig. 5 is a plot of the thrust per pound of air flow per second as a function of the velocity ratio for different flight speeds. In the same figure is plotted the ideal propulsion efficiency as a function of the velocity ratio.

Fig. 6 illustrates the jet-propulsion action of the rocket system. The arrangement illustrated utilizes the chemical reaction between a liquid fuel and a liquid oxidizer; each liquid is termed a propellant. The propellants are removed from the supply tanks at the constant rate of G pounds per second. They may be, for example, gasoline and a sufficient quantity of liquid oxygen to burn it completely. At (1) the pressure of the liquids is increased by a suitable pressurizing means, and at (2) the propellants are injected into the combustion chamber. As the propellants flow through the combustion chamber they react at substantially constant pressure, producing large quantities of hot combustion gases. The combustion is completed by the time the gases reach the entrance to the exhaust nozzle through which they expand adiabatically and from which the gases are discharged with supersonic velocity.

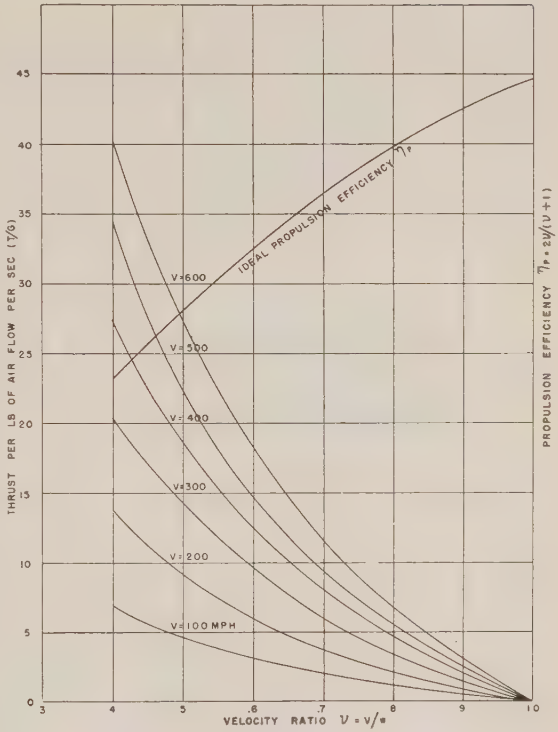

FIG. 5 THRUST PER POUND OF AIR FLOW PER SECOND AS A FUNCTION OF THE VELOCITY RATIO

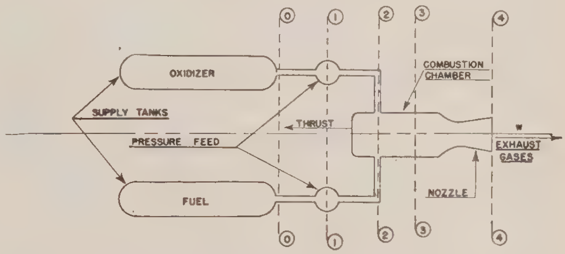

FIG. 6 JET-PROPULSION ACTION OF LIQUID ROCKET SYSTEM

If the area ratio for the nozzle is selected so that the gases are expanded to the pressure of the atmosphere surrounding the nozzle-exit section, then with a constant pressure in the combustion chamber, the thrust is given by

$$T = \frac{G}{g}w = mw \qquad [5]$$

where w is the velocity of the exhaust gases relative to the nozzle, and is called the "effective exhaust velocity."[7]

[5] Thirty-second Wilbur Wright Memorial Lecture, by Sir A. H. Roy Fedden, *Journal of The Royal Aeronautical Society*, London, Eng., May 26, 1944.

[6] "Gas Turbines and Jet Propulsion for Aircraft," by G. G. Smith, published in the United States by Aerosphere, Inc., 370 Lexington Ave., New York, N. Y., 1944.

[7] "Characteristics of the Rocket Motor Unit Based on the Theory of Perfect Gases," by F. J. Malina, *Journal of The Franklin Institute*, vol. 230, 1940, pp. 433–454.

It should be noted that whereas the thrust developed by the propeller and the thermal-jet engine depends upon the difference between the relative exit velocity of the gases and the speed of the propelled body, the thrust of the rocket motor is independent of the motion of the rocket system; it depends only upon the effective exhaust velocity of the rocket jet.

Fig. 7 illustrates a rocket motor utilizing a solid propellant, fuel plus oxidizer. This is the type of rocket motor used most extensively for Jato. To obtain the desired action the propellant characteristics must be such that the propellant burns at a

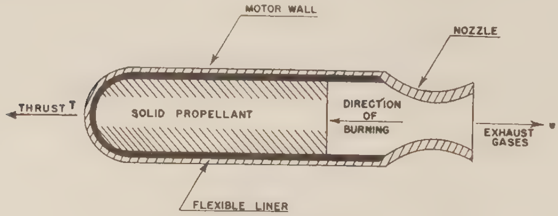

Fig. 7 Jet-Propulsion Action of Solid-Propellant Rocket Motor

uniform rate in layers perpendicular to the motor axis, that is, the burning is restricted to take place in one direction. One way of accomplishing this is by bonding the propellant stick and the steel walls of the motor with a special flexible sealing material, or liner, which allows the propellant stick the freedom of motion it requires for adapting its shape to the deformations produced by the gas pressure, thermal expansion, and internal stresses. The thrust equation for the solid-propellant rocket motor is, of course, identical with that for the liquid-propellant system.

Propulsion Power and Propulsion Efficiency

By propulsion power is meant the total power furnished the propulsion system. A portion of this power is converted into the thrust power required to maintain motion; the thrust power being the product of the thrust developed and the speed of the propelled body. The difference between the propulsion and thrust powers depends on the energy losses associated with the propulsion system. The ratio of the thrust power to the propulsion power is termed the propulsion efficiency and is denoted by η_p. Obviously, to obtain high values of propulsion efficiency the energy losses must be kept small.

The propulsion power P of the engine-driven propeller is the power supplied to it by the engine. In the thermal-jet engine the propulsion power is derived from the thermal energy of the fuel burned in the combustion chamber. For the rocket motor, the propulsion power comes from the heat of reaction of the fuel with the particular oxidizer employed. For all of these systems the energy loss, in the ideal case, is that associated with the kinetic energy of the fluid being ejected from the propulsion system. This is called the leaving or exit loss P_L and is given by

$$P_L = {}^1/_2 \frac{G}{g} c^2 = {}^1/_2 \frac{G}{g} (w - V)^2 \ldots\ldots\ldots [6]$$

Obviously, the prerequisite for a small exit loss is a low value for the absolute exit velocity of the propulsion fluid.

For all three propulsion systems, if it is assumed that the only energy loss is the exit loss, the propulsion power must be equal to the thrust power TV, plus the exit loss P_L. The ideal propulsion efficiency for each system is given by a general equation of the form

$$\eta_P = \frac{TV}{P} = \frac{TV}{TV + P_L} \ldots\ldots\ldots\ldots [7]$$

By substituting for the thrust and the exit loss from the preceding equations, the ideal propulsion-efficiency equations for each system are obtained; thus

(a) *Ideal Propeller:*[8]

$$\eta_P = \frac{TV}{P} = \frac{2V}{V + w} = \frac{2\nu}{\nu + 1} \ldots\ldots\ldots\ldots [8]$$

(b) *Ideal Thermal-Jet Engine:*

$$\eta_P = \frac{2V\,[(G_a + G_f)\,w - G_a V]}{(G_a + G_f)w^2 - (G_a - G_f)V^2}$$

If G_f is neglected so that $G = G_a \mp G_f$, then

$$\eta_P = \frac{2\nu}{\nu + 1}$$

(c) *Rocket Motor:*

$$\eta_P = \frac{TV}{TV + \frac{G}{2g}(w - V)^2} = \frac{2V}{w^2 + V^2} = \frac{2\nu}{1 + \nu^2} \ldots [9]$$

It is seen from the foregoing that the propeller and the thermal-jet engine have identical ideal propulsion-efficiency equations and that their efficiencies depend upon the velocity ratio. It should be noted, however, that for these propulsion devices operation at unity velocity ratio is impossible because the thrust then falls to zero. For these two propulsion systems the relation between the thrust power and the velocity ratio is

$$P_T = TV = \frac{G}{g} w^2 (1 - \nu)\nu \ldots\ldots\ldots [10]$$

This equation shows that for a constant rate of fluid flow G, the thrust power is a quadratic function of the velocity ratio. The maximum value for the thrust power is obtained under these conditions when the velocity ratio is $\nu = 0.5$. The corresponding value of the propulsion efficiency is then $\eta_P = 0.667$. It is seen therefore that the velocity ratio for maximum thrust power and maximum propulsion efficiency are different.

In the case of the rocket jet, the thrust depends directly on the exhaust-gas velocity w. Consequently, operation at unity velocity ratio or higher is a possibility. Thus assume that w = 6000 fps, then at unity velocity ratio the body propelled by the rocket-jet motor must be traveling at a speed of 4100 mph. Fig. 8 shows the effect of flight speed on the propulsion efficiency of a rocket jet, based on w = 6000 fps. Obviously, the rocket jet is an extremely inefficient propulsion device at travel speeds comparable to those attained by the fastest fighter aircraft.

Fig. 9 presents the estimated propulsion-efficiency curves for the propeller, thermal jet, and rocket jet at 20,000 ft altitude.[9] The propeller and thermal-jet curves intersecting at 550 mph indicate that the propeller is the more efficient at speeds below that value. These two efficiency curves intersect at lower speeds with increasing altitude. At 30,000 ft, for example, the intersection point is at 450 mph. The rocket jet has a very low propulsion efficiency except at extremely high speeds of travel.

The over-all efficiency η, of any of these propulsion systems is, of course, the product of the thermal efficiency of the power plant (or the internal efficiency η_i) and the propulsion efficiency

[8] "The Elements of Aerofoil and Airscrew Theory," by H. Glauert, The Macmillan Company, New York, N. Y., 1943, p. 203.

[9] "Aircraft Power Plant; Past and Future," by Sir A. H. Roy Fedden, *Journal of The Royal Aeronautical Society*, vol. 48, 1944, pp. 443–457.

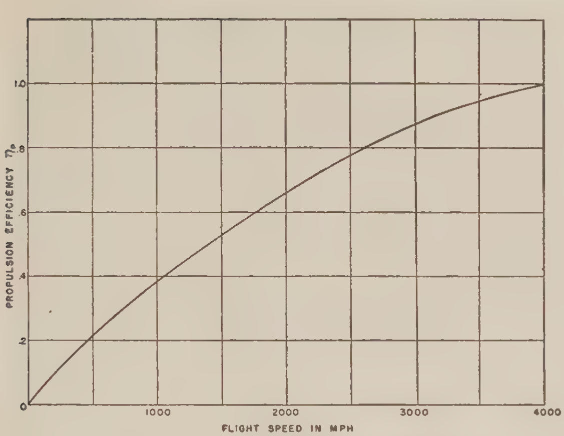

FIG. 8 EFFECT OF FLIGHT SPEED ON PROPULSION EFFICIENCY OF ROCKET MOTOR
(Based on w = 6000 fps.)

η_P); thus $\eta = \eta_i\eta_P$. Table 1 summarizes the relationships just discussed.

TABLE 1 THRUST AND POWER RELATIONSHIPS

Propeller and thermal jet

$$\text{Thrust} = T = \frac{G}{g}(w - V) = \frac{G}{g}w\left(1 - \frac{V}{w}\right)$$

$$\frac{\text{Thrust}}{\text{Pounds fluid flow per second}} = \frac{T}{G} = \frac{w}{g}\left(1 - \frac{V}{w}\right) = \frac{w}{g}(1 - \nu)$$

$$\text{Exit loss} = P_L = \frac{G}{2g}c^2 = \frac{G}{2g}(w - V)^2$$

$$\text{Thrust power} = TV = P_T$$

$$P_T = \frac{G}{g}w^2\left(1 - \frac{V}{w}\right)\frac{V}{w} = \frac{G}{g}w^2(1 - \nu)\nu$$

$$\text{Propulsion power} = P = TV + P_L$$

$$P = \frac{G}{2g}(w^2 - V^2) = \frac{G}{2g}w^2\left[1 - \left(\frac{V}{w}\right)^2\right]$$

$$\text{Ideal propulsion efficiency} = \eta_P$$

$$\eta_P = \frac{TV}{P} = \frac{TV}{P_T + P_L} = \frac{TV}{TV + P_L} = \frac{2V}{w + V} = \frac{2(V/w)}{1 + (V/w)} = \frac{2\nu}{1 + \nu}$$

Rocket jet

$$T = \frac{G}{g}w$$

$$\frac{T}{G} = \frac{w}{g}$$

$$P_L = \frac{G}{g}c^2 = \frac{G}{2g}(w - V)^2$$

$$P_T = TV$$

$$P = TV + P_L$$

$$\eta_P = \frac{TV}{TV + P_L} = \frac{2(V/w)}{1 - (V/w)^2} = \frac{2\nu}{1 + \nu^2}$$

G = fluid flow, lb per sec
V = true air speed, fps
w = relative exit velocity of propulsion fluid, fps

PROPULSION POWER AND DISCHARGE AREA

Fig. 10 is a plot of the so-called propulsion parameter $V\sqrt[3]{D^2\rho/P}$, as a function of the ideal propulsion efficiency. This curve is applicable to the engine-driven propeller and the thermal-jet engine. When applied to the propeller, D is its diameter, ρ is the atmospheric density,[10] V is the airplane speed, and P is the power furnished by the engine. For the thermal-jet engine, D is the diameter of the discharge jet, ρ is the density of the jet gases, and the propulsion power P is calculated from the increase in the kinetic energy of the gases entering and leaving the system; thus

$$P = \frac{G}{2g}(w^2 - V^2)$$

[10] It has been assumed that the density of the air is increased 15 per cent in passing through the actuator disk.

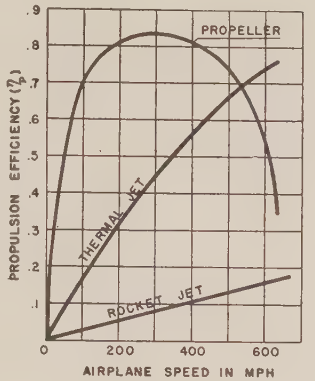

FIG. 9 COMPARISON OF PROPULSION EFFICIENCIES OF PROPELLER, THERMAL JET, AND ROCKET

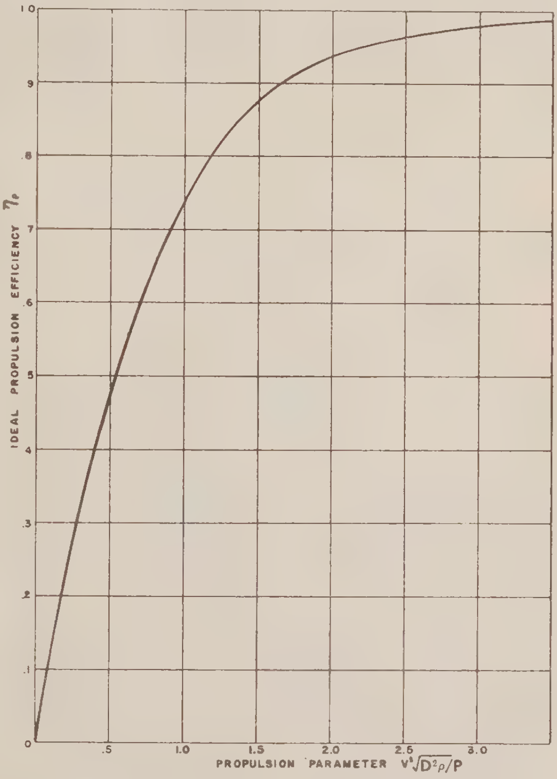

FIG. 10 PROPULSION EFFICIENCY VERSUS PROPULSION PARAMETER FOR PROPELLER AND THERMAL JET

The equations relating the propulsion parameter and the ideal propulsion efficiency are presented for reference as follows:

Propeller[11] *Thermal Jet*[12]

$$V\sqrt[3]{D^2\rho/P} = \frac{0.82\eta_P}{\sqrt[3]{1 - \eta_P}} \approx \frac{1.364\,\eta_P}{\sqrt[3]{8 - 12\eta_P + 4(\eta_P)^2}} \quad [11]$$

[11] Reference 8, p. 204.
[12] "The Modern Gas Turbine," by T. Sawyer, Prentice Hall, Inc., New York, N. Y., 1945, p. 178.

The curve gives the maximum attainable propulsion efficiency for either type of propulsion system. It is seen that to obtain high values of efficiency with low airplane speeds, D must be large, assuming the other factors constant. This operating condition is most readily met with propeller propulsion.

When flight speeds in excess of 500 mph are considered, there arise problems connected with the engine size, as well as those due to the effect of compressibility of the air upon the propeller efficiency, for the power requirements increase as the cube of the airplane speed. Thus if it takes 1200 hp to propel a given airplane at 360 mph at a given altitude, it would require an engine developing approximately 3200 hp to propel it at 500 mph, assuming all other factors remained unchanged. This calls for a heavy and complex power plant, even if this power can be furnished by a single engine. On the other hand, the thrust power for attaining this speed can be obtained from a thermal-jet engine of relatively light weight, with a jet diameter of 1 ft and an exhaust velocity of approximately 1500 fps.

Reciprocating internal-combustion engines for large power outputs require a multiplicity of accessories which result in an increase in their specific weights. Furthermore, they introduce complicated installation and maintenance problems. The probable trend in the specific weight of large internal-combustion engines, as estimated by Sir A. H. Roy Fedden, is presented in Fig. 11.

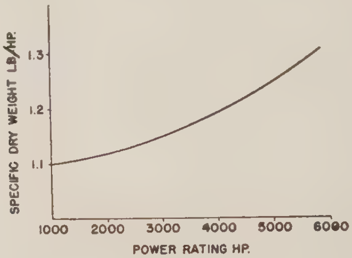

Fig. 11 Trend in Reciprocating-Engine Weights With Power Rating

The efficiency of an actual propeller will be approximately 0.85 of the ideal propulsion efficiency, or 0.833. This is due to various factors neglected by the momentum theory, such as whirl energy imparted to the slip stream, nonuniformity of the thrust distribution over the blades, blade-profile drag, and periodicity of the flow.

It was shown previously that to obtain the maximum propulsion power from a thermal-jet engine, the velocity ratio should be approximately 0.5. The propulsion efficiency for this condition is 0.667. Assuming the operating data used for the propeller previously discussed, and the discharge temperature for the jet at 1200 deg R, so that $\rho = 0.00102$ slug per cu ft, then the propulsion efficiency is about 68 per cent of that obtainable with an ideal propeller; the further assumption being made that the propeller is unaffected by compressibility. To obtain the same ideal propulsion efficiency as that for the propeller, the velocity ratio must be decreased, with the consequence that the required diameter of the jet becomes very large.

Thermodynamic Analysis of Thermal-Jet Engine

The analysis presented here is based on the following simplified assumptions:

(*a*) There is full recovery of the dynamic pressure of the entering air (100 per cent ram).

(*b*) There is no change in the mass rate of flow of fluid passing through the engine.

(*c*) There is no change in the chemical composition of the working fluid, it being assumed to be air at all sections of the flow path.

(*d*) The working fluid (air) behaves in accordance with the laws of perfect gases; the effect of temperature on the specific heat of the air is neglected.

The inaccuracies introduced by the foregoing assumptions are so small that for the practical purpose of obtaining a general insight into the behavior of the thermal-jet engine they are well justified. The advantage of this form of analysis is that general equations for the various performance characteristics can be derived.

Fig. 12 illustrates the thermodynamic process involved on the temperature-entropy (T-S) plane. The processes have been explained in the foregoing and are denoted on the diagram.

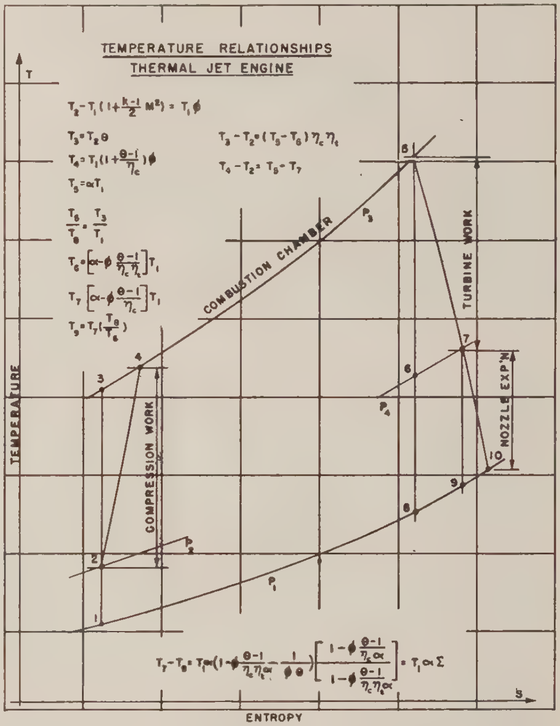

Fig. 12 Thermal-Jet Engine Cycle on Temperature-Entropy Plane

The object of the analysis is to obtain equations for the temperatures at the state points indicated on the diagram in terms of the temperature of the entering air and its Mach number. These equations are listed in the diagram. From them the equation for the temperature difference ($T_7 - T_9$), corresponding to an isentropic expansion through the nozzle, is readily obtained. Fig. 13 presents values of this temperature difference as a function of Mach number for five different altitudes ranging from sea level to 40,000 ft.

Since the temperature difference ($T_7 - T_9$) is proportional to the isentropic enthalpy change for the expansion in the nozzle, the exhaust velocity of the jet is readily calculated; thus

$$w = 223.7\sqrt{c_p\,(T_7 - T_9)} \times \text{nozzle coefficient} \quad \ldots. \quad [12]$$

where $c_p = 0.24$ Btu per lb per deg F.

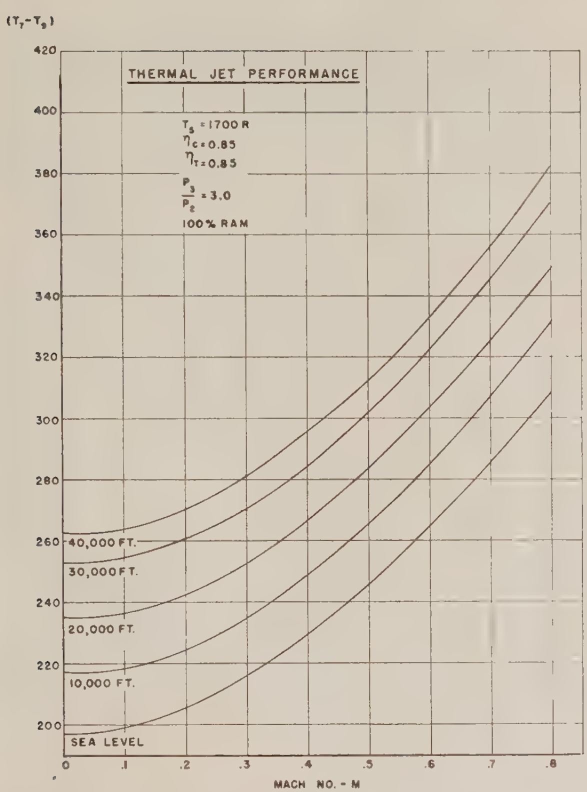

FIG. 13 TEMPERATURE DIFFERENCE (T_7 — T_9) AS A FUNCTION OF MACH NUMBER

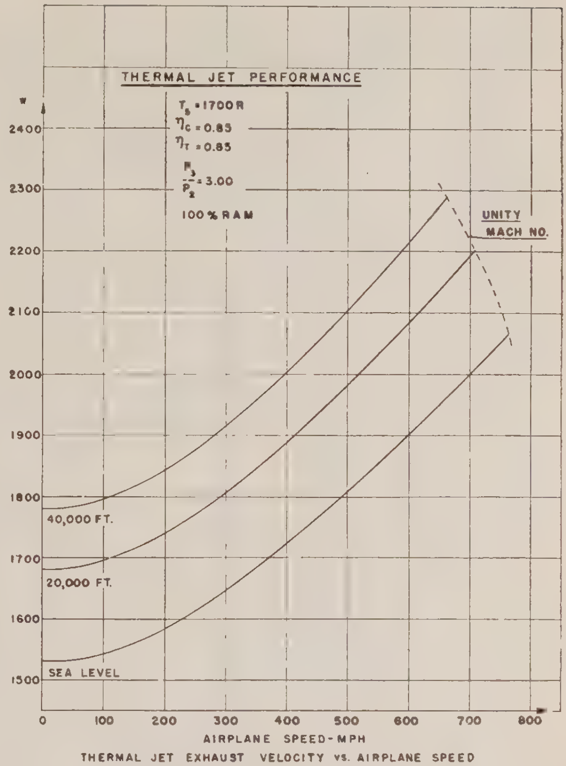

FIG. 14 THERMAL-JET VELOCITY VERSUS TRUE AIR SPEED IN MILES PER HOUR

In calculating the performance characteristics the nozzle velocity coefficient was assumed to be unity.

PERFORMANCE CHARACTERISTICS OF THERMAL-JET ENGINE

Fig. 14 illustrates how the jet velocity varies with the true air speed, in miles per hour, at different altitudes. The rapid increase in jet velocity with higher air speed is due to full recovery of the dynamic pressure of the air. Since the thrust developed by the engine depends upon the velocity difference (w — V), it is important to design the intake system so that the ram recovery is high.

Fig. 15 illustrates how the weight rate of air flow through the system varies with Mach number, with the compressor running at constant speed.

Fig. 16 shows the effect of true air speed on the thrust and the fuel consumption per pound of thrust. Assuming that the fuel has a calorific value of 18,000 Btu per lb, it is seen that the ratio of the static thrust to flight thrust is considerably less than it is for a variable-pitch, constant-speed, engine-driven propeller. As a result of the ram effect, the thrust increases at higher air speeds after passing through a minimum point. The fuel consumption per pound of thrust increases with flight velocity at all altitudes. Its rate of increase, however, diminishes somewhat with increasing altitude at speeds in excess of 250 mph.

Fig. 17 compares the thrust characteristics of the adjustable-pitch propeller with that of a thermal-jet engine. The basis of comparison is the fact that both propulsion systems provide equal thrusts at 375 mph.[13]

[13] "Gas Turbines in Aircraft," by C. D. Flagle and F. W. Godsey, Jr., *Western Flying*, June, 1945, p. 58.

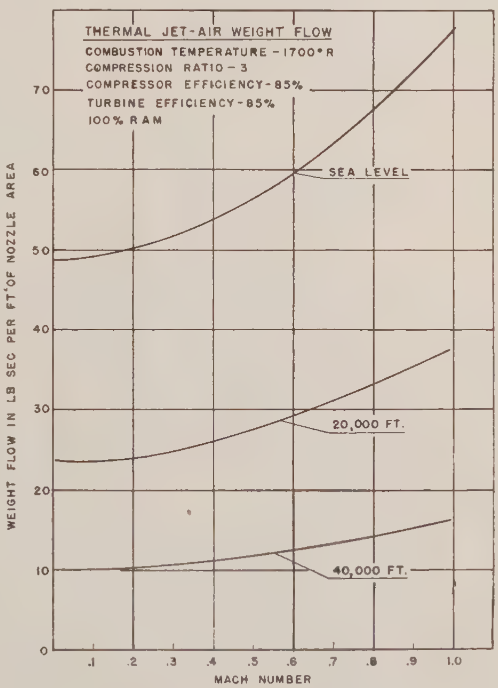

FIG. 15 WEIGHT RATE OF AIR FLOW VERSUS MACH NUMBER

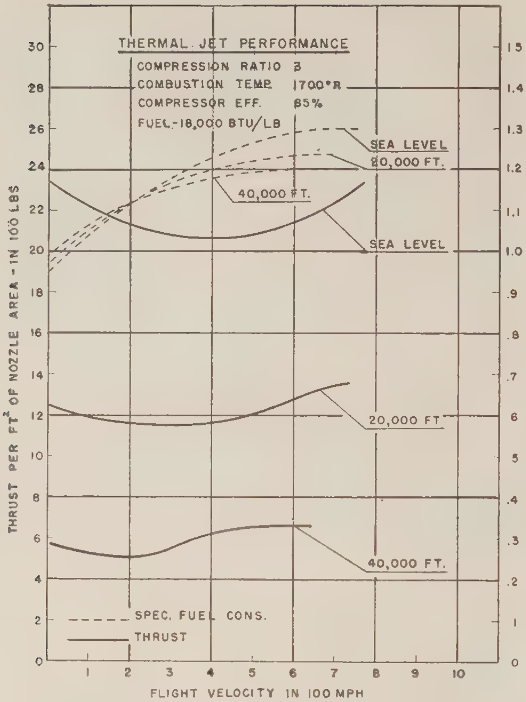

Fig. 16 Thrust and Fuel Consumption of Thermal-Jet Engine Versus True Air Speed

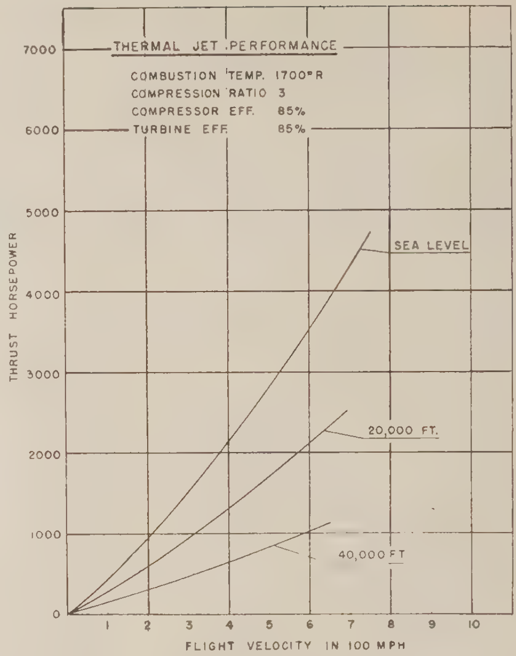

Fig. 18 Thrust Horsepower of Thermal-Jet Engine Versus Flight Speed

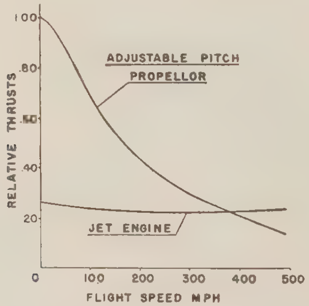

Fig. 17 Comparison of Propeller and Thermal-Jet Thrusts Versus True Air Speed

Fig. 18 shows how the thrust horsepower developed by the thermal jet varies with flight velocity and altitude.

Fig. 19 shows how increasing temperature of air leaving the combustion chamber affects static thrust and fuel consumption per pound of thrust. It is seen that raising the temperature greatly increases the static thrust at sea level but this is only at the expense of a slightly higher rate of fuel consumption per pound of thrust.

In the present state of the metallurgical art, operating temperatures as high as 1750 F are possible. If the present rate of improvement of refractory metals continues, operating temperatures as high as 2000 F appear to be probable.[14] As in the case of

[14] "The Gas Turbine in Aviation—Its Past and Future," by S. R. Puffer and J. S. Alford, presented at the Aviation War Conference, Los Angeles, Calif., June 11–14, 1945, of The American Society of Mechanical Engineers.

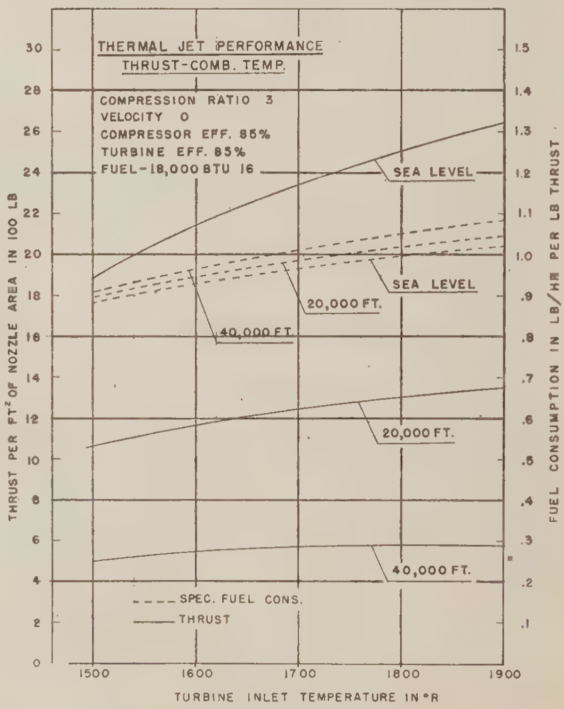

Fig. 19 Effect of Temperature on Static Thrust and Fuel Consumption of Thermal-Jet Engine

gas-turbine propulsion plants, the maximum benefits derivable from higher operating temperatures will necessitate using higher pressure ratios.[15]

Fig. 20 presents the static thrust and fuel consumption as functions of the pressure ratio of the air compressor for a constant temperature at the entrance to the turbine. Increasing the pressure ratio is effective in increasing the thrust and decreasing the fuel consumption.

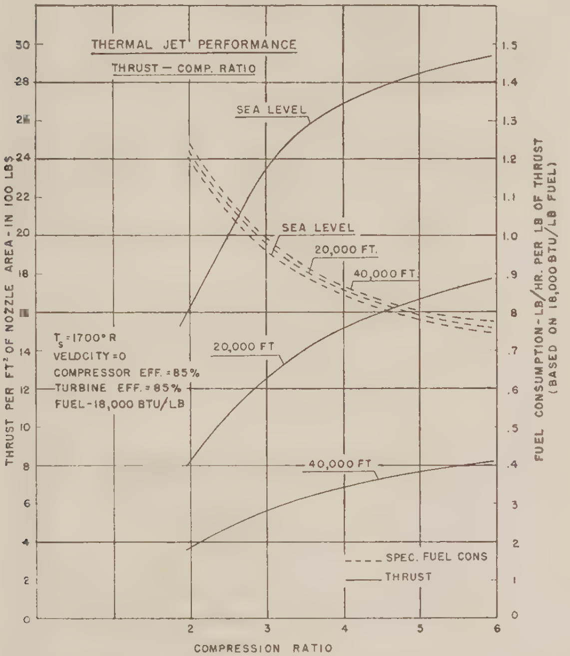

Fig. 20 Effect of Pressure Ratio on Static Thrust and Fuel Consumption of Thermal-Jet Engine

The efficiency of the unit, defined as the ratio of the thrust horsepower to the fuel supplied, is given by

$$\eta = \frac{2(\phi - 1)}{\alpha - \phi\left(1 + \dfrac{\theta - 1}{\eta_c}\right)}\left[\sqrt{\frac{\alpha\Sigma}{\phi - 1}} - 1\right] \ldots\ldots [13]$$

With the assumed machine efficiencies, $\eta_c = \eta_t = 0.85$, and a pressure of 3, the efficiency at 500 mph and 20,000 ft altitude is approximately 15 per cent. An engine-driven propeller with a propulsion efficiency of approximately 60 per cent would give the same over-all efficiency. The efficiency, as a function of true air speed for different altitudes, is presented in Fig. 21.

Fig. 22 illustrates the effect of the efficiency of the compressor upon the static thrust developed and the corresponding fuel consumption. It is seen that with a turbine efficiency of 0.85 and a pressure ratio of 3, no thrust is developed if the compressor efficiency is less than 49 per cent. Increasing the efficiency of the compressor greatly reduces the fuel consumption and increases the thrust output. As a matter of fact, the efficiency of the system is improved by making the ratio of the compression work to the available energy of combustion as small as possible.[16]

[15] "Combustion Gas Turbine," by F. K. Fischer and C. A. Meyer, *Electrical Engineering*, vol. 63, 1944, pp. 163–169.

[16] "Reaction of Fluids and Jets," by R. Eksergian, *Journal of The Franklin Institute*, vol. 237, 1944, pp. 385–410.

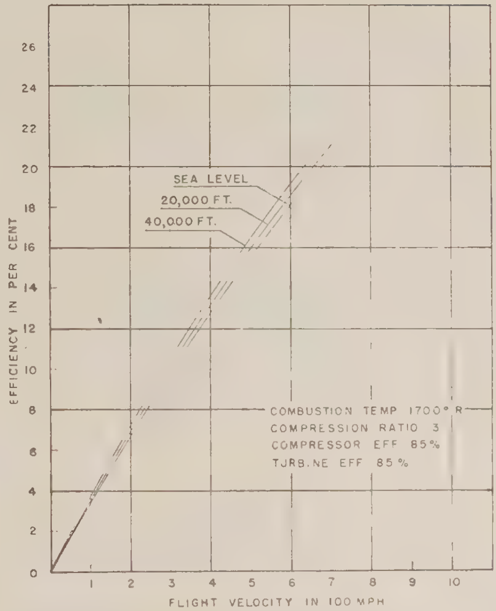

Fig. 21 Over-All Efficiency of Thermal-Jet Engine Versus True Air Speed

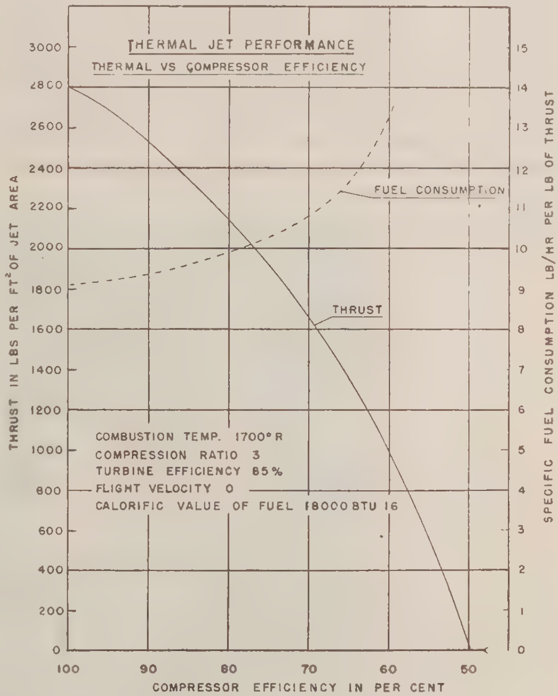

Fig. 22 Effect of Compressor Efficiency on Static Thrust and Fuel Consumption of Thermal-Jet Engine

The thermal-jet engine has made it possible to attain flight speeds higher than those obtainable with propeller propulsion. Because of its high fuel consumption, it has been widely held that its application will be restricted to short-endurance military

aircraft. Analyses of its commercial possibilities,[17] based on certain assumptions which have to be made at this stage in its development, indicate, however, that the jet-propelled transport should find a place in the postwar commercial field. These analyses indicate that for flights of the order of 500 miles, the jet-propelled transport provides economical transportation at high cruising speeds.[14] Until more of the operating data and actual performance characteristics of aircraft equipped with this form of power plant become available, the limits of the application of the thermal-jet engine to civilian aircraft cannot be stated with certainty.

Too much attention has been focused upon the apparent limitations of the thermal-jet engine, particularly upon its high fuel consumption; but these limitations appear to be related, to some extent at least, to the present state of the developments in the field of aerodynamics of high-speed flight. Thermal-jet propulsion imposes fewer restrictions upon the airplane designer. The commercial future of this method of propulsion depends to a large extent on the ability of airplane designers to produce airplanes with lower drag.

Perhaps the most attractive feature of the thermal-jet engine, apart from its ability to provide propulsion at high speeds, is its simplicity and low weight. Furthermore, this power plant is relatively free from vibration, permits using cheaper fuels than high-octane gasoline, and should require less frequent major overhauls.

Many problems are presented by the high operating temperatures required to obtain even fair fuel economy. Since weight and space are major considerations, the rate of heat liberation per cubic foot of combustion chamber has to be extremely high. For it to have reasonable life, its design must be arranged to give clean and reliable combustion, and the walls must be kept at safe operating temperatures. There are still metallurgical problems to be solved if higher operating temperatures are to be realized. In addition, there are problems in compensating for differences in thermal expansion. The fact that thermal-jet propulsion has become a practical reality is an indication, however, that great progress has been made in solving these problems.

[17] "Postwar Transport Aircraft," by E. P. Warner, *Aeronautical Engineering Review*, 1943, p. 7.

Rocket-Jet Application to Aircraft; Jato Units

The rocket has been developed to the stage where it should find a place as the power plant for short-duration high-speed aircraft (German ME 163); for long-range missiles (German V2 bomb); for superperformance at altitude, for torpedo drive, and for the assisted take-off of aircraft. Furthermore, the high-temperature high-pressure gas-producing capability of the rocket system should find application for short-duration turbine drives, or for any service where a temporary generation of such a gas can be employed for useful purposes.

In this country the principal application of rockets to aircraft apart from their use as weapons, is in the field of assisted take-off. Most of the applications have been to flying boats where the added thrust of the Jato units has made it possible to take off under conditions such that without them successful take-off would be problematical. This has increased the usefulness of such aircraft for rescue work in forward combat areas. Jato units have also been applied to carrier and land-based aircraft. As is to be expected, reductions in take-off time and/or distance have resulted from their use.[18]

Fig. 23 illustrates graphically the improvement in the sea level take-off characteristics of a DC-3 transport when equipped with a 1000-lb Jato unit having a duration of 14 sec. Fig. 24 is a similar comparison made at 6000 ft altitude. It is seen that the reduction in the distance to clear a 50-ft obstacle is from 4600 ft to 3250 ft at sea level and from 5800 ft to 3850 ft at 6000 ft altitude.

Two types of units have been applied; those employing liquid propellants, and those employing a solid propellant. To the author's knowledge, the solid-propellant units have found the greatest favor, owing to the ease of their installation, and the fewer logistic problems introduced by their use.

A liquid-propellant rocket system is ordinarily controlled from a single electric switch operated by the pilot. The switch actuates an electric control valve, thereby causing an inert pressurizing gas to communicate with the propellant tanks. The pressurizing gas passes through a regulating valve which main-

[18] "Take-Off Analysis for Flying Boats and Seaplanes," by E. G. Stout, *Aviation*, Oct., 1945, pp. 137–141; Nov., 1945, pp. 140–146, and Dec., 1945, pp. 163–166.

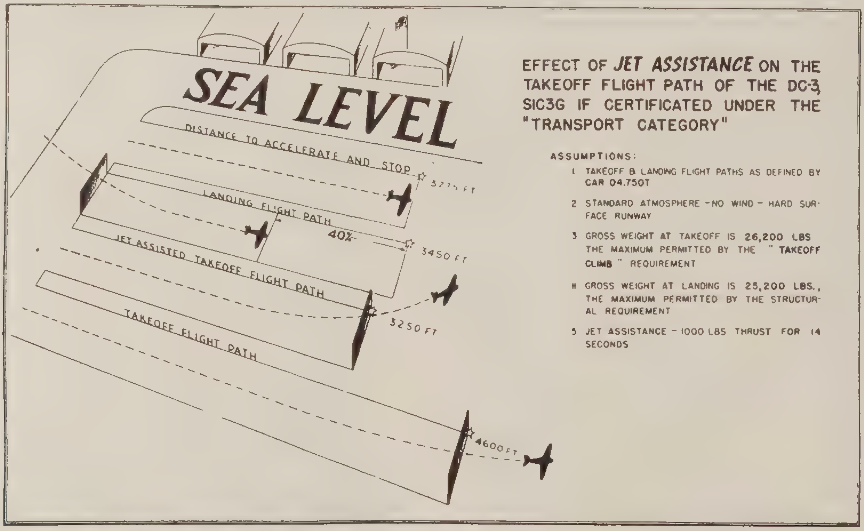

Fig. 23 Effect of Rocket-Jet Assistance on Take-Off Characteristics of DC-3 at Sea Level

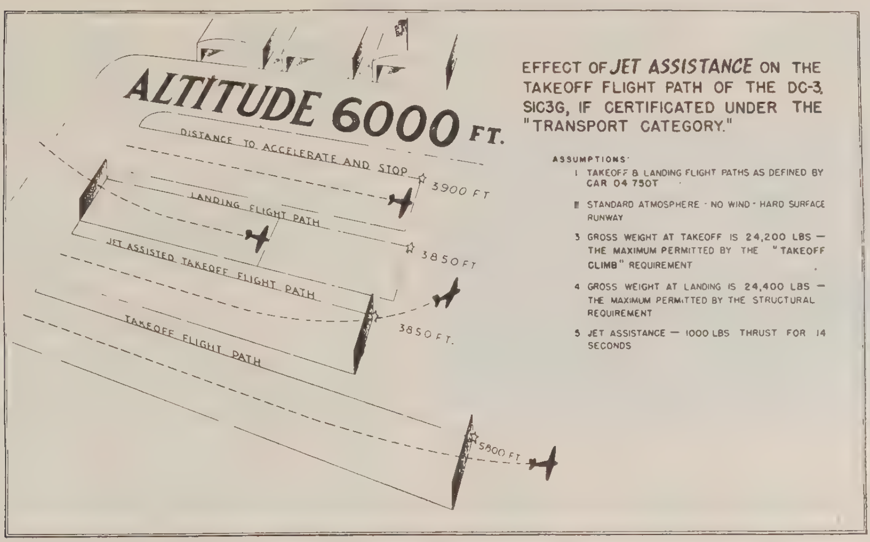

Fig. 24 Effect of Auxiliary Rocket-Jet Assistance on Take-Off Characteristics of DC-3 at 6000 Ft Altitude

tains the gas pressure on each propellant tank at a constant value. This gas is also utilized for actuating the propellant flow control valves located upstream relative to the injector which introduces the propellants into the motor. No ignition system is employed, since the propellants used with this unit react spontaneously upon contact with each other.

Liquid-propellant Jato units have been constructed so that they can be mounted either as fixed installations in the aircraft, or so that they can be dropped by parachute after use. Reliable liquid-propellant rocket motors have been developed that have operated without difficulty for continuous periods of several minutes. These motors are light in weight and are cooled by circulating one of the propellants around the combustion chamber and nozzle. For example, a rocket motor of this type developing 1500 to 2000 lb thrust can be built to weigh less than 45 lb. For security reasons, the design details of this type of motor cannot be released, nor can information be presented regarding the developments concerned with propellants or means for injecting the propellants into the motor.

Fig. 25 shows a typical solid-propellant Jato unit which delivers 1000 lb thrust for 12 sec. It is put into operation by means of the igniter which is fired electrically. Once the unit has been

Fig. 25 Typical Solid-Propellant Jato Unit for 1000 Lb Thrust, 12-Sec Duration

started it cannot be stopped and restarted, as can the liquid-propellant units.

Fig. 26 shows a PB2Y3 flying boat taking off with these units installed to supply additional thrust.

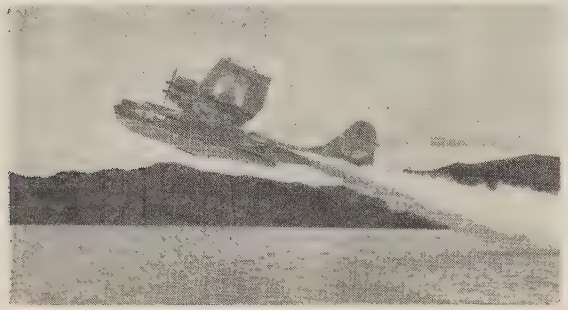

Fig. 26 Flying Boat Taking Off With Jato Assistance

Conclusions

Jet-propulsion devices have reached the development stage where they must be given serious consideration as a propulsion means. Where high-speed flight is of paramount importance, the thermal-jet engine has demonstrated its ability to accomplish that result. For shorter-duration high-speed flight, the rocket jet has potentialities.

In general, it appears that the future of jet propulsion in the postwar commercial field depends upon the success of the co-operative endeavors of airplane designers and jet-propulsion engineers in securing the maximum benefits offered by this mode of propulsion.

Acknowledgments

The author expresses his thanks to Mr. R. Sabersky, Aerojet Engineering Corporation, for his assistance in making some of the calculations for the performance characteristics of the thermal-jet engine.

(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until May 10, 1946)

Free Heat Convection Through Enclosed Plane Gas Layers

By MAX JAKOB,[1] CHICAGO, ILL.

Fifteen years ago, Mull and Reiher published elaborate experiments on heat convection through enclosed plane air layers. In the present paper it is shown that the representation and evaluation of the results of these experiments can be simplified, improved, and generalized when, as for a single surface in an extended medium, a laminar and turbulent range are distinguished. Defining slightly modified Nusselt and Grashof numbers with the layer thickness L as characteristic length, the experimental results for horizontal and vertical layers can be expressed by $(N_{Nu})_L = C_b(N_{Gr})_L^n$, or $C_v(N_{Gr})_L^n(L/H)^{1/9}$, respectively, where $n = 1/4$ and $1/3$ for the laminar and turbulent ranges, respectively, and H is the height of vertical layer. Comparison with tests of other authors indicates that, due to the experimental elimination of any influences from the edge of the enclosed layer, Mull and Reiher obtained relatively low values of the constants C_b and C_v, whereas without particular precaution these influences may considerably increase the heat transfer.

Nomenclature

The following nomenclature is used in the paper:

$A = H \cdot W$ = area of gas layer
C = constant
D = diameter
g = gravitational acceleration
H = height of vertical gas layer
h = coefficient of heat transfer by convection
k = thermal conductivity of gas
k_c = equivalent thermal conductivity, including conduction and convection
k_r = equivalent thermal conductivity, superseding radiation
$k_e = k_c + k_r$ = equivalent thermal conductivity
L = thickness of gas layer
N_{Gr} = Grashof number
N_{Nu} = Nusselt number
n = constant
q = rate of heat flow
$q'' = q/A$
t = temperature
W = width of gas layer
β = coefficient of thermal expansion
ν = kinematic viscosity
Φ, Ψ = functions

Subscripts:

c = convection
D = diameter as characteristic length
d = heat flow downward
H = height as characteristic length
h = horizontal gas layer
L = thickness of gas layer as characteristic length
m = mean
u = heat flow upward
v = vertical gas layer
1 = warmer surface
2 = colder surface

1 Introduction

In his excellent survey on heat transmission, presented in 1932, King (1)[2] dealt only briefly with the subject of heat transfer in air layers as "too involved to allow a complete discussion here." In particular, he had in mind the experimental results for plane air layers, published by Mull and Reiher in a not widely known supplement to a German magazine (2). It will be seen that these results are less involved than they appear in the original paper. The present paper will bring them to a more general and simpler representation and show their close relation to equations for free convection on horizontal and vertical surfaces.

2 Mull and Reiher's Procedure and Results

As a measure for the heat exchange in a gas layer, Mull and Reiher used an equivalent thermal conductivity k_e, which combines the effect of convection and radiation and is defined by

$$k_e = k_c + k_r \qquad [1]$$

where k_c and k_r are themselves equivalent thermal conductivities, the first one including the effect of conduction and convection and the second one superseding the effect of radiation.

Whereas k_r can be simply calculated, using the laws of radiation, it requires special experiments to determine k_c. Mull and Reiher employed two parallel plates, each 40 in. long and 24 in. wide, which were separated by air layers. These could be divided in different ways so that it was possible to study enclosed air layers having areas of 30 to 960 sq in. Air layers of seven different thicknesses from $1/2$ to $7^3/_4$ in. were employed. One of the two plates was an electrical heating plate composed of five separate sections and surrounded by an electrical ring heater. A secondary heating plate at the rear side, insulated from the front plate and ring, was held at the same temperature as these. Thus the main heat flow was confined to the front side from where it first crossed the air layer under consideration and then a cork plate which served as an auxiliary heat flowmeter. Strips of poorly conducting balsa wood, only 80/1000 in. thick, separated the air layers from each other and from the surrounding air which was in contact with the ring heater. Zinc was the material of the heating and cooling surfaces (zinc-plated steel sheets). The whole system of plates was surrounded by granulated cork and fixed in a wooden frame. This was turnable in a horizontal bearing so that the air layer could be brought into horizontal, vertical, or oblique position.

[1] Consultant in Heat Research, Armour Research Foundation; Research Professor of Mechanical Engineering, Illinois Institute of Technology; Nonresident Research Professor of Heat Transfer, Purdue University. Mem. A.S.M.E.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

The rate of heat flow by convection between a warmer surface (subscript 1) and a cooler surface (subscript 2) can be expressed by

$$q_c = k_c \frac{A}{L}(t_1 - t_2) \text{ or } q_c'' = \frac{k_c}{L}(t_1 - t_2) \quad [2]$$

where

L = the thickness of gas layer
A = $H \cdot W$, area of this layer
H = height of rectangular area A (when brought into vertical position)
W = width of this area
t_1 and t_2 are surface temperatures
$q_c'' = q_c/A$

Further, from the theory of similarity for air and probably also for any other diatomic gases (except under extreme conditions of state)

$$(N_{Nu})_L = \Phi[(N_{Gr})_L, H/L, W/L] \quad [3]$$

where slightly modified Nusselt and Grashof numbers are defined by

$$(N_{Nu})_L = \frac{q_c''}{(t_1 - t_2)} \cdot \frac{L}{k} = \frac{k_c}{k} \quad [4]$$

and

$$(N_{Gr})_L = \frac{\beta g}{\nu^2} L^3(t_1 - t_2) \quad [5]$$

with

g = gravitational acceleration
β = coefficient of thermal expansion of gas
k = its thermal conductivity
ν = its kinematic viscosity

the last two properties being taken at the mean temperature

$$t_m = {}^1/_2(t_1 + t_2)$$

From Equations [3] and [4]

$$k_c/k = \Phi[(N_{Gr})_L, H/L, W/L] \quad [6]$$

First considering horizontal layers (subscript h), the experiments, with heat flow upward, showed that $k_{c,h}/k$ is virtually independent of H and W. Then, from reasons of similitude, it will also be independent of H/L and W/L. Hence Equation [6] simplifies to

$$k_{c,h}/k = \Phi(N_{Gr})_L \quad [7]$$

Mull and Reiher represented their results by plotting $k_{c,h}/k$ versus log $(N_{Gr})_L$; obviously

$$k_{c,h}/k \longrightarrow 1 \text{ when } (N_{Gr})_L \longrightarrow 0$$

In the experimental range of $(N_{Gr})_L$ = 2112 to 8,890,000, a smooth curve of increasing steepness was obtained. Representation in bilogarithmic co-ordinates, however, will reveal that the curve has at least one bend in close relationship to other cases of free convection.

For vertical air layers (subscript v), the conditions are less simple than for horizontal layers because $k_{c,v}/k$ may not be independent of H and W. Mull and Reiher, in one of their tests, kept H/L and $(N_{Gr})_L$ constant, but reduced W/L from 25.7 to 12.8, that is, by 50 per cent. The term $(k_{c,v}/k)$ (H/L), however, changed only by 1.3 per cent. Hence, omitting W/L as independent variable and modifying Equation [6] slightly

$$\frac{k_{c,v}}{k} \cdot \frac{H}{L} = \Psi\left[(N_{Gr})_L \frac{H}{L}\right] \quad [8]$$

Similarly as for horizontal gas layers

$$k_{c,v}/k \longrightarrow 1 \quad \text{when } (N_{Gr})_L \longrightarrow 0$$

The problem also simplifies for very large Grashof numbers due to the vanishing of the frictional resistance between two parallel vertical surfaces with increasing distance; obviously

$$t_1 - t_m = t_m - t_2$$

and

$$q_{c,v}'' = h(t_1 - t_m) = h(t_m - t_2)$$

or

$$q_{c,v}'' = {}^1/_2 h(t_1 - t_2) \quad [9]$$

where h is the coefficient of convection for a single vertical surface. From Equations [2] and [9]

$$k_{c,v} = \frac{hL}{2} \quad \text{or} \quad \frac{k_{c,v}}{k} \cdot \frac{H}{L} = \frac{h}{k} \cdot \frac{H}{2} = \frac{1}{2} \cdot \frac{h}{k} \cdot \frac{H}{L} \cdot L \quad [10]$$

Hence, $k_{c,v}$ becomes proportional to L for very thick air layers and the resistance against thermal convection, $L/(k_{c,v}A)$ becomes constant. Since the resistance against heat transfer by radiation is also constant for given temperatures, it follows that the insulating power of an air layer cannot be increased infinitely by increasing the layer thickness as it would be with a solid layer, but is limited.

For layers of finite thickness, Mull and Reiher plotted $(k_{c,v}/k)$ (H/L), as measured, versus log $(N_{Gr})_L$ and built up 27 curves with H/L as parameter, making ample use of interpolation and extrapolation since a total of only 21 points from their own, and 3 from other experiments were available. Again, bilogarithmic representation will give a very much simpler and more reliable picture and will lead to simple equations for the laminar and turbulent ranges, analogous to those for single plane surfaces.

3 Dimensionless Equations for Free Convection on Vertical and Horizontal Surfaces

As already indicated, the heat convection on a plane plate is an extreme case of convection through a plane fluid layer.

The Nusselt and Grashof numbers for the free convection on a vertical surface of height H and temperature t_1 to a fluid of temperature t_m is usually defined by

$$(N_{Nu})_H = \frac{hH}{k} \quad [11]$$

and

$$(N_{Gr})_H = \frac{\beta g}{\nu^2} H^3(t_1 - t_m) \quad [12]$$

where arithmetic means, taken over the temperature difference $(t_1 - t_m)$, may be used for the properties of the fluid.

Employing the principle of similarity, Nusselt (3) showed that $(N_{Nu})_H$ is proportional to $H^{3/4}$, a result which had already been obtained analytically by Lorenz (4), 34 years before. This, however, is only a fair approximation to the actual much more complicated relation between h and H, as has been demonstrated later on by different workers, particularly by Schmidt and Beckmann (5) in a theoretical and experimental investigation. Moreover, Griffiths and Davis (6) had shown by experiments that above a certain height (about 2 ft for air) the coefficient h becomes independent of H and proportional to $(t_1 - t_m)$. They correctly

concluded that turbulence occurs above that height. It is worth noting that Nusselt (3) had already predicted an analogous phenomenon for horizontal cylinders placed in cooler air when $(N_{Gr})_D > 10^7$, the diameter D being used as characteristic length.

The occurrence of an exponent $n = 1/3$ in the equation

$$N_{Nu} = C(N_{Gr} \cdot N_{Pr})^n \qquad [13]$$

was later verified by a correlation of King (1) and by Jakob and Linke (7) in representing their own and Jakob and Fritz's experimental results (8) for not too vehemently boiling water. According to these investigations, the following equations can be used for vertical surfaces in an unrestricted fluid

$$(N_{Nu})_H = 0.555\,[(N_{Gr})_H N_{Pr}]^{1/4} \qquad [14]$$

for the laminar range and

$$(N_{Nu})_H = 0.129\,[(N_{Gr})_H N_{Pr}]^{1/3} \qquad [15]$$

for the turbulent range.

Jakob and Linke further showed that Equation [13] with $n = 1/3$ may also be used for the convection on the upper side of a horizontal plate (subscript u) on which water is boiling, and they concluded that in the range of their experiments, $q_{c,u}'' = 7$ to 5200 B hr^{-1} ft^{-2}, the coefficient of convection on a horizontal plate is independent of the size of the plate except for a possible effect of the edges. In fact, this had to be expected because it would not be understandable why the heat transfer on one place of a large horizontal plate should be different from that at any other place.

In other words, if Equation [13] is valid, then $n = 1/3$ must necessarily occur in all cases where no reason exists why h should depend upon a characteristic length. In a subsequent paper, Jakob and Linke (9) found by experiments with boiling water and carbon tetrachloride that the constant C of Equation [13] sensibly exceeds the values given in Equations [14] and [15]. For vertical surface, in the range where $n = 1/4$, $C_v = 0.61$ was found, compared to $C_v = 0.555$ in Equation [14]. This difference may be due to the action of the steam bubbles which is rather mild in the considered range, but increases rapidly at higher load (9, 10). With horizontal surface facing upward, $n = 1/3$ and $C_u = 0.16$ were observed to be compared with $C_v = 0.129$ for vertical surfaces according to Equation [15]. Here, in addition to the effect just mentioned, the difference of surface orientation seems to play a role. In fact, such an effect had previously been observed by Griffiths and Davis (6). From their tests on horizontal heating plates in air facing upward, the following equation can be derived

$$q_{c,u}'' = 0.275(t_1 - t_m)^{4/3} \qquad [16]$$

with

$q_{c,u}''$ in B hr^{-1} ft^{-2} and $(t_1 - t_m)$ in Fahrenheit units.

Combining this with their original values for vertical plates, 2 and $8^2/_3$ ft high

$$q_{c,u}''/q_{c,v}'' \approx 1.28 \qquad [17]$$

to be compared with $C_u/C_v = 0.16/0.129 = 1.24$.

For horizontal plates, facing downward (subscript d) where theoretically only conduction would be expected, Griffiths and Davis' measurements can be represented by

$$q_{c,d}'' = 0.50\,q_{c,u}'' \approx 0.64\,q_{c,v}'' \qquad [18]$$

It may be noted that

$$\frac{q_{c,u}'' + q_{c,d}''}{2} \approx \frac{1.28 + 0.64}{2}\,q_{c,v}'' = 0.96\,q_{c,v}'' \approx q_{c,v}'' \qquad [19]$$

Wilkes and Peterson (11), however, obtained about 1.6 instead of 1.28 (Equation [17]) and about 0.35 instead of 0.50 (Equation [18]). Their data are based on experiments with only 10 F temperature difference and therefore probably less exact.

4 New Equations for Free Convection Through Enclosed Plane Gas Layers

In the range of $(N_{Gr})_L$ = 10,000 to 10,000,000, Mull and Reiher's experimental data for horizontal air layers can be represented by two straight lines in bilogarithmic co-ordinates, Fig. 1. Intersection of these lines occurs at log $(N_{Gr})_L \approx 5.57$, corresponding to $(N_{Gr})_L \approx 370{,}000$. However, a state of transition may exist between $(N_{Gr})_L \approx 200{,}000$ and 800,000. In the

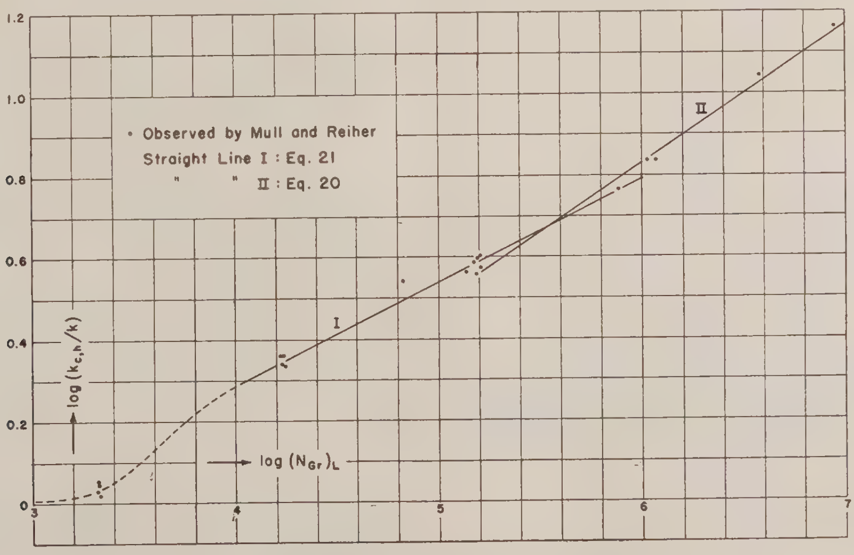

Fig. 1 Correlation for Horizontal Air Layers

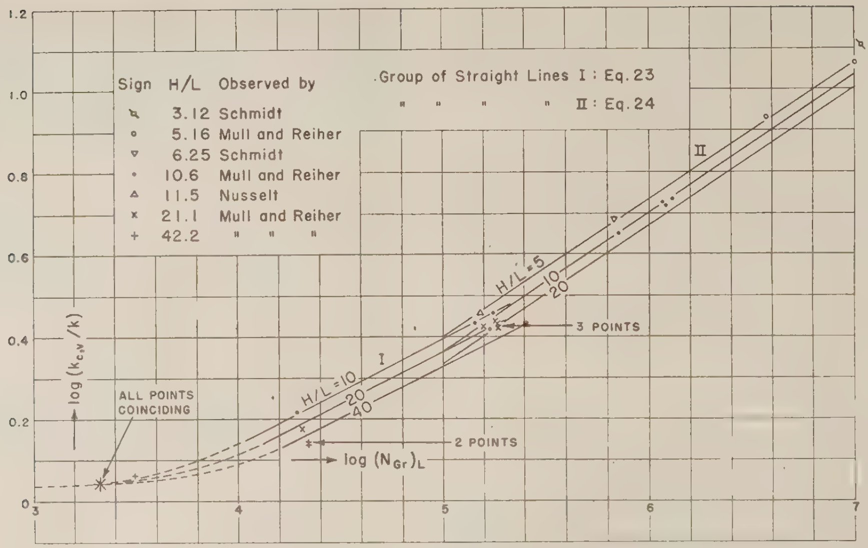

Fig. 2 Correlation for Vertical Air Layers

upper range,[3] similarly as for a heated surface, facing upward, or for a vertical surface

$$k_{c,h}/k = 0.068(N_{Gr})_L^{1/3} \quad \text{[20]}$$

In the lower range, in analogy to the behavior of vertical surfaces in a medium range of Grashof numbers

$$k_{c,h}/k = 0.195(N_{Gr})_L^{1/4} \quad \text{[21]}$$

For $(N_{Gr})_L \longrightarrow 0$, finally

$$k_{c,h}/k \longrightarrow 1 \quad \text{or} \quad \log(k_{c,h}/k) \longrightarrow 0 \quad \text{[22]}$$

as indicated by a dotted line in Fig. 1.

The meaning of Equation [20], obviously, is that above a certain thickness L the coefficient of heat transfer does not change any more, but should be the same as for a single horizontal plate, facing upward.

This may be checked by calculating an example: According to Equation [20]

$$q_{c,h}'' = 0.068k\left(\frac{\beta g}{\nu^2}\right)^{1/3}(t_1 - t_2)^{4/3}$$

For $t_1 = 75°$ C, $t_2 = 52°$ C, and 730 mm Hg atmospheric pressure, Mull and Reiher took $\beta g/\nu^2 = 0.110\,(10^9)$ m^{-3} C^{-1} and $k = 23.0\,(10^{-3})$ kcal hr^{-1}m^{-1}C^{-1}. Using these values and converting to British technical units

$$q_{c,h}'' = 0.068(23.0)10^{-3}(0.673)(0.110)^{1/3}(10^3)0.3048(1.8)(t_1 - t_2)^{4/3} = 0.277(t_1 - t_2)^{4/3}$$

in excellent agreement with Equation [16].

For vertical air layers, employing again bilogarithmic representation, all experimental points of Mull and Reiher and three points of other authors which they also used are plotted in Fig. 2.

At Grashof numbers as small as 2000, in analogy to Equation [7], no influence of H/L upon $k_{c,v}/k$ seems to exist in the range of the experiments (H/L = 10.6 to 42.2).

For still smaller Grashof numbers, the ratio $k_{c,v}/k$ approaches unity in analogy to Equation [22].

In the range of $(N_{Gr})_L$ = 20,000 to 200,000, the following equation represents the experimental data

$$k_{c,v}/k = 0.18(N_{Gr})_L^{1/4}(H/L)^{-1/9} \quad \text{[23]}$$

and in the range of $(N_{Gr})_L$ = 200,000 to 10,000,000

$$k_{c,v}/k = 0.065(N_{Gr})_L^{1/3}(H/L)^{-1/9} \quad \text{[24]}$$

Again the exponents, $1/4$ and $1/3$, of $(N_{Gr})_L$ are significant for laminar and turbulent flow. The exponent $-1/9$ of H/L, however, has been found empirically. The increase of $k_{c,v}/k$ with decreasing H/L is probably an effect of the lower and upper edge of the gas layer.

It is questionable how far beyond the range of the experiments Equations [23] and [24] are valid and, in particular, down to which smallest value of H/L Equation [24] can be used; certainly not down to $H/L = 0$, where it would yield an infinitely large heat transfer whereas the behavior of single vertical plates should be approached. However, the experimental point for $H/L = 3.12$ at N_{Gr} = 10,820,000 in Fig. 2 is in excellent agreement with the value calculated from Equation [24] ($\log[k_{c,v}/k]$ = 1.105, observed and 1.103, calculated) so that Equation [24] actually seems to be valid down to the ratio of $H/L \approx 3$.

Lines for H/L = 5; 10; 20; and 40 are drawn in Fig. 2, to facilitate interpolation. It is seen that they are in good agreement with the experimental points.

Mull and Reiher's paper includes one measurement for 45 deg inclination of the air layer. At H/L = 10.5 and $(N_{Gr})_L$ = 1,050,000 they obtained $k_c/k = 6.13$. Under the same conditions Equations [20] and [24] yield $k_{c,h}/k = 6.92$ and $k_{c,v}/k = 5.09$, the mean of which is 6.005. This result suggests that for angles from 0 to 90 deg linear interpolation between these equations may give reasonable results.

5 Discussion of Different Experimental Results

It must be mentioned that experiments, performed by Wilkes and Peterson (12), led to about 50 per cent higher values of q_c'' for vertical layers. An air layer with $L = 0.302$ and $H = W =$

[3] Particularly after correcting the highest point, at $(N_{Gr})_L$ = 8,890,000, for which Mull and Reiher erroneously have entered $k_{c,h}/k = 13.5$ (instead of 14.5) in their original graph.

2.667 ft was used, and it seems that the temperatures (t_1 = 60 deg F and t_2 = 40 deg F) were not varied either, so that the result would refer to $(N_{Gr})_L \approx 1{,}700{,}000$ only. The data given in the authors' paper do not allow one to check the accuracy of the results. The higher heat transfer obtained by them may be due to inclusion of the flow of heat through the studs confining the air layer and to the lack of a ring heater with air layer. Griffiths and Davis (6), employing a vertical layer with $L = ^1/_6$ ft and H/L = 24.8, arrived at almost 40 per cent higher heat transfer than Mull and Reiher, assumedly for the same reason. Experiments of Schmidt (13), on the other hand, performed with vertical layers, 0.263 or 0.526 ft thick, led to results in good agreement with those of Mull and Reiher (see Fig. 2). In these experiments a layer of 5-ft height was divided into three layers, each $1^2/_3$ ft high, by means of thin, horizontal strips of plywood, the adjacent layers giving some heat protection to each other. A test by Nusselt (14), on a vertical layer (L = 0.13 ft, H = 1.5 ft) led also to satisfactory agreement with our representation (see Fig. 2); this, however, may be accidental, considering the not too accurate experimental procedure.

With their air layer in horizontal position, Wilkes and Peterson, again at $(N_{Gr})_L \approx 1{,}700{,}000$, obtained a 40 per cent higher value than according to Mull and Reiher, whereas an experiment of Schmidt (13), at $(N_{Gr})_L \approx 30{,}000$, led to a 33 per cent smaller value than according to Mull and Reiher. This experiment was performed on a twin-plate device for measuring thermal conductivities, using a 1-in. air layer.

Considering the results of Wilkes and Peterson, and of Griffiths and Davis, the author looked for reasons why Mull and Reiher might have obtained too small values of convection. Referring to Equation [1], the value k_r was 27 to 49 per cent of k_e in their experiments. They determined k_r individually in each test by bringing the plates in horizontal position, heating the upper one, and assuming that no convection at all occurred in this case. According to Equation [20] or Fig. 1, this would mean that at $(N_{Gr})_L = 1{,}700{,}000$, after deduction of the radiated heat, the heat flow downward would amount to 12.3 per cent of that upward. This is not an unreasonable result, considering that Wilkes and Peterson found 23 per cent, without taking such precautions as the use of a heated air ring and thin balsa-wood strips to secure a strictly vertical heat flow downward. Anyway, k_r may have been found only a few per cent too large and k_c some per cent too small in Mull and Reiher's procedure. Incidentally, the emissivity of the zinc-plated steel sheet determined by them was rather too small than too large, according to the known values from literature, and there was no systematic influence of L on the measured value of that emissivity as would have been if convection had played a role.

Hence the only explanation for the mentioned differences is that, as just given, the Mull and Reiher values belong to the case of an enclosed partition, cut out somewhere in the middle of an extended gas layer between two parallel surfaces, each at uniform temperature, and with a practically nonconducting border strip, whereas in Griffiths' and Davis' results, as well as in those of Wilkes and Peterson, conduction effects on the edge of the layer are included. Therefore Equations [20], [21], [23], and [24] and Figs. 1 and 2 of the present paper yield values which may have to be increased by empirical factors if less ideal conditions exist at the edge of the air layer.

6 Free Convection Through Enclosed Plane Liquid Layers

According to the theory of similarity, Equations [20], [21], [23], and [24] should hold for any diatomic gases, except in such extreme ranges where the Prandtl number appreciably differs from $N_{Pr} = 0.72$, as assumed for air. Since for horizontal and vertical surfaces Equation [13] has been found to hold for gases and for liquids, even in the state of not too vehement boiling, with only moderately different values of C, the equations mentioned for confined layers may tentatively be used for other than diatomic gases and for liquids by multiplying the right sides by $(N_{Pr}/0.72)^n$, with $n = ^1/_4$ and $^1/_3$, respectively.

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Discussion

W. J. King.[4] This paper constitutes a useful contribution to the literature of heat transmission, by making the data of Mull and Reiher available to American engineers in the form of a convenient and significant correlation.

The use of an "equivalent conductivity," k_e, has the advantage of expressing heat transfer through air spaces in the same units as for solid insulation or building materials, which may be convenient for combining terms. On the other hand, there is a disadvantage in that the conductivity of an air space is by no means a constant property of the material, as in the case of cork or brick, so that the actual relationship between heat transfer and thickness of the air space is somewhat obscured.

For example, it is shown that for certain conditions the conductivity is proportional to the $^3/_4$ power of the thickness. Actually, since the heat-transfer coefficient h (heat flow per unit time per unit area per degree temperature difference), is equal to the conductivity divided by the thickness, i.e.

$$h = \frac{k}{L}$$

[4] Research Engineer, Battelle Memorial Institute, Columbus, Ohio. Mem. A.S.M.E.

it turns out that the heat flow varies inversely with the $^1/_4$ power, or 4th root of the thickness, in this region.

It is also important to understand that for the wider air spaces or higher Grashof's numbers, where the conductivity is directly proportional to the thickness, the actual heat-transfer rate, as Btu per hour per square foot per degree temperature difference, is independent of the thickness.

Probably the most questionable feature of this correlation is the effect of height upon heat transfer across vertical air layers, as expressed by the term $(H/L)^{-1/}$ in Equations [23] and [24] of the paper. As the author points out, for the higher values of Grashof's number, to which Equation [24] applies, the heat transfer should be similar to free convection from exposed plane surfaces, in which case the effect of height is sometimes more pronounced. In fact, for vertical surfaces about 1 ft high, the effect of the height H, may be represented by the $-^1/_4$ exponent, under ordinary conditions.

Tests on enclosed air spaces made by E. R. Queer[5] in this country showed that the heat transfer across an air layer $4^1/_2$ in. high was about 80 per cent greater than for a height of 35 in. For a height ratio of 1 to 8, as in this case, the $-^1/_9$ exponent would give a 1.27 ratio of heat-transfer rates, instead of the 1.8 ratio observed by Queer. It is very likely that the effect of height becomes more pronounced as the height is decreased.

The author recognizes the fact that this exponent is not constant, pointing out that Equation [24] would give infinitely large heat-transfer rates for $H/L = 0$. This brings out the danger of applying such equations as these to conditions too far removed from those under which the original data were obtained.

Author's Closure

In discussing the concept of "equivalent conductivity," Mr. W. J. King uses the word "equivalent" only once and then omits it for brevity. Then, of course, all statements concerning that quantity become odd, as, for instance, the one that the conductivity be proportional to the 3/4 power of the thickness. The sense of introducing an "equivalent" conductivity, however, is only to show how much more heat is actually transferred than would be by conduction alone. The ratio $k_c/k = 1.5$, for instance, means that the heat transfer by convection is 50 per cent larger than the heat transfer by mere conduction.

This sort of representation is neither novel nor unique. The rate of heat exchange by radiation q_r, for instance, is often expressed by means of an "equivalent" coefficient of radiation, h_r. For black surfaces h_r is defined by the equation

$$q_r = \sigma A(T_1^4 - T_2^4) = h_r A(T_1 - T_2)$$

wherein

T = absolute temperature
σ = constant of Stefan-Boltzmann's law

It is seen that h_r is not a constant as σ, but depends upon T_1 and T_2 in a rather odd way. However, when convection and radiation occur in the same field, it is practical to use h_r, because the total coefficient of heat transfer simply becomes

$$h = h_c + h_r$$

Also the ratio k_c/k should just be considered as a practical form or as a Nusselt number (see Equation [4] of the paper).

The term $(H/L)^{-1/9}$ of Equations [23] and [24] was chosen empirically as best representing the points in Fig. 2 of the paper. However, its influence is small and it would have been possible to neglect it entirely and to represent each of the point groups I and II of Fig. 2, by a single straight line, yielding values of $k_{c,v}$ with extreme deviations of ± 10 per cent from the measured values. Doing so, Equation [23] would contain only the exponent $^1/_4$, in analogy to Equation [14].

Mr. King has further directed the author's attention to a paper of E. R. Queer,[5] who observed a much larger increase in heat transmission when the height H was reduced than did Mull and Reiher. This seems to be partly due to the heat conduction through the wooden strips which confined the air layers and the thick plates extending over the top and bottom of the apparatus. According to Fig. 2 of Queer's paper, these probably were thick enough to cause the large heat transfer observed by him for small ratios H/L. A part of the discrepancy, however, is due to the difference in the absolute values of heat transfer, observed in the two investigations for great ratios of H/L. For example, at $L = 1/12$ ft, $t_m \approx 75$ deg F, $t_1 - t_2 = 50$ F, the author's Equation [23] yields $q_{c,v}'' = 21.5$ Bhr^{-1} ft^{-2} for $H = 4.5/12$ ft, and $q_{c,v}'' = 17.2$ for $H = 35/12$ ft, in agreement with Mull and Reiher's experiments, whereas Queer obtained $q_{c,v}'' = 27.5$ and 15.3, respectively.

When the author by illness was prevented from presenting his paper personally, Mr. King kindly presented it for him, adding some interesting comments which form the subject of his discussion. The author wishes to express his appreciation for Mr. King's co-operation and valuable contribution.

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A Graphical Determination of Unshielded-Thermocouple Thermal Correction

By W. M. ROHSENOW,[1] ANNAPOLIS, MD.

This paper presents a simple and rapid method for determining the thermal error of an unshielded thermocouple, with slide-rule precision.

Nomenclature

The following nomenclature is used in the paper:

Q = rate of heat transfer by convection
R = rate of heat transfer by radiation
C = rate of heat transfer by conduction
T_N = absolute temperature of N, deg R
t_N = temperature of N, deg F
h = film coefficient of heat transfer, Btu hr^{-1} ft^{-2} F^{-1}
k = thermal conductivity of thermocouple wire, Btu hr^{-1} ft^{-2} (F/ft)$^{-1}$
σ_b = Stefan-Boltzmann constant = 1.723×10^{-9} Btu hr^{-1} ft^{-2} R^{-4}
ϵ = emissivity
η = transmissivity
α = absorptivity
G = mass flow per unit cross section of duct area, lb ft^{-2} sec^{-1}
A = thermocouple bulb surface area, ft^2
D = thermocouple wire diameter, ft
d_0 = thermocouple bulb diameter, in.
b = exposed length of thermocouple, ft
T_m = $[(T_T^4 - T_W^4)/4(T_T - T_W)]^{1/3}$
T_a = average temperature between T_T and T_W = $1/2(T_T + T_W)$
θ = $(T_G - T_T)/(T_T - T_W)$

Subscripts:

G = gas
T = thermocouple
W = wall
S = thermocouple surface
GT = gas to thermocouple
TW = thermocouple to wall

Introduction

The problem of measuring the temperature of a gas stream by employing a thermocouple has been well treated by King (4),[2] Fishenden and Saunders (6), Bennett and Pirani (7), and Bosanquet (8). The accurate measurement of the temperature of a hot gas flowing in a duct is more difficult than the accurate measurement of the temperature of a solid or liquid. The temperature indicated by a thermocouple placed in the gas stream, Fig. 1, will always lie between the temperature of the gas itself and of the surrounding walls.

The thermocouple reading is in error because heat is exchanged between the thermocouple and the wall by radiation, and heat is transferred along the thermocouple wires by conduction. The analyses for the cases in which the gas is hotter than the wall and in which the wall is hotter than the gas are identical. When the gas is hotter than the wall, the thermocouple temperature is such that the rate of heat transfer by convection from the gas to the thermocouple plus the radiation from gas to thermocouple is equal to the radiation from the thermocouple to wall plus the conduction along the thermocouple wires. This may be stated in equation form

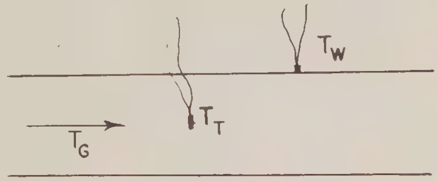

Fig. 1 Sketch of Duct With Temperatures Involved

$$Q_{GT} + R_{GT} = R_{TW} + C \quad [1]$$

The purpose of this paper is to present a simple and rapid solution of this equation in mathematical form and in chart form. The solution of Equation [1] involves solving a fourth-degree algebraic equation which is quite cumbersome to manipulate. The method of solution presented in this paper is simple and greatly reduces the time necessary to determine the thermocouple error. This method has slide-rule precision, which is more than sufficient when one realizes the inability to predict accurately emissivities and film coefficients of heat transfer.

Analysis

From the laws of convection and radiation, which may be found in McAdams (1), Equation [1] may be written in the form

$$Ah(T_G - T_T) + A\sigma\epsilon_g\left(\frac{1+\epsilon_s}{2}\right)(T_G^4 - T_T^4) = A\sigma\eta_G\epsilon_s(T_T^4 - T_W^4) \quad [2]$$

in which the conduction term is neglected. The conduction term is usually small; so it is neglected in the development of the equation from which the charts are constructed. Later a solution will be obtained for determining the magnitude of this conduction error.

The gaseous radiation term R_{GT}, and the gaseous emissivity ϵ_G of Equation [2] may be evaluated as indicated by McAdams,[3] and the thermocouple surface emissivity ϵ_s may be estimated as indicated by McAdams.[4]

If it is assumed that $\epsilon_G = \alpha_G$, then the transmissivity of the gas is

$$\eta_G = 1 - \epsilon_G$$

Dividing Equation [2] by $h(T_T - T_W)$ and simplifying

[1] Ensign, U.S.N.R., U. S. Naval Engineering Experiment Station. June A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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Note: Statements and opinions advanced in this paper are to be understood as individual expressions of the author and are not to be construed as official nor reflecting the views of the Navy Department or naval service at large.

[3] Reference (1), p. 68.

[4] Reference (1), p. 393.

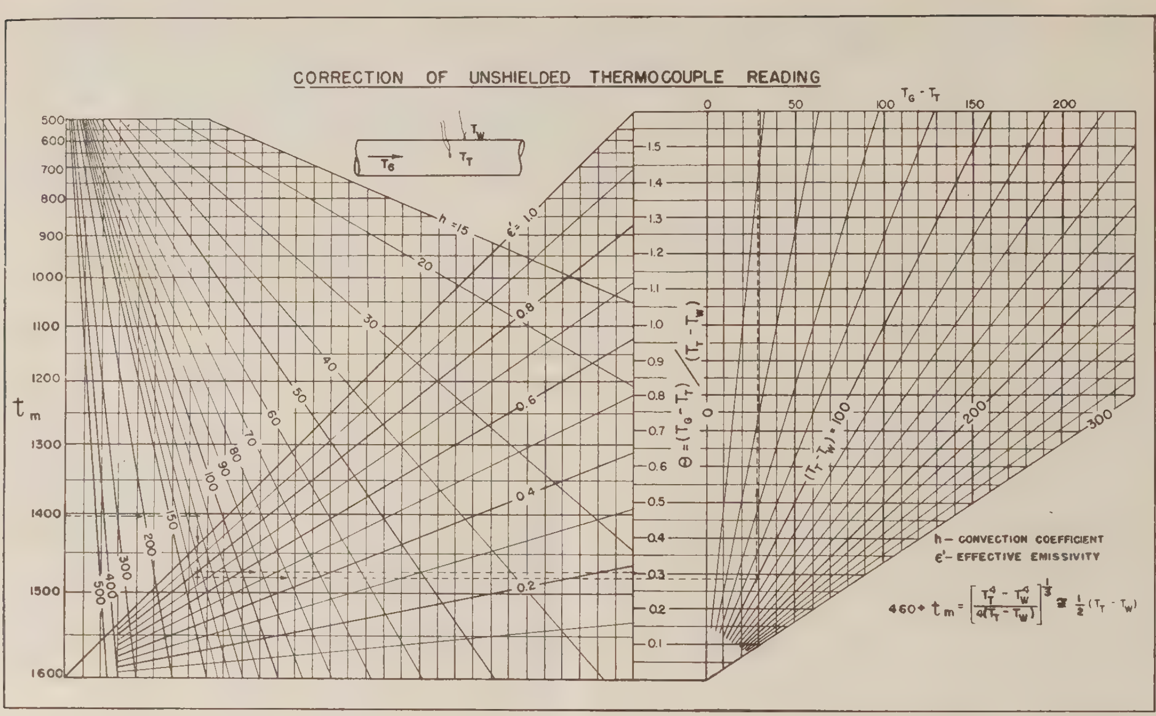

Fig. 2 Diagram for Unshielded-Thermocouple Correction

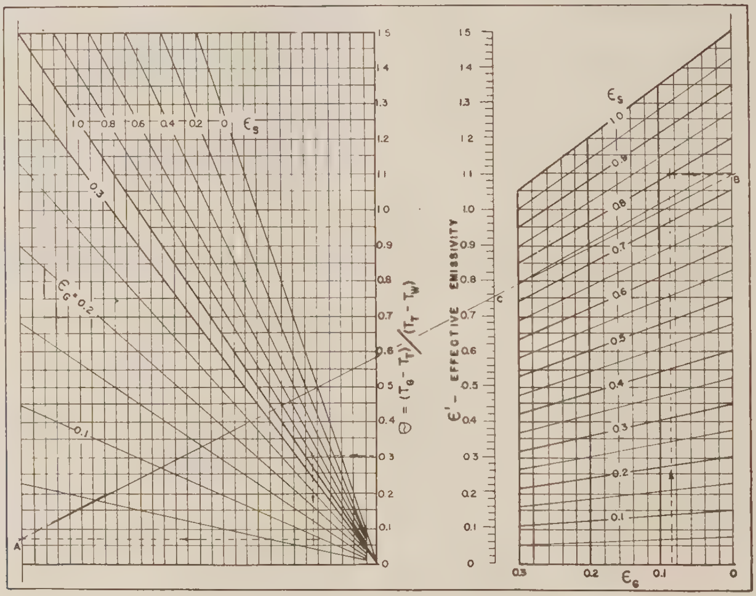

Fig. 3 Diagram for Determining Effective Emissivity

$$\frac{T_G - T_T}{T_T - T_W} = \frac{\sigma}{h} \cdot \frac{T_T^4 - T_W^4}{T_T - T_W} \left\{ \epsilon_s (1 - \epsilon_G) + {}^1/_2(1 + \epsilon_s) \cdot \epsilon_G \cdot \frac{T_G^4 - T_T^4}{T_T^4 - T_W^4} \right\} \ldots\ldots [3]$$

Now

$$\frac{T_G^4 - T_T^4}{T_T^4 - T_W^4} \equiv \frac{T_G - T_T}{T_T - T_W} \cdot \frac{T_G + T_T}{T_T + T_W} \cdot \frac{T_G^2 + T_T^2}{T_T^2 + T_W^2} \quad [4$$

Since T_G, T_T, and T_W are all large and of the same order of magnitude and their differences are small compared with the absolute magnitudes, the last two terms of the identity [4] are approximately equal to unity; hence

$$\frac{T_G^4 - T_T^4}{T_T^4 - T_W^4} \cong \frac{T_G - T_T}{T_T - T_W} \ldots\ldots [5]$$

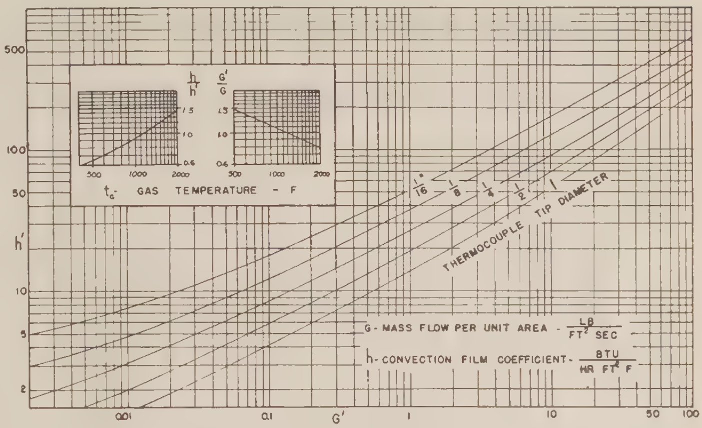

FIG. 5 DIAGRAM FOR DETERMINING FILM COEFFICIENT OF HEAT TRANSFER FOR SINGLE CYLINDERS NORMAL TO GAS STREAM

(Computed from Fig. 111, p. 221 of McAdams, Bibliographg reference 1.)

This approximation does not produce a serious error because it appears in the second term, in braces, of Equation [3]; and since ϵ_G is small compared with ϵ_s, the second term is much smaller than the first term.

Substituting Equation [5] in Equation [3]

$$\theta = 4 \frac{\sigma}{h} T_m^3 \epsilon' \ldots\ldots\ldots\ldots [6]$$

where the terms are defined in the nomenclature and where

$$\epsilon' = \left\{ \epsilon_s(1 - \epsilon_G) + {}^1/_2(1 + \epsilon_s)\epsilon_G \frac{T_G - T_T}{T_T - T_W} \right\} \ldots\ldots [7]$$

in which ϵ' is called the effective emissivity.

The solution of Equation [6] may be obtained from the diagram in Fig. 2, and ϵ' may be obtained from the diagram in Fig. 3.

Expand T_m as defined in the nomenclature

$$T_m^3 = \frac{T_T^4 - T_W^4}{4(T_T - T_W)} = \frac{T_T + T_W}{2} \cdot \frac{T_T^2 + T_W^2}{2} \ldots [8]$$

Since ${}^1/_2(T_T + T_W) = T_a$ and ${}^1/_2(T_T^2 + T_W^2) \cong T_a^2$, where T_a is the average temperature between T_T and T_W, Equation [8] indicates that

$$T_m \cong T_a \ldots\ldots\ldots [9]$$

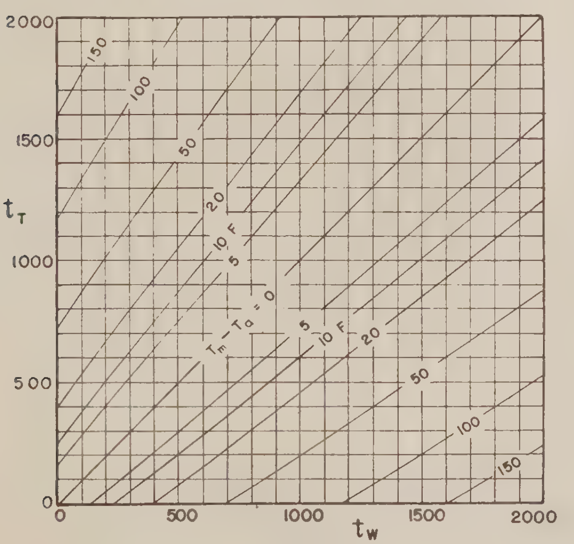

FIG. 4 DIAGRAM FOR DETERMINING $T_m - T_a$

The error in the assumption that $T_m = T_a$ is found in Fig. 4. Usually this error is less than 10 deg F and may be neglected.

If $\epsilon_G = 0$ then $\epsilon' = \epsilon_s$ and if T_m is assumed equal to T_a, Equation [6] becomes

$$\theta = 4 \frac{\sigma}{h} T_a^3 \epsilon_s \ldots\ldots\ldots\ldots [10]$$

For all practical purposes

$$\sigma = \sigma_b = 1.723 \times 10^{-9} \ldots\ldots [11]$$

USE OF THE DIAGRAMS

It is noted that a value of T is necessary to evaluate ϵ' in Fig. 3; hence the solution is of the trial-and-error type. First, solve for θ from Fig. 2, assuming $\epsilon' = \epsilon_s$. Then with this value of θ determine ϵ' from Fig. 3. This new value of ϵ' will be sufficiently close to the true value to obtain a solution. Now return to Fig. 2, with the new value of ϵ' and solve for θ and for $(T_G - T_W)$.

In order to obtain a solution of Equation [3] from Fig. 2, it is necessary to know h, the film coefficient of heat transfer between the flowing gas and the thermocouple. If the thermocouple approximates a cylinder normal to the gas stream, the curves in Fig. 5 may be used to estimate the value of h. For a given gas temperature, obtain from Fig. 5 the ratios G'/G and h/h'; then for any gas flow G, the value of G' is known. Now determine the value of h' for a given thermocouple tip diameter; then h is the product of h' and h/h'. The curves in Fig. 3 are computed from data presented by McAdams[5] for air flowing normal to a single cylinder. If this condition is not approximated, some other data must be used for estimating the value of h.

CONDUCTION ALONG THE LEADS

In the foregoing analysis the conduction heat-transfer term C of Equation [1] has been neglected. This term is usually small, but in order to determine its effect consider the conduction of heat to the cold wall along the two wires of the thermocouple in a gas stream as shown in Fig. 6. The differential equation is

$$\frac{d^2t}{d^2x} = \frac{4h}{kD}(t - t_G) \ldots\ldots\ldots\ldots [12]$$

[5] Reference (1), Fig. 111, p. 221.

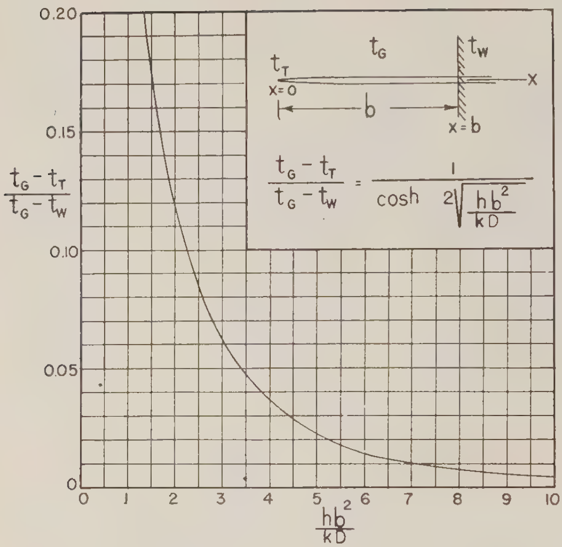

FIG. 6 DIAGRAM FOR DETERMINING CONDUCTION ERROR

whose solution[6] at $x = 0$ is

$$\frac{t_G - t_T}{t_G - t_W} = \frac{1}{\cosh 2\sqrt{\frac{hb^2}{kD}}} \quad [13]$$

This equation is shown graphically in Fig. 6.

ILLUSTRATIVE EXAMPLE

Determine the gas temperature of a gas flowing through a duct with the following conditions:

$G = 7.2$ lb sec^{-1} ft^{-2}
$t_T = 1450$ F
$t_W = 1350$ F
$D = 0.0675$ in. $= 0.005625$ ft
$d_0 = {}^1/_8$ in.
$b = 6$ in. $= 0.5$ ft
$k = 220$ (assuming copper)
$\epsilon_G = 0.085$
$\epsilon_s = 0.80$

Solution: From Fig. 5, $G'/G = 0.90$ and $h/h' = 1.20$ for a temperature of 1450 F; then $G' = 0.90 \times 7.2 = 6.48$ lb sec^{-1} ft^{-2}. For a ${}^1/_8$-in. tip diam, $h' = 100$ from Fig. 5, and $h = 1.2 \times 100$ or 120 Btu hr^{-1} ft^{-2} F^{-1}.

Now $t_a = {}^1/_2(t_T + t_W) = 1400$ F, and from Fig. 4, $(t_m - t_a) = 1^1/_2$ F; so $t_m = 1400 + 1^1/_2 = 1402$ F.

As a first approximation determine T from Fig. 2 by assuming $\epsilon' = \epsilon_s = 0.8$; enter the diagram in Fig. 2 at the left with $t_m = 1402$; then follow the dotted line in the direction of the arrows to $h = 120$, $\epsilon' = 0.8$, and to $\theta = 0.305$. Use this value of θ to determine ϵ' from Fig. 3. Enter Fig. 3 with $\theta = 0.305$; follow the arrows to $\epsilon_s = 0.80$, $\epsilon_G = 0.085$ and then to the left alignment line to locate point A. Locate point B by following the arrows from $\epsilon_G = 0.085$ to $\epsilon_s = 0.80$ to the right alignment line. Connect points A and B with a straight line to locate point C, reading $\epsilon' = 0.755$.

Return to Fig. 2 with the new value of ϵ' to obtain the new value of $\theta = 0.285$. Continue to $(t_T - t_W) = 1450 - 1350 = 100$ F and up to obtain the thermocouple correction $(t_G - t_T) = 28$ deg F; hence $t_G = 1450 + 28 = 1478$ F.

If gaseous radiation is neglected, $\epsilon_G = 0$ and $\epsilon' = \epsilon_s$, and if t_m is assumed to be equal to t_a, the solution from Fig. 2 yields $(t_G - t_T) = 30$ F, and $t_G = 1450 + 30 = 1480$ F.

An estimate of the effect of conduction along the thermocouple wires is obtained from Fig. 6. For this example

$$\frac{hb^2}{kD} = \frac{(120)(0.5)^2}{(220)(0.005625)} = 24.2$$

then from Fig. 6, the error term due to conduction alone is

$$\frac{t_G - t_T}{t_G - t_W} = \frac{t_G - 1450}{t_G - 1350} < 0.001$$

from which $t_G < 1451$ F. Hence the error due to conduction alone is less than 1 deg F in this case.

CONCLUSIONS

To determine the gas temperature solve Equation [6] mathematically or graphically by using the diagrams in Figs. 2, 3, and 4.

For a less precise solution solve Equation [10] mathematically or graphically by using only Fig. 2, with $\epsilon' = \epsilon_s$ and $t_m = t_a$.

Either of these methods will yield results which are better than our knowledge of surface and gaseous emissivities and heat-transfer film coefficients warrants.

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[6] See reference (6) or (2).

(Owing to travel emergency conditions existing when this paper was contributed, discussion will be accepted until May 10, 1946)

Coaster Gate and Handling Equipment for River Outlet Conduits in Shasta Dam

By J. E. WARNOCK[1] and H. J. POUND,[2] DENVER, COLO.

At Shasta Dam, a coaster gate is used to close the intake of any one of the eighteen 102-in. conduits in the spillway section of the dam. Each conduit is provided with a control valve. The coaster gate is used to close the intake of each conduit whenever required for inspection and servicing of the control valve and conduit. It may also be used for emergency closure in the event of failure of a control valve. Since the gate design is predicated on emergency conditions of maximum head, when it is subjected to large unbalanced pressures, hydraulic model studies were conducted to determine the best shape of gate to minimize the downpull force to which it might be subjected in an emergency closing. The procedure followed in conducting the studies and the final installations are treated in this paper.

Introduction

Shasta Dam, one of the major features of the Central Valley Project, is located on the Sacramento River, 9 miles above Redding, Calif. It is designed as a multipurpose dam with facilities for flood control, river regulation, and power generation.

Release of stored water for river regulation in excess of the capacity of the powerhouse turbines is accomplished by eighteen 102-in-diam conduits in the spillway section of the dam. Each of these conduits is provided with a control valve. A coaster gate is used to close the intake of any one of the 18 conduits whenever required for inspection and servicing of the control valves and conduits. The gate may also be used for emergency closure in the event of failure of a control valve.

Normally the gate is operated under balanced hydrostatic pressures with no flow in the conduits. However, design conditions were taken as those which exist during emergency closure under maximum head when the gate is subjected to large unbalanced pressures.

As the gate is lowered under emergency conditions, the increase in velocity under the gate acts to decrease the pressures on the downstream face while those on the upstream face remain substantially constant. The frame of the gate must resist the resultant force which pushes it against the face of the dam, and the rollers supporting the gate must have a low frictional resistance or the gate cannot be lowered into the closed position by its own weight. Another effect of the high-velocity flow is the reduction in pressure on the bottom of the gate which creates an additional force, referred to as downpull. Consideration of this force is important as it may be equal to or greater than the weight of the gate.

The hydrostatic pressures on the top of the gate can be calculated readily; but unfortunately, calculations of the downpull force can be only approximate without detailed hydraulic-laboratory studies. The pressures on the bottom of the gate are a function of the flow velocity under the gate, the shape of the gate bottom, and the gate opening, and they may vary from full hydrostatic pressure to the vapor tension of water. The value of the downpull force is therefore dependent upon the velocity distribution and flow pattern beneath the gate, since the pressure reduction at any point on the gate bottom is equal to the velocity head at that point.

In previous designs an approximation of the downpull force was considered satisfactory, but at Shasta Dam the estimated downpull on the coaster gate was so large that the total load on the handling equipment was about 60,000 lb in excess of the permissible load. The gate was to be handled by a 125-ton gantry crane, operating on the bridge across the spillway section of the dam. The capacity of this bridge was the limiting factor. Such a circumstance arose from the fact that the original plan was to handle the coaster gates from a barge and the bridge was designed for normal traffic load. When later in the design the handling equipment was transferred from the barge to the bridge, it was necessary to determine more accurately the downpull force and to reduce it, if possible, to avoid a drastic change in the design of the bridge across the spillway.

Hydraulic Model Studies

Hydraulic model studies were made to check the computed downpull and to develop a new shape for the gate bottom. Various shapes were tested and a satisfactory design was developed. A combination of this design and a properly proportioned recess in the face of the dam above the inlet reduced the downpull from the original value of 260,000 pounds to 70,000 lb.

The studies were made with a model of the conduit and coaster gate built to a scale of 1 to 17. Test data included pressure measurements on the gate and in the outlet for several heads and gate openings. The downpull was determined by integrating the pressure curves shown in Fig. 1. In the tests of the final design, the value obtained by pressure integration was verified by direct measurement with a spring scale.

With the gate of the original design in the full-open position, and a head representing the maximum of 323 ft, the downpull was approximately 75,000 lb. As the gate was lowered the downpull gradually increased to a maximum of 260,000 lb at an opening of about 8 ft 6 in. and then decreased gradually to zero for the closed position.

The typical variation of downpull with gate opening, which is shown graphically in Fig. 3, may be explained by a consideration of the variation in magnitude and distribution of velocity under the gate. When the gate was fully opened the discharge was a function of the size of the outlet conduit and its frictional resistance. As the gate was lowered it created a restriction in conduit cross section; and since the discharge was not reduced in the same proportion, there was an increase in velocity and a reduction in pressure under the gate and for a short distance downstream. The decrease in pressure caused an increase in downpull.

Since the conduit was vented downstream from the gate, air re-

[1] Chief, Hydraulic Laboratory, Branch of Design and Construction, Bureau of Reclamation.

[2] Chief, High-Head Gate Section, Branch of Design and Construction, Bureau of Reclamation.

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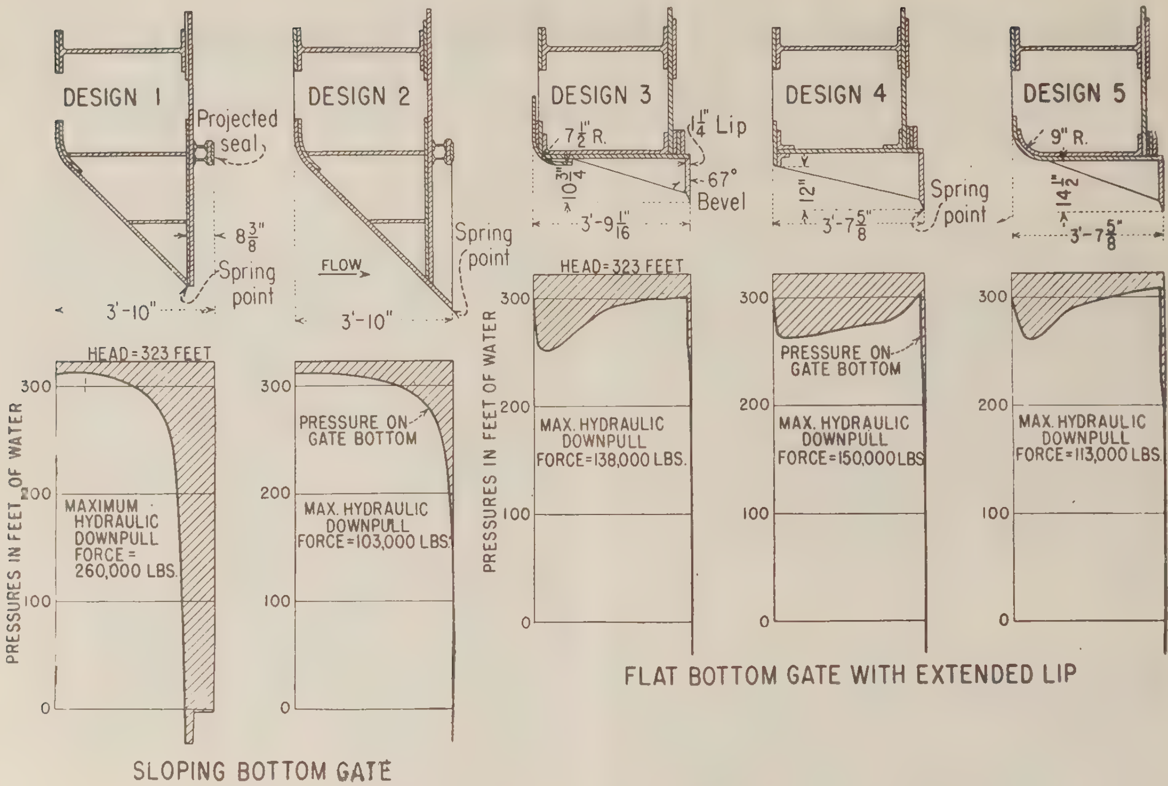


Fig. 1 Pressure Studies of Gate Bottom

lief was obtained when the restriction in area was sufficient to lower the pressure at that point to atmospheric pressure. When the pressure downstream from the gate was restrained from decreasing to any appreciable extent by the presence of air relief, it may be said that the gate became a control. The discharge under this condition became a function of gate opening and head and was not affected by the conduit. At the gate opening corresponding to maximum downpull, the gate became a control and the velocity along its bottom reached its maximum value. For lesser gate openings, the velocity at the downstream edge of the bottom was increased slightly by a minor decrease in pressure downstream from the gate. However, at these openings the proportions and shape of the gate bottom relative to the size of the opening changed in such a manner that the velocity along it actually decreased and pressures increased enough to lower the value of downpull.

At small openings on the original design the jet under the gate impinged on the top of the conduit and interfered with proper action of the vent.

Reduction of Downpull in Original Design

The hydraulic downpull force of 260,000 lb, as determined by the pressure tests, did not agree with the original analytical estimate of 160,000 lb. A review of the original calculations indicated that the estimate of 160,000 lb considered only the pressure reduction on the sloping portion of the gate bottom and assumed that a recess in the face of the dam above the outlet entrance would balance the pressures on the projected seals. The model test was made without a recess, and the unbalanced pressures on the top seal contributed at least 100,000 lb to the total downpull of 260,000 lb. A recess in the model of the original design would have tended to balance the pressures on the top seal, and it was possible that a closer agreement might have been obtained. However, the downpull would still have been excessive; therefore consideration was given to reduction of the force by revision of the shape of the gate bottom. The effectiveness of a recess in the face of the dam was determined only for the most satisfactory gate bottom and will be discussed subsequently.

Studies of the flow under the gate and the pressure on its bottom indicated that a large reduction of downpull could be obtained by a simple revision of the bottom. The flow under the gate and into the outlet was studied by observing the movements of paper particles in the head tank through a window. The particles approached the outlet from all directions, moving slowly until they were within a few inches of the outlet, where they appeared to be drawn instantaneously into it, indicating a rapid increase in velocity close to the outlet entrance. This rapid increase in velocity was also shown by the pressure gradient across the 45-deg sloping portion of the bottom of the gate, Fig. 1, design 1. At the upstream edge the pressures were practically hydrostatic. On approaching the outlet the pressure reduction was gradual at first, but it became rapid close to the downstream edge. At the point where the 45-deg slope ended, the pressure reached a minimum. This was the spring point of the gate, that is, the point where the water normally sprang free of the gate bottom to form a jet in the conduit.

The face of the gate projected a distance of $8^3/_8$ in. beyond the spring point as shown in Fig. 1. The pressure forces acting upward on this projection were much lower than those acting downward so that it contributed an excessively large part of the downpull when its relatively narrow width was considered. By comparison, the much larger area of the 45-deg sloping bottom played

only a minor role because the low pressure prevailed only in the vicinity of the spring point. The original bottom design was revised by extending the sloping portion until it underlay the projecting seal, as shown in Fig. 1, design 2. This change placed the spring point close to the face of the dam and eliminated the undesirable projecting area. The pressure distribution on the revised gate bottom is shown in Fig. 1, design 2. The shape of the pressure curve remained the same but the elimination of the projection and its unbalanced loading reduced the downpull from 260,000 to 103,000 lb.

In contrast to the original design, where the maximum downpull occurred when the conduit was not completely filled with water, the maximum value with the revised gate occurred while the conduit was filled with water, just before a slight additional closure would lower the pressures so that the conduit would take some air through the vent. In addition to reducing the downpull, this revised design changed the shape of the jet flowing into the outlet so that the jet at no time impinged on top of the conduit to restrict the air vent, as was the case in the original design when the gate was open between 2 and $3^1/_2$ ft.

Effect of Extended Lip Below Downstream Edge of Flat-Bottom Gate

Although the revision of the bottom accomplished the desired reduction in downpull, the gate was not satisfactory structurally. The sloping bottom would have been difficult to fabricate and heavy plates would have been required to withstand the loads on its downstream edge. Thus it was necessary to make further tests to develop some other type of gate which would have even less downpull or at least a more acceptable structural design for the bottom.

The original tests indicated that a gate having a minimum downpull would be one with a lip placed at the downstream edge of the bottom and extended vertically below the gate. This would place the spring point at a greater distance from the bottom and the effect of the rapid drop in pressure which occurs near the spring point would be exerted on the vertical plane of the extended lip and would not contribute to downpull. To verify these indications, tests were made with an extended lip. The length of the lip was varied in successive steps, from zero to a length nearly equal to the thickness of the gate.

The complete shape of the gate bottom which will be referred to hereafter as the basic shape is shown in Fig. 2. The bottom was faired into the upstream plate on a 9-in. radius, to minimize the local reduction of pressure on the bottom where the flow down the upstream face of the gate changes direction below the gate. The extended lip was $^3/_4$ in. thick, and its bottom was beveled at 45 deg to place the spring point at the downstream edge of the lip and reduce the effect of the thickness on the downpull. The extended lip was supported by gusset plates attached to the bottom of the gate. These plates were in the plane of flow so that their effect on the downpull would be small and could be ignored.

The pressure gradients, as determined in what will be referred to as general tests, were similar to those of design 5, Fig. 1, except that negative pressures occurred on the bottom of the lip. There was some reduction of pressure near the upstream plane, as was anticipated, since the flow down the upstream face of the gate had to change its direction. However, the pressure increased rapidly, becoming a maximum at the downstream corner where the lip joins the gate bottom. This effect indicated that not only does the extended lip keep the rapid reduction in pressure near the spring point on the vertical plane of the lip where it cannot cause downpull, but it also tends to form a stagnation point which increases in the downstream corner.

Although the bottom of the lip was beveled at 45 deg to place the spring point on its downstream edge, the spring point actually occurred at the upstream edge of the lip. The resulting negative pressures on the bottom of the lip would cause a downpull of approximately 15,000 lb when the lip was $^3/_4$ in. thick, as indicated in Fig. 2.

The variation of maximum downpull force with length of lip extension is shown graphically in Fig. 2. The longest extension considered was approximately equal to the thickness of the gate,

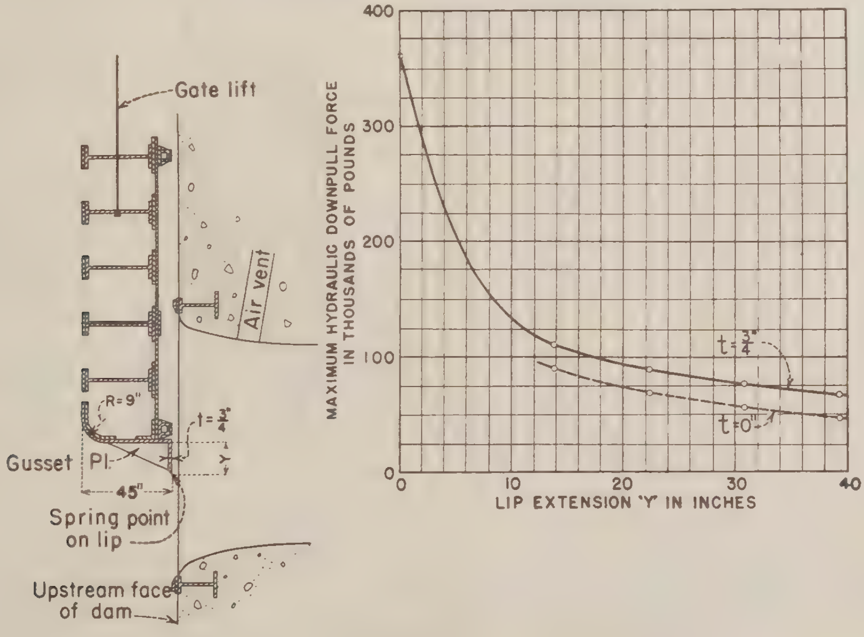


Fig. 2 Effect of Lip Extension on Hydraulic Downpull

as a greater extension would be impractical structurally. The graph shows that an extension of 40 in. would reduce the downpull to approximately 65,000 lb. As the extension was decreased, the downpull increased gradually until, at a lip extension of 14 in., the value was 110,000 lb. A further decrease in the extension caused a more rapid increase in downpull which finally became 360,000 lb when there was no extension.

Structural Designs of Flat-Bottom Gate With Extended Lip

A flat-bottom gate with an extended lip below its downstream edge, which would develop a downpull equal to that of the sloping-bottom gate of design 2, Fig. 1, or about 100,000 lb, would require a lip extension of 17 in. This was not practical, since the horizontal forces on it would be excessive. Nevertheless, a flat-bottom gate having an extended lip was a simple design compared with the sloping bottom of design 2.

Design 3 was more acceptable than the basic shape from a structural viewpoint, having a lip extension of $10^3/_4$ in. which, from the curve in Fig. 2, corresponds to a downpull of approximately 130,000 lb. The radius of the curved portion of the bottom was made $7^1/_2$ in. instead of 9 in., and the thickness of the lip was made $1^1/_4$ instead of $^3/_4$ in. It was anticipated that the smaller radius would increase the downpull a small amount. The bottom of the lip was beveled at a steeper angle, 67 deg, which placed the spring point at its downstream edge and reduced the downpull on the lip, Fig. 1. The tests showed that the downpull would be approximately 138,000 lb. Piezometers placed on the lip to determine pressures indicated that the portion of the downpull due to the $1^1/_4$-in. lip was nearly equal to that of the $^3/_4$-in. lip of the general test in which the spring point was at its upstream edge. Accordingly, the curve in Fig. 2 was used to predict the downpull of a gate having $1^1/_4$-in. lip with a 67-deg bevel at its bottom.

In design 4, Fig. 1, a simplification of the structural details was made by using flat plates and angles to eliminate the curved section of the gate bottom. The end of the plate at the upstream edge was rounded to avoid a sharp corner at that point. A $1^1/_4$-in. lip having a steep bevel, similar to design 3, was extended 12 in. below the upstream edge. The downpull was 150,000 lb, which represented a 25 per cent increase over that obtained in the general test with a 9-in. radius and a 12-in. lip extension. Neither design 3 nor 4 was satisfactory for the Shasta Dam outlet coaster gates because their downpulls, of 138,000 and 150,000 lb, respectively, exceeded the allowable limit.

A new analysis of the stresses on the bottom of a gate with the basic shape used in the general test, Fig. 2, revealed that it would be structurally sound if the lip extension did not exceed $14^1/_2$ in. Fig. 2 indicated that such a design would develop a downpull of 110,000 lb. Since this value was not excessive, design 5 was constructed and tested, Fig. 1. It differed from the gate of the general test in that a $1^1/_4$-in. lip with a 67-deg bevel at its bottom was used instead of $^3/_4$-in. lip with a 45-deg bevel. The measured downpull of 112,000 lb checked the predicted value of 110,000 lb.

The final design of the Shasta outlet coaster gate was developed from design 5. The width was increased from $43^5/_8$ to $44^{11}/_{16}$ in. for structural reasons, and a clearance of $^1/_2$ in. between the lip and the seal seats was introduced to facilitate operation of the gate. The maximum downpull as determined by pressure measurements was increased to 122,000 lb by these changes. An independent check of the downpull by direct measurement with a spring scale indicated that a slightly larger downpull of 131,000 lb occurred at a gate opening of 55 per cent. The latter value was accepted as the more accurate one as it was impractical to take the number of pressure measurements required for a comparable accuracy.

The over-all effect of the gusset plates was determined by a test with the plates removed. No appreciable difference in downpull could be detected. If plates were terminated at the point where the bevel of the lip begins as shown in Fig. 1, their effect on downpull was negligible.

Effect of Recess in Face of Dam

It had been appreciated that any one of the various designs of the gate could have been improved from the standpoint of downpull by introducing a recess in the face of the dam. This improvement was deliberately withheld until the best shape of gate bottom was determined.

To understand the action of the recess, it must be noted that the seals of the gate project beyond the skin plate. With no recess, the pressures on the top seal were unbalanced when the gate was partially closed. The full reservoir head acted on the outside, or topside, while the low

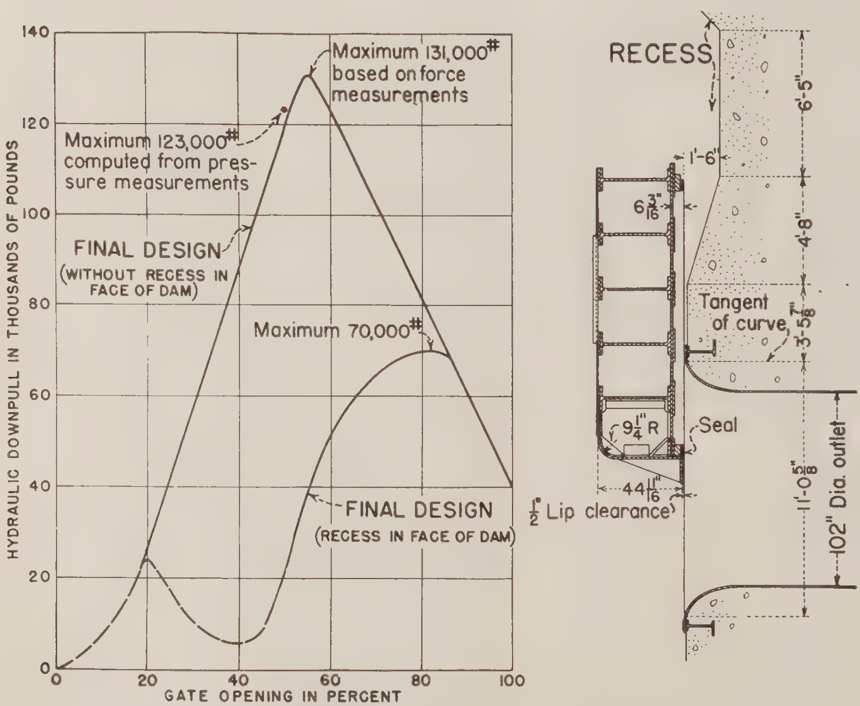


Fig. 3 Hydraulic Downpull of Final Design

pressure which prevailed downstream from the gate was exerted on the inside, or lower side. The resulting downward force, which at times was as large as 100,000 lb, depending on the pressure in the outlet entrance downstream from the gate, contributed a large portion of the downpull on the gate. With a recess, starting a short distance above the gate seat, the increase in the clearance between the seal and the face of the recess lessened the velocity of flow around the seal and reduced the pressure differential.

Tests showed that the recess performed its desired function of reducing the maximum downpull if its depth was made several times larger than the seal extension. At all openings the pressures on the upper seal were nearly balanced. For gate openings less than 40 per cent, the presence of a recess with a uniform depth introduced an undesirable complication. With the downpull force on the top seal eliminated by the recess, the increase in pressure which occurred at small gate openings over that portion of the bottom which underlay the seal was large enough to reverse the direction of the net hydraulic load. The gate was actually subjected to an uplift force large enough to offset its weight.

The effectiveness of the recess in balancing the pressures on the top seal was directly proportional to its depth. By varying the depth in the manner shown in Fig. 3, the reversal of net force on the gate was eliminated and the downpull varied as shown in Fig. 3. The maximum downpull of 70,000 lb occurred at a gate opening of 80 per cent.

Since the dead weight of the gate and lifting mechanism is 90,000, and 10,000 lb, respectively, the maximum total load on the gantry crane, including a downpull of 70,000 lb, will be 170,000 pounds. This is less than the permissible load on the bridge, so the design was accepted as being satisfactory.

The Coaster Gate

The coaster gate is mounted on endless roller trains and is lowered by its own weight in structural guides provided in the face of the dam to an accurate position over the intakes, Fig. 4. The gate consists primarily of a downstream skin plate mounted on horizontal beams which are supported by vertical girders at the sides.

The roller trains around each vertical girder transmit the water load on the gate to tracks on the face of the dam at the inlet. The tracks are fastened to large CB-sections embedded in blockouts provided in the original concrete of the dam, Fig. 5. A rectangular steel framework embedded in the concrete around the circular inlet supports accurately finished seal seats which project slightly from the face of the dam.

When the gate is used to close one of the conduits on the lower tier, it is subjected to a head of some 330 ft, equivalent to 22,000 psf or a total load of approximately 2,750,000 lb. As the gate weighs only 90,000 lb, roller trains were selected to minimize the friction forces so that the gate would close under its own dead weight.

Metal-covered rubber "music-note" seals, Fig. 5, are provided on the downstream face of the skin plate. The design utilizes the flexibility of a rubber hinge which permits close local adjustment but retains the rigidity of metal to support the load of approximately 125 lb per linear in. to which the seals are subjected when the gate is in the closed position. The rubber core also permits simple butt and miter joints to be used in the seal as the load on the seal is converted to axial compression which acts to seal the joints.

Advantage is taken of the pressure differential across the gate to retract the seals when the gate is closing under flow. This eliminates drag on the seals and materially reduces the force resisting closure just as the gate is seated. To accomplish retrac-

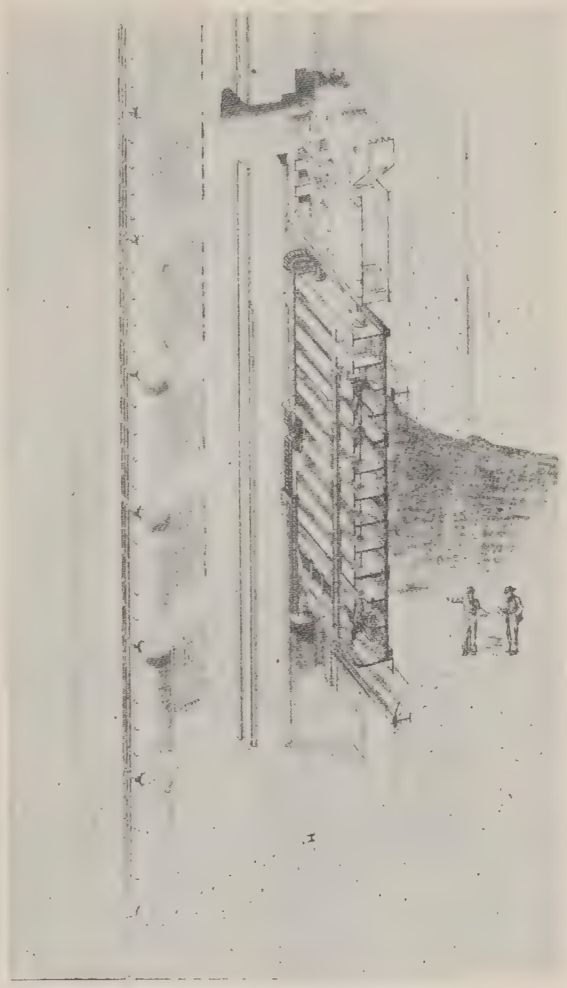


Fig. 4 Typical Coaster-Gate Installation

tion the area immediately back of the seals is closed at the sides. This forms a continuous watertight rectangular chamber having the movable portion of the rubber seal for one of the sides. Pipes connect this chamber, through a two-way valve, to either the reservoir pressure on the upstream side of the gate or to the reduced pressure on the downstream side of the gate.

The valve is operated by the overtravel of the relatively heavy gate-lifting stem. While the gate is supported by the lifting stem, the valve position admits the downstream pressure to the seal chamber and the seal retracts approximately $^3/_{16}$ in. As soon as the gate is seated in the closed position over the inlet, a 4-in. overtravel of the lifting stem reverses the valve to admit reservoir pressure to the seal chamber and the seals are forced into contact with the seat.

Channel-shaped guide shoes at each corner of the gate engage tongues on the guides in the face of the dam, Fig. 5. The distance, face to face, of the guide tongues is held accurately during installation to a clearance of $^1/_4$ in. in each shoe, except for a short distance near the conduit opening. Below an elevation slightly greater than one gate height above the top of the conduit opening the clearance is reduced practically to zero by making longer tongues on the guides. This causes the gate to be squared

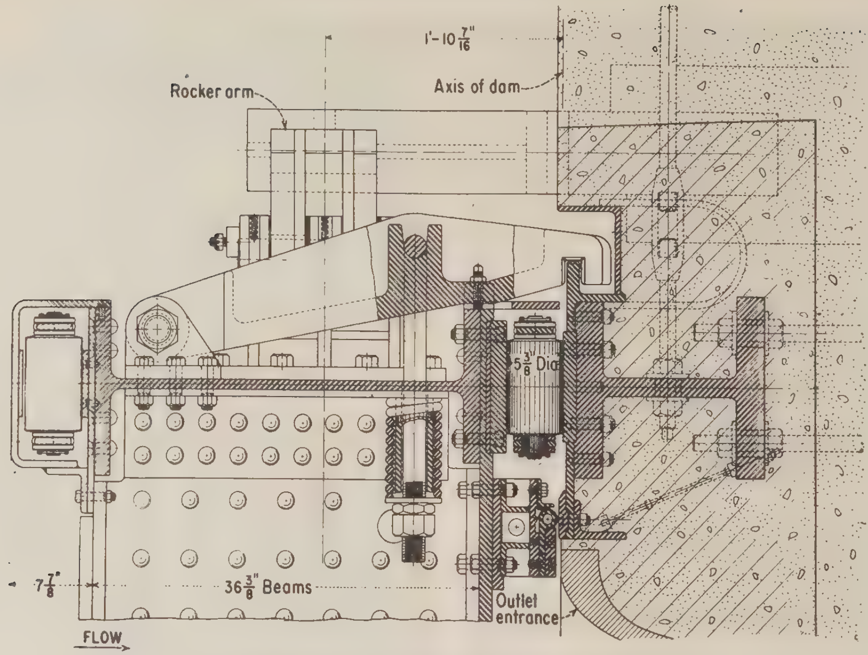


Fig. 5 Shasta Dam Outlet Coaster Gate Slot Section

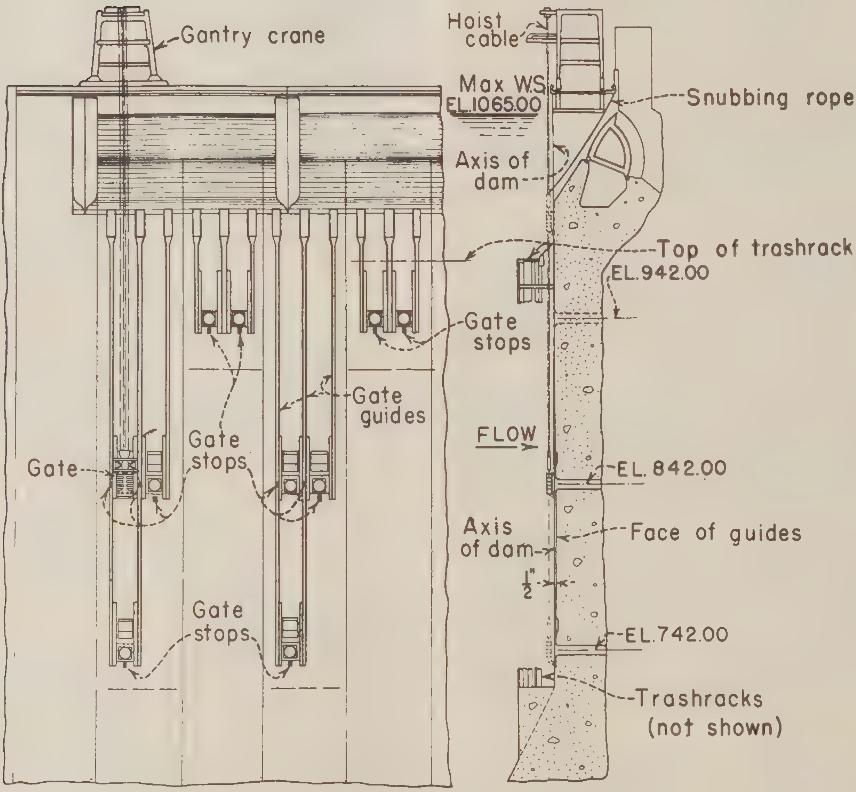


Fig. 6 Location of Outlets and Guides on Upstream Face of Dam

as accurately as possible with the tracks just before it receives the water load when closing under emergency-flow conditions. The shoes are spring-loaded with snubbed springs set with an initial compression sufficient to prevent deflection during this operation.

After the gate receives water load, it is practically impossible to guide the rollers or the gate, because of the extremely high unit contact pressure between the rollers and the track. The spring-loading of the shoes allows the gate to move laterally as much as $^1/_2$ in. in either direction, while closing under load, without excessive binding or probable breaking of the shoes. When the load on the gate is removed, the capacity of the springs is sufficient to square it again.

Operation of Gate

The gate is stored in a covered pit in the top of the dam. Except in case of emergency, it is used only during a few months of the year, when conditions are most favorable for inspection and servicing of the control valves and conduits.

The inlets to the conduits through the dam are arranged so that fourteen sets of guides serve the eighteen conduits. The four conduits of the lower tier, which are 335 ft below the spillway bridge, are served by the same set of guides as four of the conduits of the intermediate tier, which are 235 ft below the spillway bridge, Fig. 6. The other ten conduits are served by individual sets of guides.

The placing of the gate in the guides on the face of the dam requires careful handling. The gate must be lowered approximately 90 ft below the bridge, and 75 ft below the normal water surface before engaging the guides. The upper ends of each set of guides are tapered in the plane parallel to the face of the dam in such a way that in the event the gate is lowered slightly off-center, it is forced to correct itself and allow the channel-shaped shoes on the gate to engage the tongues on the vertical guides.

Engagement of the shoes in the direction normal to the face of the dam is provided for in the following manner:

The gate is lowered in a plane slightly upstream from the face of the dam until it is below the sloping face of the spillway crest. It is then snubbed back into contact with the face of the dam by snubbing ropes, Fig. 6, from the downstream edge of the bridge structure. Continued lowering will then cause the gate shoes to engage the guides and the operation can continue.

A gate-lifting frame, Fig. 7, with a semiautomatic grappling mechanism is used to handle the gate. After the gate is seated in front of a conduit, the lifting frame may be released from the gate and hoisted to the bridge. This arrangement is necessary as the gate is required to remain in place in front of a conduit for several weeks at a time and the crane must be free to perform its other functions.

The position of the gate cannot be observed after it is lowered into the water. Consequently, a safety device is provided to indicate proper engagement of the gate shoes. This device consists of ropes attached to a tripping mechanism at each lower corner of the gate. When the shoes engage the tongues of the guides, the rope is released by the tripping mechanism. Two men are stationed on the bridge to pay out the telltale ropes as the gate is lowered. If both ropes are released after lowering the gate a reasonable distance below the top of the guides, it is evident that the guides have been engaged properly.

Lowering is continued until the gate is supported by stops on the guides at the inlet of the conduit. The stops are located so that the gate comes to rest exactly in the closed position in front of the inlet. The lifting frame continues downward approximately 4 in. until it comes to rest on the top of the gate. The 4-in. travel of the lifting stem down into the gate causes the gate seals to close. The same motion causes the grappling mechanism in the lifting frame to release and a slack-cable limit switch, located on the crane, stops the lowering motion of the hoist.

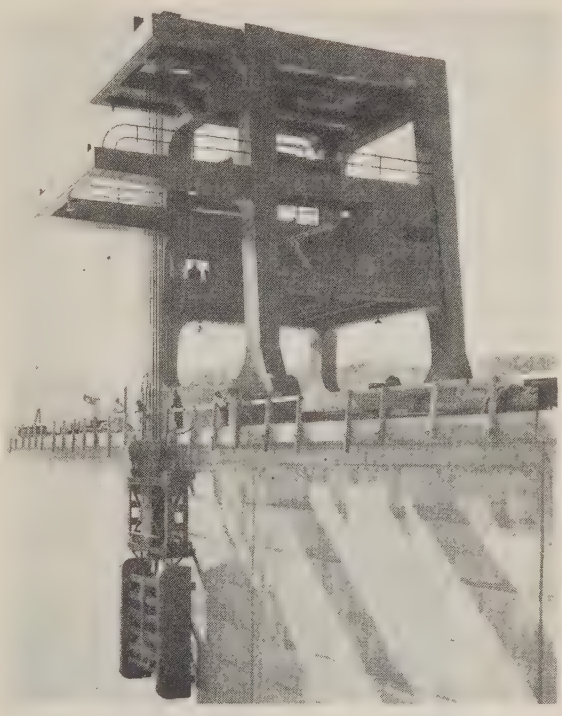


Fig. 7 Outlet Conduit Coaster Gate and Gate-Lifting Frame With Semiautomatic Grappling Mechanism

If only one conduit is served by a set of guides, a single stop is provided in the face of the dam directly below the conduit opening. Where two conduits are served by the same set of guides, that is, where one conduit is located above the other, a single stop is provided directly under the lower one. Stops for the upper one are located at the sides of the conduit, Fig. 6. Rocker arms, or pawls, are provided at each side of the gate and are interconnected by a linked tension bar. When it is desired to close the upper conduit, the bar is extended and the rocker arms are pushed outward to a position where they will contact the stops at the sides of the upper conduit and act as an equalizer as the gate comes to rest. If the lower conduit is to be closed, the bar is retracted so that the rocker arms, Fig. 7, will clear the stops at the side of the upper conduit, and the gate comes to rest on the signal stop below the lower conduit.

To remove the gate the operations described are reversed, except that the telltale ropes are attached to the tripping mechanism at each lower corner of the lifting frame.

Gantry Crane

The crane, an outdoor, traveling, gantry type, Fig. 8, is electrically operated and is provided with a 125-ton fixed hoist which handles the gates, and a 25-ton trolley hoist which handles parts of gates during assembly. The latter is also used infrequently to install stop logs in front of the power penstocks. The 125-ton capacity was determined by the installation and servicing conditions of the coaster gates for the main power-penstock inlets. These inlets are located in the curved portion of the dam, where the solid construction will support practically any crane load.

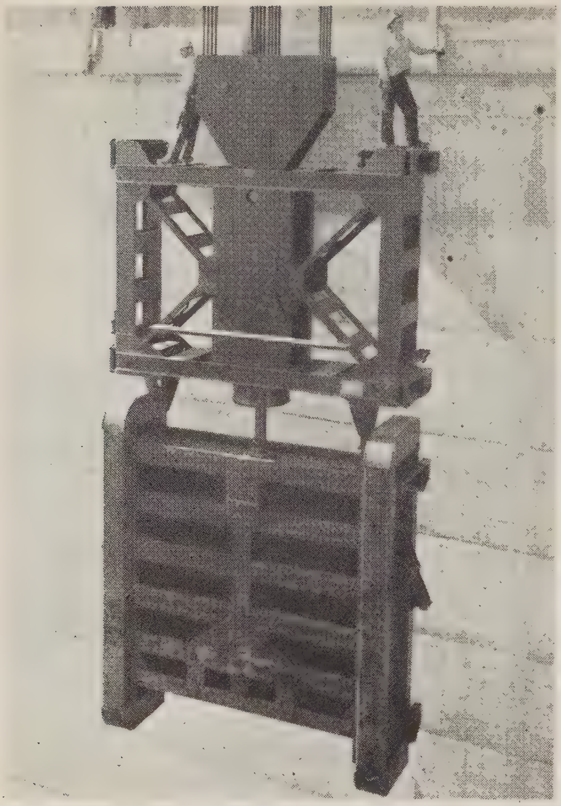


Fig. 8 Gantry Crane in Use With Lifting Mechanism and Outlet Conduit Coaster Gate Suspended From 125-Ton Fixed Hoist

As previously mentioned, the bridge structure over the spillway portion of the dam in which the river outlets are located made it economically desirable to limit the crane load at this point to only a fraction of the hoist capacity in order to avoid a heavy and expensive structure which would not be otherwise required. The crane can transfer a load from any point on the dam to the overhung position upstream from the face of the dam. All gates are installed and operated from the overhung position.

The crane operates on 27-ft-gage tracks. The tracks are straight for approximately 375 ft on the bridge structure over the spillway section at the center of the dam. The remainder of the track has a 2500 ft radius of curvature for a distance of 1425 ft along the roadway on top of the right abutment.

The combination of straight and curved track made it impractical to use the conventional drive with one motor driving a shaft leading to the trucks on either side. Instead, separate motors were mounted on each of four trucks with 50 per cent of the track wheels driving. Direct current for operation of the four 8-hp shunt-wound motors is provided by a generator set. The use of direct current was made necessary by the unequal loading of the motors.

Accurate spotting of the gates is provided by direct-current-motor-operated hoists and Maxspeed control. In case the gate should encounter some obstruction in the guides, a special type of control was provided for the hoist. This control limits the maximum lifting effort when the crane is located on the bridge. If a conventional hoist with alternating-current motor and standard control had been provided, the maximum lifting effort on the crane hook under a 275 per cent breakdown torque of the motor would have been approximately 350 tons. The modified Maxspeed control limits the lifting effort at the hook to 175 tons. The control is the full-magnetic, reversing, master type. It has definite time-limit acceleration relays and six speeds in each direction of operation. Due to the extremely long lift of 360 ft, a mechanical load brake was not considered feasible and direct-current dynamic braking was provided.

A panel of indicating lights in the operator's cage warns him when the gate is approximately 16 ft above the closed position in front of any conduit. A lighting system, including floodlights, was installed on the crane so that the gates could be operated at night.

Speed and Feed Selection in Carbide Milling With Respect to Production, Cost, and Accuracy

By HANS ERNST[1] AND MICHAEL FIELD[2]

The present paper reports findings of a research on milling cast iron with carbide-tipped cutters in addition to those presented previously in 1945 by Michael Field and W. E. Bullock.[3] An important finding of the earlier work was the existence of a definite maximum tool-life point, i.e., at 290 fpm, in relationship of cutting speed to tool life, when milling a cast iron of medium hardness (Bhn 190). Subsequently, it was found that a similar maximum tool-life point, i.e., 200 fpm, occurs when milling a considerably harder and less machinable cast iron (Bhn 240). An important outcome of the earlier phases of the research was the development of a combination of tool angles which gave much greater tool life and a considerably higher production rate. The original work was done with single-point tools, whereas a multitooth cutter was subsequently developed, details of which and the results obtained are included in the paper. A comprehensive account is given of the recent studies, illustrated by numerous test curves, and a practical analysis of milling costs.

In a previous paper[3] by Michael Field and W. E. Bullock, results were given in part of a research on the milling of cast iron with carbide-tipped cutters, which had been conducted at the University of Cincinnati. The present paper covers certain subsequent findings of this research, together with a further elaboration and application of earlier findings.

One of the important findings reported in the previous paper was the existence of a definite maximum tool-life point in the relationship of cutting speed to tool life, when milling a cast iron of medium hardness (190 Bhn), designated as Meehanite A. This relationship is reproduced in the upper curve of Fig. 1. Here it will be seen that the average of the tool-life values for 210 fpm falls well to the left of the average of the values for 290 fpm. At higher cutting speeds all the values of tool life are also to the left of the value for 290 fpm and fall approximately on a line representing the equation $VL^{0.55} = 7800$, where V = cutting speed (ft per min) and L = tool life (cu in. of metal removed).

In a subsequent series of tests, it has now been found that a similar maximum tool-life point occurs when milling a considerably harder and much less readily machinable special grade of alloy cast iron, designated as Meehanite C-304B. The Brinell hardness of this material was 240; its chemical composition is given in Table 1, together with that of Meehanite A.

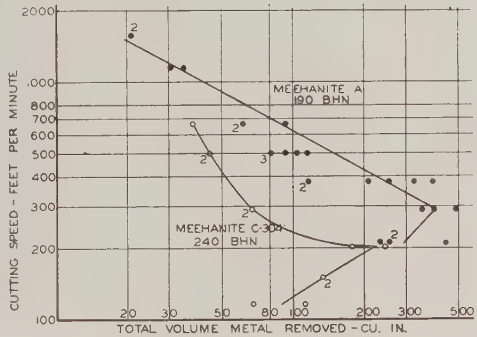


Fig. 1 Cutting Speed Versus Total Volume of Metal Removed to Dull Tooth With Axial Rake of +3 Deg, Radial Rake of +3 Deg, and Corner Angle of 30 Deg

(Width of cut = 6 in.; depth of cut = 0.187 in.; feed per tooth = 0.015 in.; carbide, 44A.)

TABLE 1 CHEMICAL PROPERTIES OF CAST IRONS USED IN CUTTING TESTS

	Meehanite A, per cent	Meehanite C-304B, per cent
Total carbon	2 85-3.15	2.90 3 20
Silicon	1 20-1.50	2.45-3 00
Phosphorus	0.12 Max	0.18 Max
Sulphur	0.20 Max	0.20 Max
Manganese	0.80-1.20	0.75-1.00
Chromium	....	1.00
Copper	....	
Molybdenum	....	0.50

The tool-life values for Meehanite C-304B, expressed in cubic inches per tool grind, are shown by the lower curve in Fig. 1. The maximum tool-life point here occurs at about 200 fpm in contrast to 290 fpm for the softer Meehanite A. The portion of the line to the left of this point is quite definitely curved, again in contrast to the straight line of Meehanite A.

The existence of a rather critical maximum tool-life point (which has now been found in a considerable number of tests) is a matter of real importance in practical milling. It indicates clearly the impossibility of predicting tool-life values in the normal speed range by extrapolation from the results of tests above 500 fpm.

Another important outcome of the work[3] previously reported, was the development of a combination of tool angles which gave a much greater tool life (in terms of metal removed per grind), and a considerably higher production rate, than could be obtained with the angles used in conventional practice. The initial work was done with single-point tools in which the effective angles could be readily changed. In later work a multitooth cutter was used. One such cutter is shown in Fig. 2.

The basis for this development was established in 1937, in an investigation covering the performance of face-milling cutters with very large corner angles. In a report of this investigation by M. Kronenberg,[4] it was shown that very large corner angles had

[1] Research Director, The Cincinnati Milling Machine Company, Cincinnati, Ohio. Fellow A.S.M.E.

[2] Research Fellow, University of Cincinnati, Cincinnati, Ohio. Jun. A.S.M.E.

[3] "Milling Cast Iron With Carbides," by Michael Field and W. E. Bullock, *Mechanical Engineering*, vol. 67, Oct., 1945, pp. 647–658.

Contributed by the Research Committee on Metal Cutting Data and Bibliography and presented at the Fall Meeting of the Cincinnati Section, Cincinnati, Ohio, Oct. 2–3, 1945, of The American Society of Mechanical Engineers.

[4] Research Department, Cincinnati Milling Machine Company (confidential report).

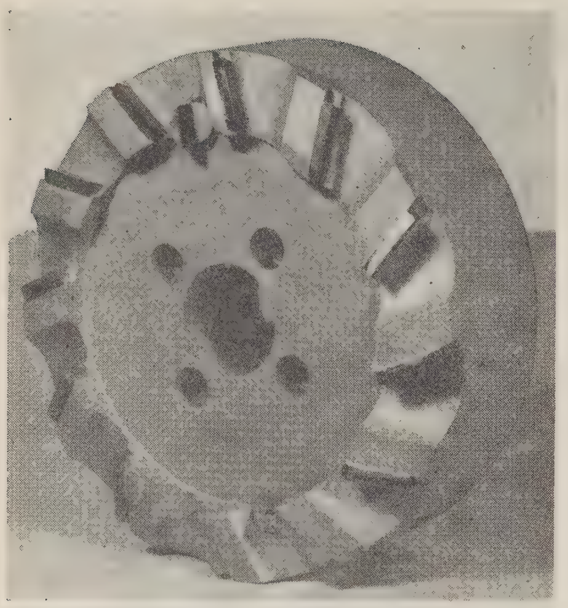


Fig. 2 Cutter Developed in Research, Capable of Operating at Extremely High Feed Rates

(The 12 major teeth have an axial rake of +15 deg, a radial rake of —30 deg, a 75 deg corner angle, a true rake of +6 deg, and a face angle of 5 deg. The finished surface is produced by a separate tooth which has a face $^3/_4$ in. long and extends axially several thousandths of an inch beyond the roughing teeth.)

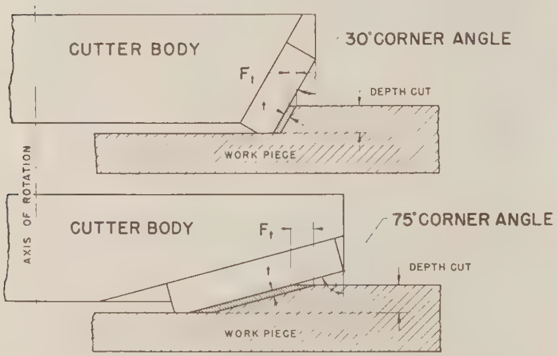


Fig. 3 Effect of Corner Angle on Feed per Tooth for a Constant Value of Undeformed Chip Thickness

the unique property of distributing the work of chip removal over a much greater length of cutting edge than conventional corner angles of zero to 45 deg, thus permitting the use of a very high feed per tooth, and a consequent high rate of metal removal.

The upper view in Fig. 3 shows in diagrammatic fashion a face-milling cutter with a conventional 30 deg corner angle. In this figure the workpiece is assumed to be feeding to the left, and the small cross-hatched area represents the cross section of the metal which will be removed by the next tooth. Thus for a given feed per tooth F_t, the thickness of the undeformed chip will have the value t.

In the lower view in Fig. 3 the corner angle of the cutter is 75 deg. For the same value of the undeformed chip thickness t, it is evident that in this case the feed per tooth F_t must have a much larger value. It is also clear that the length of cutting edge engaged with the work is here much greater, even though the depth of cut is the same as in the upper view. Thus for a given value of t, the cross-sectional area of metal removed per tooth will be much greater even though the load on each element of length of the cutting edge remains virtually unchanged.

In the numerical example, illustrated in Fig. 4, it is shown that for a constant value of 0.013 in. for the undeformed chip thickness, the feed per tooth with the 30 deg corner angle cutter is 0.015 in., while with the 75 deg corner angle it is 0.050 in. This represents a rate of production 333 per cent that of the lower corner angle.

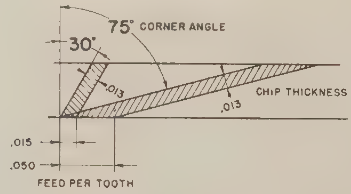


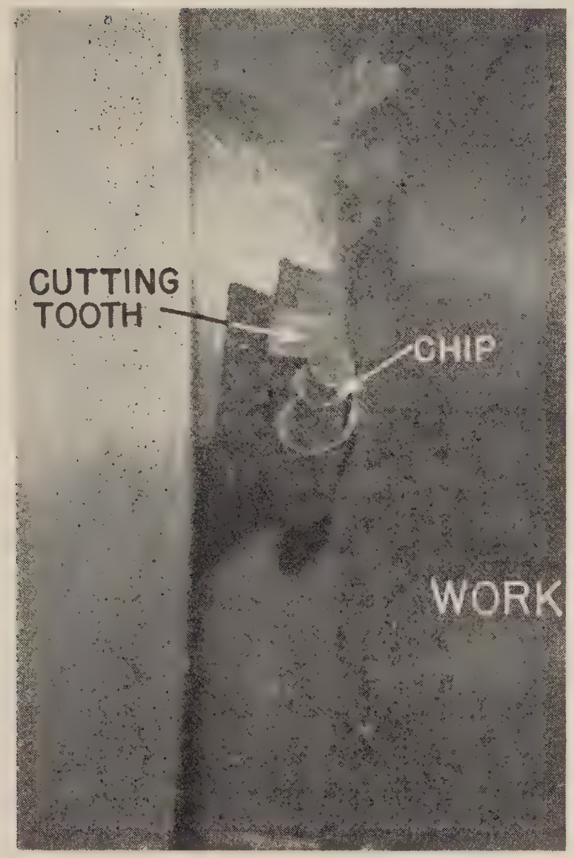
Fig. 4 Example Showing Increase of Feed per Tooth From 0.015 In. to 0.050 In. as Corner Angle Is Increased From 30 to 75 Deg for a Constant Undeformed Chip Thickness of 0.013 In.

In practice it is difficult to utilize such large corner angles and high potential feed rates with the present commercially used numerically small values of axial and radial rake angle (either positive-positive, negative-negative, or positive-negative). Problems of chip disposal, high impact loads, smoothness of finish, flatness, etc., are all limiting factors. These difficulties have been overcome to a high degree of satisfaction with the combination of angles and arrangement developed in this investigation; viz., large negative radial rake (—30 deg to —45 deg); large positive axial rake (+15 deg to +20 deg), and large corner angle (70 deg to 80 deg). In the 12-tooth cutter, shown in Fig. 2, the values used are axial rake, +15 deg; radial rake, —30 deg; and corner angle 75 deg, giving a true (or resultant) rake of +6 deg. This designation may be abbreviated, +15, —30, 75, +6. (a, r, c, t).

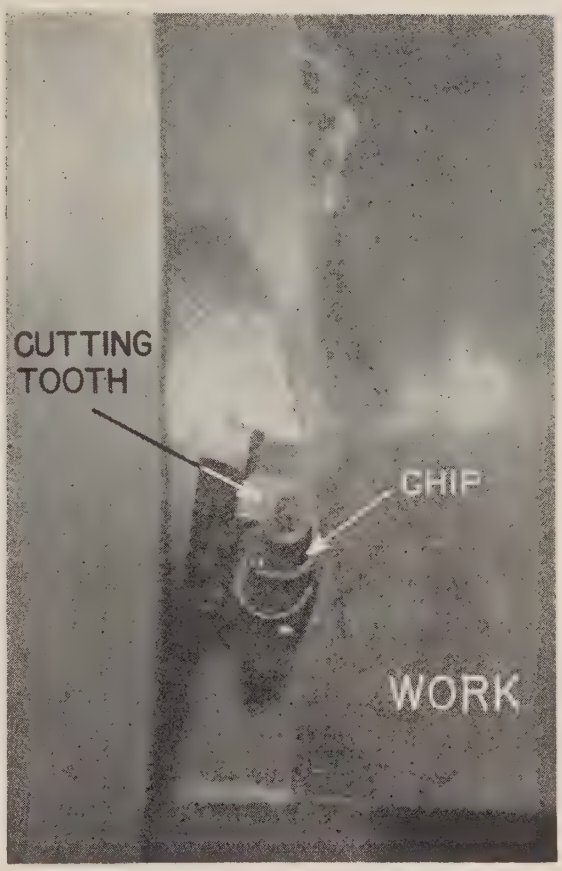
The very large feed per revolution obtainable with such a cutter ($^1/_2$ in. to 1 in. or more, depending on number of teeth and material cut) presents a difficult problem in surface finish. This is particularly true with large carbide-tipped cutters owing to the difficulty of grinding the ends of all teeth exactly alike. This problem has been solved in the present instance by providing a separate finishing tooth with fine axial adjustment and a face sufficiently broad to cover at least the feed per revolution; it is set so as to extend a few thousandths of an inch beyond the "highest" of the roughing teeth. This separate finishing tooth is shown at the top in Fig. 2.

With this arrangement, the main teeth are ground so as to cut on their angularly disposed edges only. The adjacent edges do not lie in a radial plane but are sharply relieved so as to clear the finished surface, the latter being produced entirely by the separate finishing tooth. By separating the functions of roughing and finishing, it is also possible to use for the finishing tooth a very hard and abrasion-resistant grade of carbide; and if desired, a large positive axial rake angle, as this tooth does not have to withstand an impact load.

A characteristic feature of this cutter is the large positive angle of inclination (32 deg in the cutter shown in Fig. 2). The angle of inclination is the angle between the cutting edge and a plane normal to the path of motion. This large positive angle produces a helically curled chip which flows radially out of the cutter body, even with very high values of feed per tooth. The electronic flash photographs of a single-tooth cutter with this



(a)



(b)

FIG. 5 FLASH PHOTOS SHOWING UNUSUAL HELICAL CHIP FORMATION IN CAST IRON, WITH CUTTER HAVING AXIAL RAKE OF +15 DEG, RADIAL RAKE OF —30 DEG, AND CORNER ANGLE OF 75 DEG

(In *b*, the chip is somewhat further developed than in *a*.)

angle combination, Figs. 5 (*a*) and 5 (*b*), made with an exposure time of about 0.000004 sec, clearly show this helical chip formation.

In early tests with the multitooth cutter, cast-iron test blocks were milled at feed rates of over 100 ipm. Such a cut is shown by the high-speed flash photograph, Fig. 6.

SELECTION OF FEED AND SPEED IN PRODUCTION MILLING

In practical applications of carbide milling, the question naturally arises as to what factors actually determine the speeds and feeds which can properly be used. The performance illustrated in Fig. 6, although spectacular, may not be possible, or desirable, on a given job. In any milling operation certain quality specifications such as flatness, breakout, and surface finish must first be satisfied; any of these requirements may limit the feed rate to a value appreciably lower than the maximum capacity of the machine or cutter.

In order to study the quality of surface obtainable when milling with the cutter shown in Fig. 2, the head and pan surfaces of an automobile cylinder block were milled on a special fixed-bed milling machine, Fig. 7. This machine was provided with a hydraulic feeding mechanism by means of which the table could be operated at feed rates up to 300 ipm. The spindle was driven by a separate 40-hp motor. The cylinder block was held in a fabricated fixture. The cylinder-block head surface was $7^1/_4$ in. wide × $23^1/_2$ in. long, while the pan sur-

FIG. 6 CAST-IRON BLOCK BEING MILLED AT FEED RATE OF 100 IPM

(Width of cut = 6 in.; depth of cut = 0.150 in.; cutter, $9^5/_8$ in. diam, 12 teeth, +15, —30, 75, +6; cutting speed, 350 fpm.)

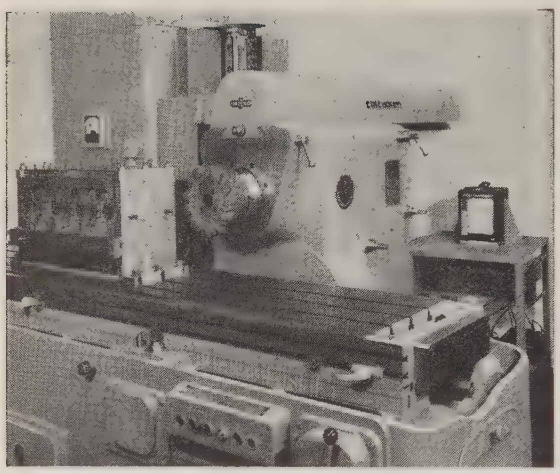


Fig. 7 Fixed-Bed Milling Machine Used for Milling Tests on Cylinder Blocks With Special Cutter Shown in Fig. 2

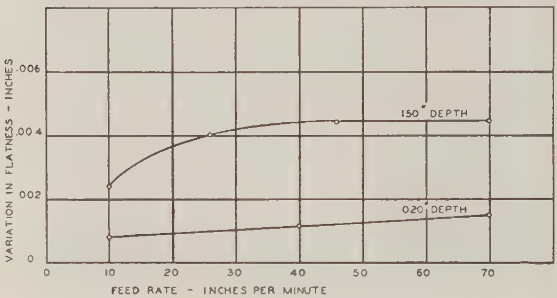


Fig. 8 Variation in Flatness Versus Feed Rate
(Milling head surface of cast-iron cylinder block with special cutter shown in Fig. 2.)

face was $11^1/_4$ in. wide × 25 in. long. Cuts were made at 280 fpm, at feed rates up to 70 ipm. The thin wall sections and cored openings in the block presented a difficult test for these high feed rates.

Flatness. The variations in flatness obtained in milling the head surface of the block at both 0.150-in. and 0.020-in. depths are shown in Fig. 8. Taking a roughing cut 0.150 in. deep, the departure from surface flatness was 0.0044 in. at a feed rate of 70 ipm. This may be considered satisfactory, since such a heavy cut would always be followed by a finishing cut. At 0.020 in. depth, the flatness error never exceeded 0.0015 in. at any feed rate up to 70 ipm.

Breakout. In milling cast iron, the breakout on the work at the side where the cutter teeth emerge from the cut may possibly limit the maximum feed. The relation of breakout to feed rate, measured at the cylinder bores while taking a 0.150-in-depth cut, is shown in Fig. 9. The cylinder bores had 0.125-in. finishing stock, so that the 0.070-in. breakout at 70 ipm left a comfortable margin of safety.

Surface Finish. The surface finish obtained in milling the pan surface of the cylinder block is shown in Fig. 10. The surface finish improved somewhat as the feed rate was reduced. However, even the worst surface finish (230 microinches at 70 ipm, with a depth of 0.150 in.) would be satisfactory as a roughing cut. Figs. 11 (*a*) and (*b*) are reproductions of Faxfilm replicas showing the finish obtained at 70 and 26 ipm, respectively. By careful attention to the shape and smoothness of the cutting edge on the finishing tooth, a very high quality of surface can be obtained on finishing cuts.

Tool Life. There are, of course, other factors than quality specifications that influence the choice of speed and feed. Chief among these is tool life.

Fig. 12 shows the relationship between tool life (measured in terms of cubic inches of metal removed per sharpening) for a single-tooth cutter having the combination of angles developed in this investigation, and used in the multitooth cutter, Fig. 2. In this case the work material was Meehanite A. Here again a definite maximum tool-life point occurs at about 300 fpm; both above and below this speed the tool life rapidly decreases. Higher cutting speeds permit higher feed rates, and, conse-

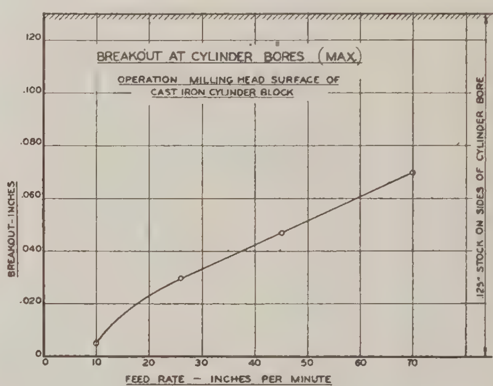


Fig. 9 Maximum Breakout at Cylinder Bores Versus Feed Rate
(Milling head surface of cast-iron cylinder block with special cutter shown in Fig. 2. Depth of cut, 0.150 in.)

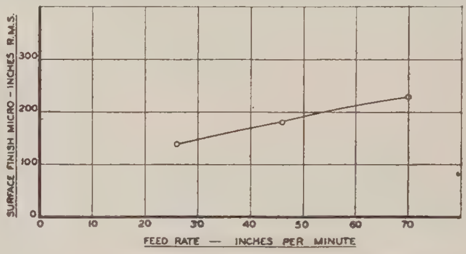


Fig. 10 Surface Finish Versus Feed Rate
(Milling pan surface of cast-iron cylinder block with special cutter shown in Fig. 2. Depth of cut, 0.150 in.)

(*a*) (*b*)

Fig. 11 Surface Finish Obtained on Cast-Iron Cylinder Block With Special Cutter Shown in Fig. 2
(Depth of cut, 0.150 in. Fig. *a:* 70 ipm feed rate, 230 microinches, rms. Fig. *b:* 26 ipm feed rate, 140 microinches rms; magnification, ×20.)

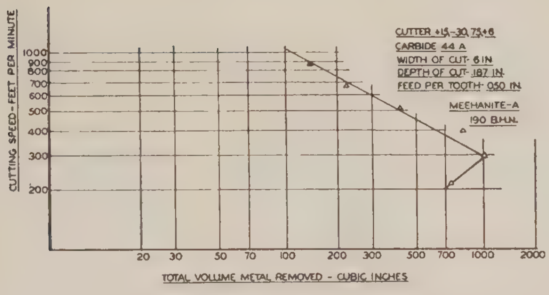


Fig. 12 Cutting Speed Versus Total Volume of Metal Removed to Dull Tooth With Axial Rake of +15 Deg, Radial Rake of —30 Deg, and Corner Angle of 75 Deg

quently, higher production, but as is evident from the curve, this can be gained only at the expense of cutter life.

Cost of Milling and Rate of Production

From the foregoing discussion it is evident that the selection of speed and feed in milling must take into consideration a number of factors such as tool life, surface quality, production rate, breakout of trailing edge of work, etc., any one of which may be of paramount importance in a particular case. In the final analysis, the best combination must be determined by economic considerations; in other words, by the best balance between production rate and cost per piece. Hence it is necessary first to study in detail all of the elements entering into the cost of production.

In the paper[3] previously mentioned, the factors that determine milling cost were combined into the following equation

$$C = \underbrace{\left[\left(M_{LO}\right)\left(T_m + \frac{T_o}{N_L} + \frac{T_r}{N_S}\right)\right]}_{\text{Milling cost}} + \underbrace{\left[C_u\right]}_{\text{Initial cost of cutter}} + \underbrace{\left[\left(G_{LO}\right)\left(\frac{T_S}{N_S} + \frac{T_B}{N_B}\right) + C_a + W\right]}_{\text{Cutter preparation cost}}$$

where

C = total cost to mill one piece, dollars
M_{LO} = labor + overhead on milling machine, \$/min
G_{LO} = labor + overhead on cutter grinder, \$/min
B_{LO} = labor + overhead on brazing unit (assumed = G_{LO}), \$/min
T_m = total time to mill one piece, min
= feeding time + rapid traverse time + loading and unloading time
T_o = original set-up time on milling machine, min
N_L = total number of pieces in lot
T_r = time to change and reset cutter, min
T_s = time to resharpen cutter, min
N_s = number of pieces milled per cutter sharpening
T_B = time to rebraze teeth (or reset blades), min
N_B = number pieces milled per brazing (or resetting)
C_u = original cutter cost per workpiece
C_a = carbide (or blade) cost per workpiece
W = wheel (diamond or abrasive) cost per sharpening

From this equation the production rate in pieces milled per 60-min hr (P) can be derived

$$P = \frac{60}{T_m + \frac{T_o}{N_L} + \frac{T_r}{N_s}}$$

The cost equation also indicates that the two major factors which determine the cost of milling are the actual milling cost (or metal-removal cost), and the cutter preparation cost. The original cutter cost per piece milled is generally negligible, except for short jobs that require special cutters.

In order to illustrate the application of a cost analysis to speed and feed selection in practical cases, specific examples will be presented to show the effect of such factors as work material, cutting speed, feed per tooth, number of teeth in the cutter, loading and unloading time, number of pieces in the lot, and the length of cut. The following assumptions will be common to all of the examples:

Operation: Face mill cast-iron block
Cut dimensions: 6 in. wide, $^3/_{16}$ in. deep, 20 in. long (except where noted)
Cutter: 10 in. diam, +15 deg axial rake, —30 deg radial rake, 75 deg corner angle, +6 deg true rake, 12 teeth (except where noted)
Feed per tooth: 0.050 in. (except where noted)
Approach to work: 10 in.
Overtravel of cutter past work: 10 in.
Number of pieces in lot: 10,000 (except where noted)
Time to load and unload: 1 min (except where noted)

Labor cost on milling machine..................	\$1.00 per hr
Overhead on milling machine..................	\$4.00 per hr
Labor cost on cutter grinder..................	\$1.00 per hr
Overhead on cutter grinder..................	\$1.25 per hr

Then M_{LO} = \$.0832 per min
G_{LO} = \$.0374 per min

Rapid traverse rate = 300 ipm

T_o = 60 min
T_r = 10 min
T_s = 20 min per tooth
T_B = 5 min per tooth
$N_B = 4\ N_s$

$$C_u = \frac{\text{Original cutter cost}}{\text{No. of pieces milled with cutter}} = \frac{\$200}{50{,}000} = \$0.004$$

$$C_a = \frac{\$2.00 \times \text{No. teeth}}{3\ N_B}$$

$$W = \frac{\$0.20 \times \text{No. teeth}}{N_s} \text{ (for diamond grinding wheel)}$$

The time to mill one piece T_m will then be

$$T_m = \frac{10 + l}{f} + \frac{10 + l + 20}{r} + T_L$$

where

l = length of cut, in.
f = feed rate, ipm
r = rapid traverse rate, ipm
T_L = time to load and unload, min.

The number of pieces milled before the cutter has to be resharpened, N_s, will be

$$N_s = \frac{\text{No. teeth} \times L}{\text{Cubic inches removed per piece}}$$

where

L = tool life, cu in. removed to dull one tooth, obtained from cutting-speed versus tool-life curves (such as Fig. 12)

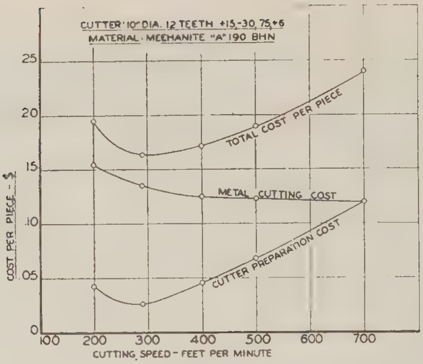


Fig. 13 Analysis of Total Milling Cost in Terms of Its Principal Components

(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide, 20 in. long.)

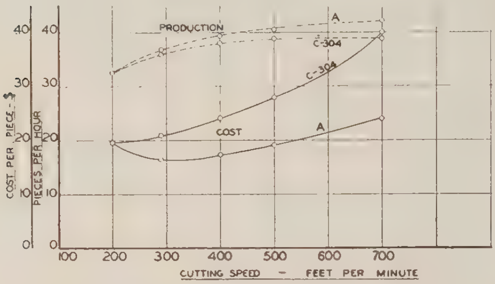


Fig. 14 Production Rate and Cost per Piece Versus Cutting Speed

(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide, 20 in. long.)

The manner in which the cutting cost components vary with cutting speed, in milling Meehanite A, is shown in Fig. 13. The cutter preparation cost is a minimum at 290 fpm, because at this speed the tool life was a maximum (see Fig. 12). The metal-cutting cost drops rapidly as the speed is increased from 200 to 400 fpm, but at higher speeds the milling cost remains practically constant because the increase in feed rate is offset by the time lost in changing and resetting cutters which results from the decreased tool life. The total cost per piece is the sum of the cutter preparation cost and the metal-cutting cost. This total cost is likewise seen to be a minimum at 290 fpm.

But the cost per piece may not be the most important factor. Under certain conditions, the production rate may be equally important as the cost per piece, and in some cases may be even more important. The production rate required on a machine may be determined by the capital investment and by the inconvenience or delay in obtaining additional equipment. For example, it may be desirable to sacrifice tool life by operating at higher cutting speeds in order to obtain a definite production rate with a given number of milling machines.

The production rate, measured in number of pieces milled per 60-min hr, and the cost per piece, vary with cutting speed as shown in Fig. 14. The production rate for both of the materials tested (Meehanite A and Meehanite C-304) rises rapidly as the speed is increased from 200 to 400 fpm, but at the higher speeds these increases are less pronounced owing to the increase in time lost when changing cutters. The total cost per piece for Meehanite C-304 is practically constant between 200 and 300 fpm, but at the higher speeds the cost rises rapidly. The differences in cost and production between the two grades of Meehanite are due to the differences in their machinability characteristics. From Fig. 14 the speed which gives the best combination of cost and production can be readily determined. In milling Meehanite A it is obvious that 300 or 400 fpm would be a more economical speed than 200 fpm.

Quantitative comparisons may also be made. Thus in milling Meehanite C-304, the cost at 300 fpm would be 1.07 times as high as at 200 fpm, whereas the production rate at 300 fpm would be 1.11 times as high. Thus at 300 fpm, for a 7 per cent increase in cost, one could obtain a production increase of 11 per cent. In a like manner, the economy of operation at other cutting speeds can be compared.

In the case just discussed, the feed per tooth was held constant at 0.050 in. A thorough analysis of cost and production, however, must take into consideration a wide range of feeds per tooth as well as cutting speeds.

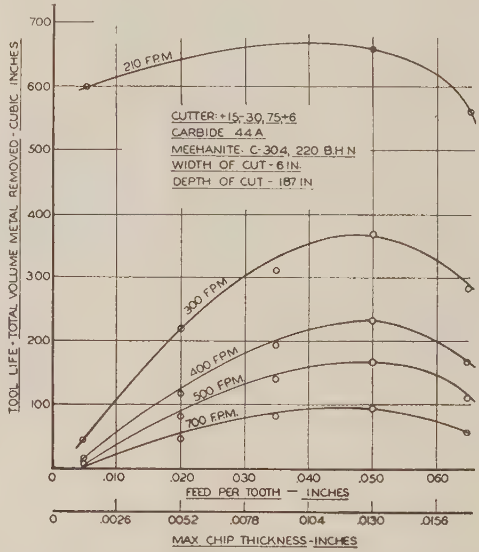


Fig. 15 Tool Life, in Terms of Total Volume of Metal Removed Versus Feed per Tooth, at Constant Cutting Speed

In Fig. 18 of the paper by Field and Bullock[3] (reproduced here as Fig. 15), the interrelation of feed per tooth and cutting speed, with tool life, is shown for the 75-deg corner angle cutter when milling Meehanite C-304. Here the tool life (expressed in cubic inches of metal removed per tooth) is plotted against feed per tooth at constant cutting speeds. Note that here the maximum tool life occurs at 0.050 in. feed per tooth, regardless of the cutting speed. In Fig. 16 the cost and production rate have been plotted against feed per tooth, at constant cutting speeds, using the tool-life data from Fig. 15. Inspection of Fig. 16 shows that the minimum cost likewise occurs at 0.050 in. feed per tooth for all cutting speeds.

Many interesting facts are revealed by a study of the curves in Fig. 16. For instance, it will be noted that for cutting speeds of 200 to 400 fpm, the cost curves are relatively flat with feeds per tooth between 0.040 in. and 0.060 in., but rise rapidly with feeds per tooth below 0.040 in. For cutting speeds above 400 fpm, the cost curves show a sharp minimum at about 0.050 in. feed per tooth. In contrast, all the production-rate curves rise rapidly with feeds per tooth up to 0.040 in., while above 0.040 in. only those for cutting speeds of 400 fpm and under continue to rise appreciably. At 500 fpm the production-rate curve is rela-

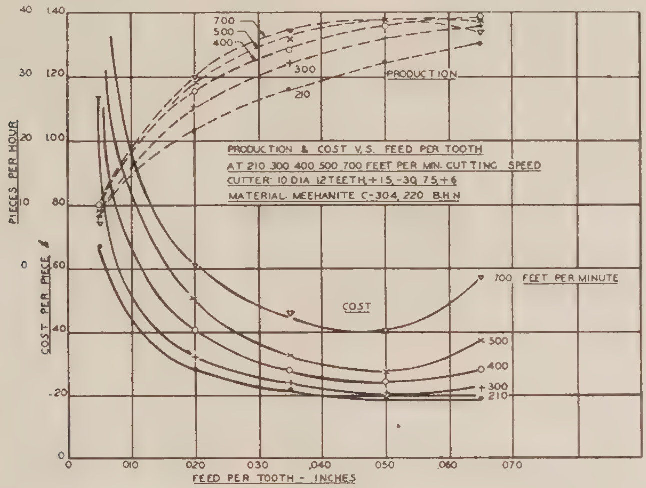


Fig. 16 Pieces per Hour and Cost per Piece Versus Feed per Tooth at Constant Cutting Speeds
(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide, 20 in. long.)

tively flat above 0.040 in. feed per tooth, while at 700 fpm, the curve actually peaks at a feed per tooth of 0.050 in. In this neighborhood the vertical interval between the production-rate curves decreases as the speed is increased, while the interval between the cost curves rapidly increases.

From the curves in Fig. 16 it is therefore apparent that for this particular example of cutter and work, the most desirable combination, from the standpoint of both cost and production, is with a cutting speed of about 300 fpm, and a feed per tooth of about 0.050 in. to 0.055 in. If the cutting speed were increased from 300 to 400 fpm, the cost per piece would be increased 10 per cent, while the production rate would increase only 5 per cent. On the other hand, if the cutting speed were reduced from 300 to 200 fpm, the cost would be decreased only 5 per cent, while the production rate would decline 10 per cent.

With a cutter of $9^5/_8$ in. mean diam, (as in Fig. 2), a 300-fpm cutting speed would correspond to a rotation of 120 rpm. With 12 teeth in the cutter and a feed per tooth of 0.055 in., the feed rate would then be 120 × 12 × 0.055, or 80 ipm. Under these conditions, the total volume of cast iron, Meehanite C-304, which could be removed before the cutter would need resharpening (using the data from Fig. 15), would be about 4200 cu in. or approximately 1200 lb.

If a cutting speed of 400 fpm were used, the feed rate would be 107 ipm, but the total volume of metal removed before sharpening would be reduced to 2640 cu in. Under these conditions, using the data from the curves in Fig. 16, it can be shown that the production rate would be increased 5.4 per cent while the cost per piece would be increased 22 per cent.

From the tool-life curve in Fig. 12, it is interesting to note that if Meehanite A had been used instead of Meehanite C-304, the volume of metal removed before sharpening, at a cutting speed of 300 fpm, would be 12,000 cu in., or approximately 3100 lb.

Other Factors Influencing Production Rate and Milling Cost

There are still other important factors than quality specifications and tool life, however, that influence production rate and cost of milling.

Number of Teeth in Cutter. In Fig. 17 is shown the effect of the number of teeth in the cutter when using an optimum combination of cutting speed and feed per tooth. Here it will be

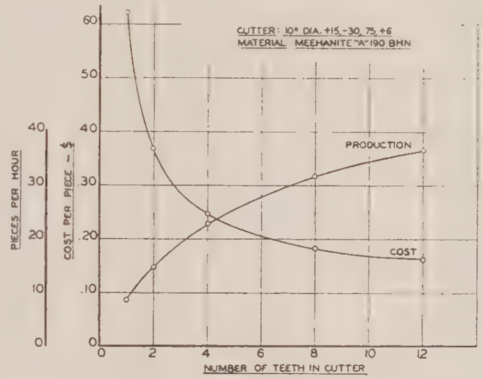


Fig. 17 Pieces per Hour and Cost per Piece Versus Number of Teeth in Cutter
(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide, 20 in. long.)

seen that there is a rapid drop in cost and a rapid rise in production rate as the number of teeth is increased from 1 to 4 in this 10-in-diam cutter. As the number of teeth is further increased, both the cost and the production curves flatten out because the feeding time eventually becomes small in comparison with other fixed portions of the cycle time, such as the rapid traversing time, and the loading and unloading time.

Loading Time. The effect of loading and unloading time on cost and production is shown in Fig. 18, for two conditions: (*a*) 280 fpm cutting speed and 65 ipm feed rate; and (*b*) 500 fpm cutting speed, and 105 ipm feed. From this chart it will be seen that the cost at 500 fpm is a constant amount higher than at 280 fpm, regardless of the loading time. Furthermore, the cost per piece is a linear function of the loading time. The production-rate curves are flat at a high loading time because the feeding time here is short compared to the time to load and unload. However, when the loading time is decreased below one min, the production rates increase sharply. It will also be noted that no appreciable increase in production will be obtained by selecting the high speed and feed, unless the loading time is decreased. For example, at 0.1 min loading time, 102 pieces per hr could be milled at 500 fpm, compared to 80 pieces per hr at

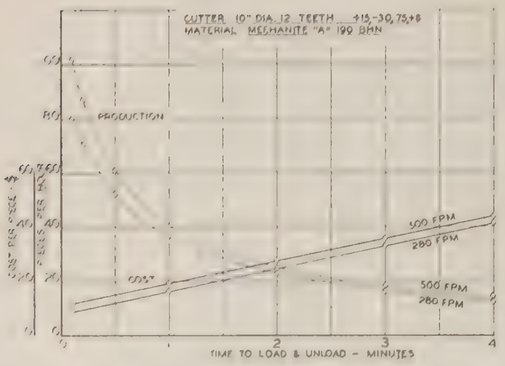


FIG. 18 PIECES PER HOUR AND COST PER PIECE VERSUS TIME TO LOAD AND UNLOAD
(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide, 20 in. long.)

280 fpm. This is an increase of about 27 per cent. In contrast, for a 2-min loading time, 22.5 pieces per hr could be milled with the high speed and feed, compared to 21 pieces per hr with the low speed and feed; an increase of only 7 per cent.

Number of Pieces in Lot. The effect of the number of pieces in the lot being milled is shown in Fig. 19. If the number of pieces in the lot is small, the original setup time may seriously decrease the production rate and may add appreciably to the cost of milling. In the example chosen, it is seen that the cost and production curves change rapidly if there are fewer than 200 pieces in the lot. However, for larger lots the changes in the curves are very small. It is furthermore interesting to note the small difference between 800- and 10,000-piece lots. There is a constant difference in cost per piece between the 500- and 280-fpm curve. The difference in production rates between the two cutting speeds is small for very small lots. However, this difference increases and eventually becomes practically constant above a 300-piece lot.

Length of Cut. The effect of the length of cut on cost and production is shown in Fig. 20. The cost was about the same for both cutting speeds for a 5-in. length of cut. As the length of cut increased, the cost per piece to mill at the higher cutting speed increased over that of the lower cutting speed. However, the production rate at 500 fpm was substantially higher than at 280 fpm for lengths of cuts of 5 to 30 in.

MILLING OF STEEL

Turning now to the milling of steel, we may cite as an example an analysis made of an actual production milling job performed in the authors' plant. The workpiece was a steel forging (S.A.E. 4340, 415 Bhn) used as a "drag link" on the landing gear of the B-29 Superfortress. Two major milling operations were performed on this forging, namely, milling the inside and outside surfaces of the yoke on each end. The inside of the yoke was milled, with the setup shown in Fig. 21, on a Cincinnati horizontal Hydrotel using two staggered-tooth slotting cutters, 14 in. diam, having 8 teeth, tipped with a steel-cutting grade of carbide. The teeth were ground with an axial rake of —20 deg, a radial rake of —20 deg, and $^1/_4$ in. radius. Each cutter took a cut $^5/_{16}$ in. deep and $3^3/_4$ in. wide. The work was fed into the cutter at a rate sufficient to provide a feed per tooth of 0.008 in. The return feed was made at the same rate so as to reduce the dimensional error due to springing of the workpiece, and to avoid marring the work, as a finish under 50 microinches (rms) had to be obtained.

Three cutting speeds were chosen, 235, 290, and 370 fpm, respectively. From Fig. 22 it is seen that minimum cost and maximum production were obtained at 290 fpm. Designating the cost and production rates at this optimum speed 100 per cent, it is seen from the curves that, at a cutting speed of 235 fpm, the cost increased to 104 per cent while the production dropped to 94 per cent, whereas at 370 fpm, the production remained at 100 per cent while the cost rose to 107 per cent. Obviously, a cutting speed in the neighborhood of 300 fpm was the best for this particular operation.

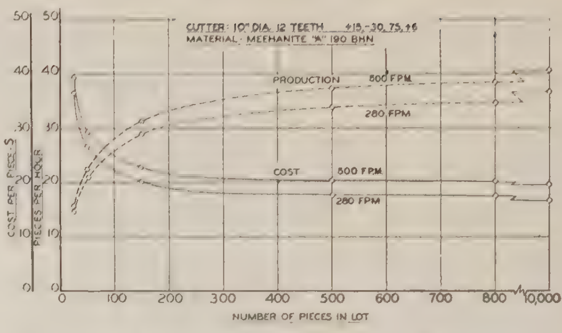


FIG. 19 PIECES PER HOUR AND COST PER PIECE VERSUS NUMBER OF PIECES IN LOT
(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide, 20 in. long.)

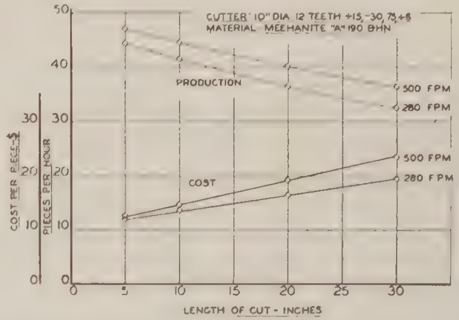


FIG. 20 PIECES PER HOUR AND COST PER PIECE VERSUS LENGTH OF CUT
(Dimensions of cut: $^3/_{16}$ in. deep, 6 in. wide.)

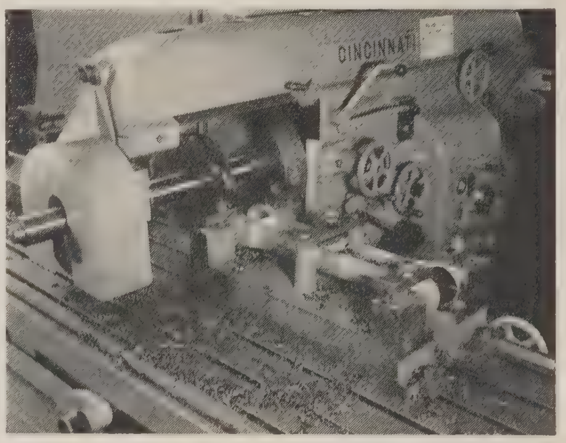


FIG. 21 SETUP FOR MILLING INSIDE OF YOKE ON ALLOY STEEL FORGING
("Drag link" for B-29 Superfortress.)

The second major operation analyzed was the straddle-milling operation on the outsides of the yoke. This operation was performed on a Cincinnati 4-48 duplex Hydromatic miller, with the setup shown in Fig. 23, using a pair of right- and left-hand, 6-in-diam 6-tooth shell-end mills. The cutters had an axial rake of —20 deg, a radial rake of —20 deg, and a corner angle of 30 deg. These cutters also took a cut $^5/_{16}$ in. deep and $3^3/_4$ in. wide. Again, in order to obtain the specified surface finish and accuracy, the cutters were fed fully across each yoke surface. As shown in Fig. 24, the minimum cost per piece and maximum production rate occurred at 285 fpm. Designating this condition as 100 per cent, it is seen that at 210 fpm the cost remained at 100 per cent, while the production dropped to 88 per cent. At 380 fpm, however, the cost jumped to 181 per cent while the production rate dropped to 88 per cent. It is therefore evident that a cutting speed of about 300 fpm was the optimum for this operation.

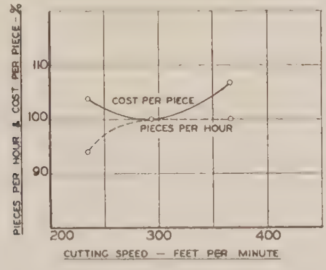


Fig. 22 Production Rate and Cost per Piece Versus Cutting Speed for Milling Inside of Yoke on Drag Link

Cutters: 14 in. diam, 8 teeth, —20 deg axial rake, —20 deg radial rake, $^1/_4$ in. radius. Material: S.A.E. 4340, 415 Bhn. Depth of cut, $^5/_{16}$ in.; width of cut, $3^3/_4$ in.)

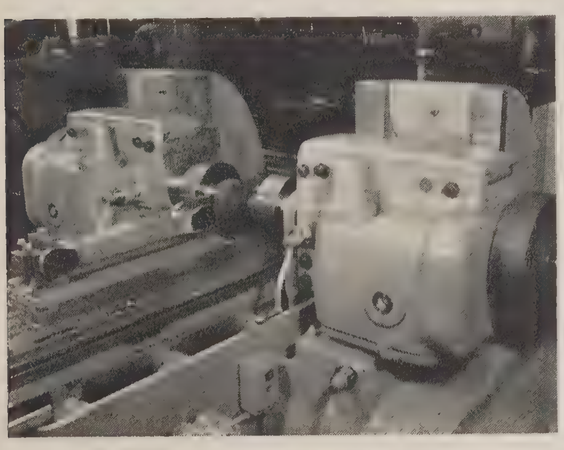


Fig. 23 Setup for Milling Outside of Yoke on Drag Link

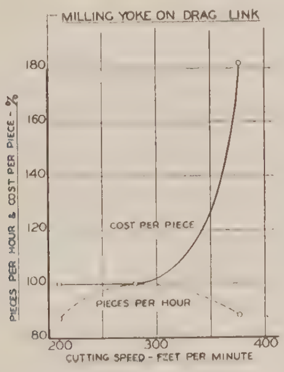


Fig. 24 Production Rate and Cost per Piece Versus Cutting Speed for Milling Outside of Yoke on Drag Link

(Cutters: 6 in. diam., 6 teeth, —20 deg axial and radial rake, 30 deg. corner angle. Material: S.A.E. 4340, 415 Bhn. Depth of cut, $^5/_{16}$-in.; width of cut, $3^3/_4$ in.)

Conclusions

The following conclusions can be drawn from the results of the work done thus far:

1 It is not possible to extrapolate tool-life curves into the low-speed range from the results of tests made at cutting speeds over 500 fpm, for the following reasons:

(*a*) From the results of tests made thus far it appears that the cutting-speed versus tool-life curve may often contain a maximum tool-life point toward the lower end of the speed range. This point seems to exist at a lower value for the harder grades of cast iron.

(*b*) In certain cases the curve of cutting speed versus tool life is not a straight line in the lower speed range when drawn on log-log paper.

2 Practical tests of multitooth cutters with the combination of very large corner angle (70 to 80 deg), large negative radial rake (—30 to —45 deg), and large positive axial rake (+15 to +20 deg), have shown that this combination has unusually long life in terms of volume of metal removed before grinding. It also permits the use of very high feed rates and consequent high production without clogging of chips, and yet meets normal quality specifications of flatness, surface finish, and breakout.

3 Curves showing the relationship between cutting speed, feed, and tool life are essential for an intelligent selection of speed and feed. However, selection of speed and feed for practical milling operations cannot be made directly from these tool-life curves, but must take into consideration also the effect of tool life on both cost and production. Proper balance between these two factors can be made only on the basis of a complete cost analysis.

4 Selection of speed and feed must also be made on the basis of meeting quality specifications, such as flatness, surface finish, and breakout.

5 Cost and rate of production are also affected by number of teeth in the cutter, loading time, length of cut, and number of pieces in the lot. A complete analysis of a milling operation should take into consideration the relationship between these factors and cutting speed and feed.

Acknowledgment

The authors wish to express their appreciation to Dr. M. E. Merchant and Mr. Norman Zlatin of the research department of the Cincinnati Milling Machine Co., for their aid in the preparation of this paper, and to Mr. Walter Deer of the engineering service department, who helped in conducting the shop tests.

(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until May 10, 1946)

Analysis of Initial Contact of Milling Cutter and Work in Relation to Tool Life

By M. KRONENBERG,[1] CINCINNATI, OHIO

Sintered-carbide tools, made of the hardest metal man has ever produced, have played a predominant role in speeding up war production. While appreciable experience has been acquired in connection with the turning of artillery shells and other implements of war, the application of sintered-carbide tools is often more difficult in milling than in turning operations because face-milling cutters, end mills, straddle mills, side mills, etc. are subjected to several hundred impacts per minute, which may destroy the tools prematurely due to their brittleness. Control of impact is therefore vital, a problem not encountered with high-speed steel tools.

It is the purpose of this paper to present methods for determining readily the impact conditions for any shape of the cutting edge as a function of positive and negative rake angles, corner angle, cutter diameter, position of the cutter relative to the workpiece, feed per tooth, and depth of cut. Furthermore, tests are discussed indicating a variation in the wear of sintered-carbide tools with a variation of impact conditions obtained by altering the relative position of cutter and work.

Introduction

While carbide-tipped tools with negative rake angles have been successfully employed for a number of years in intermittent turning operations, this technique is a relatively recent development in the case of milling operations. The reason for this may well lie in the more complex nature of initial contact of milling cutter and work compared with that of a turning tool and work.

In the simple case of machining a slotted flange in a plunge-cut operation, the location of the initial contact can be easily controlled. As shown in Fig. 1, a negative back rake is the only requirement for shifting the initial contact from the sharp point "S" of the tool to a point farther back, where the tool is stronger.

Turning a shaft with a keyway, using a longitudinal feed, is a more difficult case in that three tool angles affect the location of the initial contact when the tool is "on center," namely, back rake, side rake, and side-cutting-edge angle.

In the case of milling operations, a fourth quantity, which is not a property of the tool itself, must be taken into consideration; it depends on the position of the milling cutter relative to the work. This fourth quantity, later referred to as the "angle of engagement," plays a significant part in the location and magnitude of impact.

Three questions present themselves for the analysis of the impact problem. They can conveniently be used for subdividing the subject of this paper as follows:

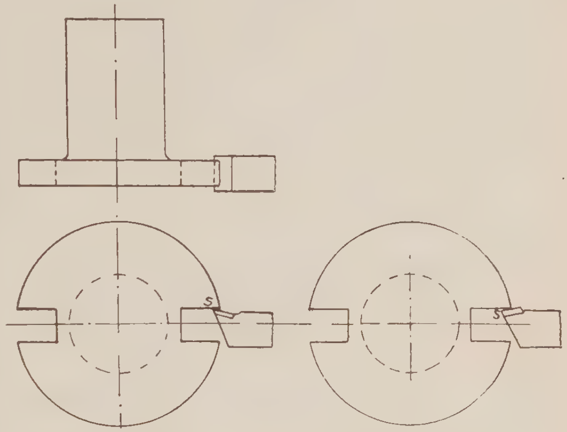


Fig. 1 Initial Contact Produced by Positive and Negative Back Rake, When Turning a Slotted Flange

1 Where does the impact of tooth and work occur? That is, what point on the tool face hits the work first? (Location of impact.)

2 How great is the impact? (Magnitude of impact.)

3 How is tool wear affected by location and magnitude of impact?

1—LOCATION OF IMPACT

Model, Demonstrating Location of Contact

A face-milling cutter is shown in Fig. 2 at an instant shortly before the tooth engages the side wall of the work, termed "plane of engagement." This plane is always assumed parallel to the cutter axis and extending in the direction of feed motion. It is evident from the close-up in the upper corner of Fig. 2 that the material removed by a tooth with a straight cutting edge is indicated by a parallelogram *S-T-U-V*. Its height depends on the depth of cut, while the width is determined by the feed per tooth represented by a dotted line on the top of the work. This parallelogram *S-T-U-V* develops gradually as the tooth enters the work, and the question arises, what spot on the contour of the parallelogram is generated first. This spot will obviously be the point of initial contact.

It will be seen from Fig. 2 that the tooth, before contacting the work, passes through the removed portion of the plane of engagement. This happens when the cutter axis is on the "workside" of the plane of engagement, as shown in Fig. 2.

In the analysis of initial contact, it is necessary to consider only the region of cutter and work indicated in the close-up, Fig. 2. This same region is represented in Fig. 3 as a large-scale model, seen in the direction of the arrow of cutter rotation, Fig. 2.

In Fig. 3 the "tool face" is a glass plate shaped to the contour of a tooth, intersecting the "plane of engagement" in line *L-M*.

[1] The Cincinnati Milling Machine Company.

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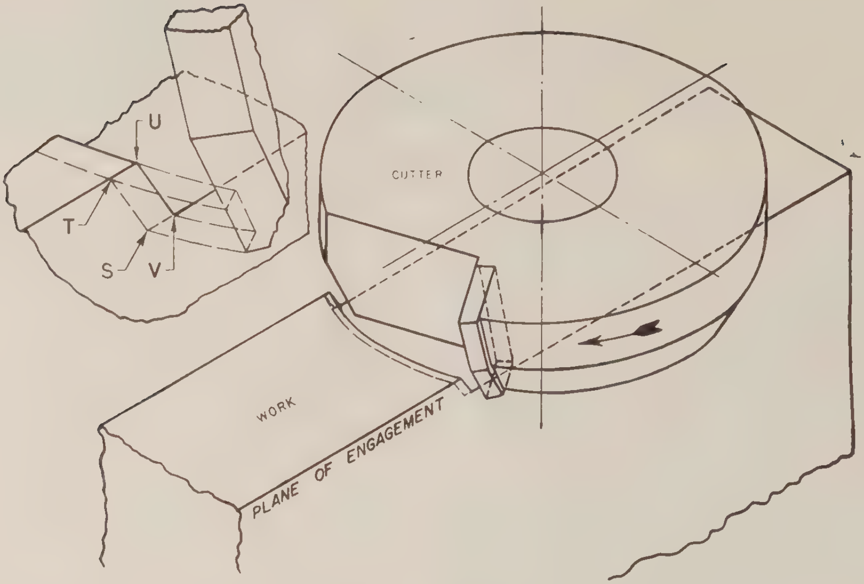


Fig. 2 Diagrammatic View of Cutter and Work, Illustrating an Instant Shortly Before Initial Contact

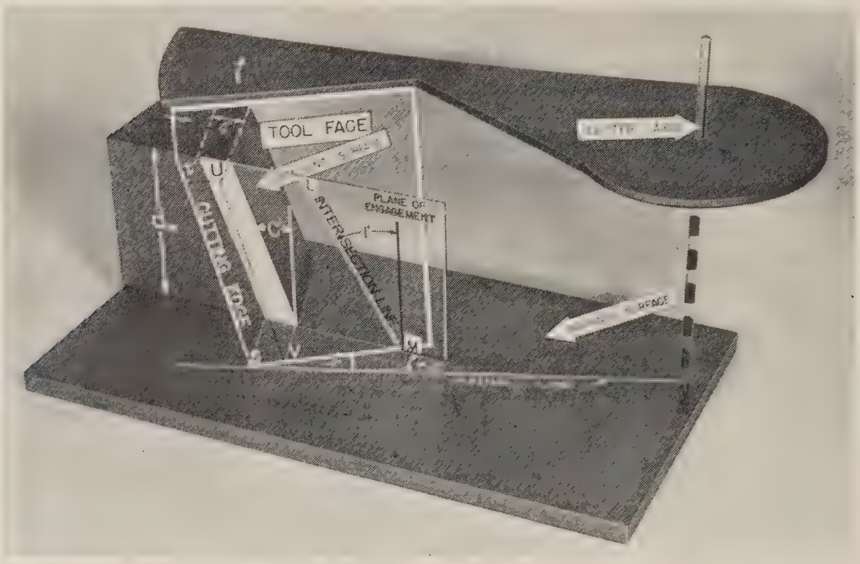


Fig. 3 Model of Cutter and Work Illustrating Intersection Line of Tool Face and Plane of Engagement

The axial rake and the radial rake are both negative and the cutting edge is a straight line in this model.

The direction of cutter rotation is indicated by an arrow pointing into the workpiece.

The "cutter axis" is connected with the tool face by a top piece symbolizing the face of the cutter body. The cutter axis is actually more distant from the tool face than shown in the model.

On the workpiece, dimension d_m is the depth of cut, while dimension f_t represents the feed per tooth.

The chip cross-sectional area is indicated by a white parallelogram on the workpiece. The corresponding points on the tool face are marked S, T, U, and V, in clockwise order.

The distance traveled by the tooth from the instant shown, to the instant of contact, is very short and it may therefore be assumed that the tool moves in a straight line during this very short interval of time, instead of traveling in a circular path. In other words, the tool face is assumed to advance parallel to itself in the direction of the white dotted lines which connect the points, S, T, U, V, on the tool face with the white parallelogram. The white lines are continued on the top of the work because the cutter movement is also considered straight for a short distance within the workpiece.

Point S is the sharp point of the tool and its weakest spot. Initial contact there is undesirable.

Point T is the top point of the active cutting edge and is located at a height above the machined surface equal to the depth of cut (d_m).

Point U indicates a point on the tool face, at the same height as point T, but remote from the cutting edge by approximately the feed per tooth.

Point V is a point on the face edge, located at a distance from point S approximately equal to the feed per tooth.

It is evident from Fig. 3 that in the case of this model, point U on the tool face will contact the work first, because this point is nearest to the white parallelogram. This can be judged by comparing the length of the four dotted white lines connecting points, S, T, U, and V with the work. Point U has the shortest distance. This type of contact is called "U-contact."

If the tool face would be inclined at a positive axial rake, point V would have the shortest distance on the model and contact first (V contact).

It shall now be assumed that the tool face in Fig. 3 is advancing parallel to itself, in the direction of the white dotted lines toward the workpiece. Point U on the tool face will then travel toward the upper right corner of the white parallelogram; simultaneously point L of the intersection line will likewise move toward the upper right corner on the workpiece. At the instant when point U contacts the white parallelogram at its upper right corner, point L of the intersection line must have arrived at that same point also.

If the tool face would be inclined at a positive axial rake and the angle i' would be to the right of the vertical line in the plane of engagement, it would be points V and M which would arrive at the same instant but at the lower right corner of the white parallelogram.

Consequently, the spot on the tool face which contacts the work first is indicated by the point of the intersection line LM, which first contacts the white parallelogram.

This is an important finding, because it furnishes not only the basis for the derivation of mathematical formulas for contact conditions in the case of straight cutting edges but also principles for a graphical solution of the more involved problems connected with nonstraight cutting edges and with the magnitude of impact.

Mathematical Conditions for Location of Initial Contact

Mathematical formulas are required for preparing diagrams and for other graphical methods which can directly be employed by production and tool engineers in their daily work.

The following nomenclature will be used in connection with the derivation of formulas:

a = axial rake
A' = distance of cutter axis from plane of engagement (see Fig. 4)
c = corner angle of tool
c' = slope of transient surface produced by corner angle c ($\tan c' = \tan c/\cos \epsilon$)
c_a = chamfer angle of tool
d_m = depth of cut
D = cutter diameter
ϵ = angle of engagement
f_t = feed per tooth
i = index angle
i' = intersection angle of tool face and plane of engagement ($\tan i' = \tan i/\cos \epsilon$)
I = impact factor (rate of increase of chip cross-sectional area) sq in. per sec
L_d = distance traveled by index line in direction of depth of cut during penetration time
L_f = distance traveled by index line in the direction of feed during penetration time
r = radial rake
T_p = penetration time, i.e., time elapsing between instant of initial contact and instant when both cutting edge and face edge of tool have engaged the work
T_e = exit time, i.e., time elapsing between instant when first point of tool leaves work and instant when both cutting edge and face edge are disengaged

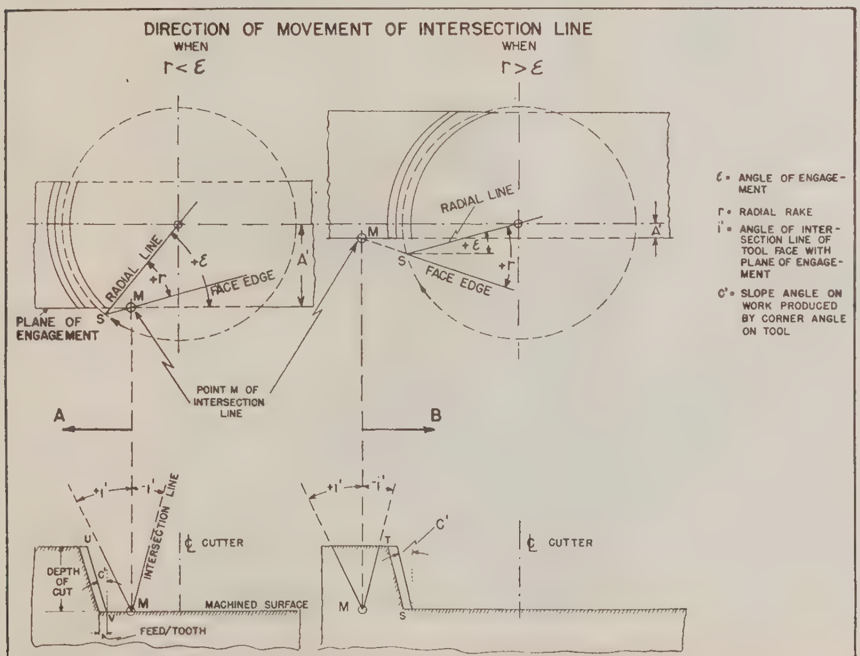


Fig. 4 Direction of Movement of Intersection Line

Referring to Fig. 3, two important angles, c' and i', will be seen in the plane of engagement.

Angle c' is the slope of the "transient surface" measured in the plane of engagement. This angle has been produced by the preceding tooth and depends therefore on the shape of the cutting edge and on the angle of engagement, as will become evident later.

Angle i' is an angle formed by line LM (which is the intersection line of tool face and plane of engagement) and a line parallel to the cutter axis. Angle i' is called "intersection angle."

The model, Fig. 3, refers only to the case where the intersection line is outwardly inclined with respect to the cutter axis and where the cutter axis is on the workside of the plane of engagement.

For a more general consideration, reference is made to Fig. 4. It will be seen from the lower sketches that the intersection angle i' can be positive or negative with respect to the vertical line parallel to the cutter axis, that is, it can be outwardly or inwardly inclined. Furthermore, the radial rake, formed by the face edge and a radial line, is shown in Fig. 4, marked r. In addition, the angle formed by the plane of engagement and the radial line is indicated; it is the angle of engagement (ϵ).

For determination of the direction of movement of the intersection line, it is sufficient to consider only its lower point M which is also a point of the face edge of the tool. In the case of the left-hand sketch in Fig. 4, point M moves toward the left as the cutter continues to rotate. Hence the intersection line will always move "away" from the cutter axis, due to cutter rotation, when $r < \epsilon$. Consequently, in cases where $r < \epsilon$, initial contact can occur only at the right-hand contour of the chip cross-sectional area, that is, in the case of a straight cutting edge, either at point U, or along line U-V or at point V, depending on the relationship of the two angles c' and i'.

Changing the setup (right-hand sketch in Fig. 4), by moving the workpiece into a different position with respect to the cutter axis, will change the relationship between the angles r and ϵ.

Point M of the intersection line is now located on the other side and will move "toward" the cutter axis, as the cutter continues to rotate. Consequently, in cases where $r > \epsilon$, initial contact can occur only at the left-hand contour of the chip cross-sectional area, that is, in the case of a straight cutting edge, either at point S, or T, or along line ST.

The following mathematical relationships for initial contact of straight cutting edges can therefore be derived from inspection of Fig. 4

S = contact occurs when $r > \epsilon$ and $i' > c'$........[1a]

S-T (line) = contact occurs when $r > \epsilon$ and $i' = c'$........[1b]

T = contact occurs when $r > \epsilon$ and $i' < c'$........[1c]

U = contact occurs when $r < \epsilon$ and $i' > c'$........[1d]

U-V (line) = contact occurs when $r < \epsilon$ and $i' = c'$........[1e]

V = contact occurs when $r < \epsilon$ and $i' < c'$........[1f]

There are three more types of initial contact possible in the case of straight cutting edges, namely, when $r = \epsilon$. In these special cases, the intersection line and therefore point M is at infinity because the face edge enters the work parallel to the plane of engagement. The mathematical conditions for these cases will be discussed later.

In order to express the contact conditions as functions of axial rake, radial rake, angle of engagement, it is now necessary to find the relationship between these three angles and the intersection angle i'.

Referring to Fig. 5: Dropping a perpendicular line from point L of the intersection line L-M, produces a vertical triangle L-N-M

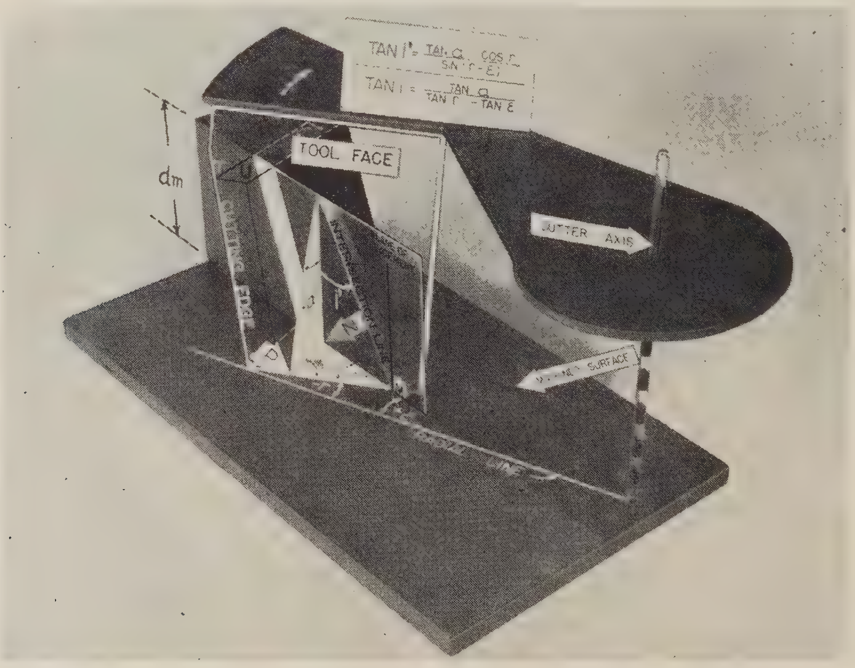
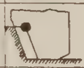
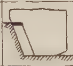


Fig. 5 Model of Cutter and Work Illustrating Relationship Between Axial Rake, Radial Rake, Angle of Engagement, and Intersection Angle

TABLE 1 MATHEMATICAL CONDITIONS FOR THE NINE TYPES OF INITIAL CONTACT FOR STRAIGHT CUTTING EDGES

	TYPE OF CONTACT	Relationship: radial rake (r) & angle of engagement (ϵ)	Relationship: corner angle (c) and index angle (i)	Relationship corner angle (c) axial rake (a) radial rake (r) and angle of engagement (ϵ)
1	 S	$r > \epsilon$	$c < i$	$\tan c < \frac{\tan a}{\tan r - \tan \epsilon}$
2	 T	"	$c > i$	$\tan c > \frac{\tan a}{\tan r - \tan \epsilon}$
3	 U	$r < \epsilon$	$c < i$	$\tan c < \frac{\tan a}{\tan r - \tan \epsilon}$
4	 V	"	$c > i$	$\tan c > \frac{\tan a}{\tan r - \tan \epsilon}$
5	 ST	$r > \epsilon$	$c = i$	$\tan c = \frac{\tan a}{\tan r - \tan \epsilon}$
6	 UV	$r < \epsilon$	$c = i$	$\tan c = \frac{\tan a}{\tan r - \tan \epsilon}$
7	 VS	$r = \epsilon$	$i = 90°$	$a = +$
8	 UT	"	"	$a = -$
9	 FACE	"	NO LINE INTERSECTION	$a = 0$

in the plane of engagement. This triangle L-N-M contains angle i'. The base line N-M equals

$$d_m \cdot \tan i'$$

The axial rake is measured, according to definition, in a plane located in the direction of cutter rotation, that is, in a plane perpendicular to the radial line. Such a plane is represented in Fig. 5 by the white vertical triangle L-N-P, containing the axial rake ($-a$) which is shown negative. The base line N-P equals

$$d_m \cdot (-\tan a)$$

The two base lines N-M and N-P, together with portion P-M of the face edge, form the horizontal, white triangle P-N-M, with the angles

$$(90 + r)$$

and

$$(\epsilon - r)$$

Applying the sine law to the horizontal triangle results in the formula

$$\tan i' = \frac{\tan a \cdot \cos r}{\sin (r - \epsilon)} \quad \ldots\ldots\ldots\ldots [2]$$

Equation [2], however, is inconvenient for use in a diagram, because the radial rake r appears twice.

A simplification can be obtained by multiplying both sides by $\cos \epsilon$. This transformation gives

$$\boxed{\tan i = \frac{\tan a}{\tan r - \tan \epsilon}} \quad \ldots\ldots\ldots\ldots [3]$$

It will be noticed that Equation [3] refers to an angle i instead of angle i'. This is due to the multiplication of Equation [2] by $\cos \epsilon$. Equation [3] represents therefore a different angle, called "index angle" (i).

The relationship between the intersection angle (i') and the index angle (i) follows from

$$\tan i = \tan i' \cos \epsilon \ldots\ldots\ldots\ldots [4]$$

Index angle i is a projection of intersection angle i', on a vertical plane which can be erected in the radial line. This plane, called reference plane, forms the angle ϵ with the plane of engagement, but is not shown in the model.

In order to maintain the mathematical relationship between slope angle c' and intersection angle i' when determining the location of initial contact (see formulas [1a] to [1f], inclusive), it is necessary to multiply the slope angle c' also by $\cos \epsilon$ as was done with angle i'. The slope angle c' is thus substituted by the corner angle c (or bevel angle) as ground to the tool itself, which is measured in the reference plane. The corner angle c is mathematically a projection of the slope angle c' on the reference plane. The relationship between slope angle c' and corner angle c is given by

$$\tan c = \tan c' \cdot \cos \epsilon \ldots\ldots\ldots\ldots [5]$$

The multiplication of Equation [2] by $\cos \epsilon$ does not only simplify the formula, but allows using the corner angle c of the tool instead of the slope angle c' on the work.

The following general equation is obtained for initial contact in the case of straight cutting edges by substituting Equations [3] and [5] into formulas [1a] to [1f]

$$\tan c \overset{>}{\underset{<}{=}} \frac{\tan a}{\tan r - \tan \epsilon} \ldots\ldots\ldots\ldots [6]$$

The individual formulas for the nine possible contacts in case of straight cutting edges are summarized in Table 1. The last three cases, where $r = \epsilon$, follow if Equation [6] is solved for $\tan a$

$$\tan a \overset{>}{\underset{<}{=}} \tan c (\tan r - \tan \epsilon) \ldots\ldots\ldots\ldots [6a]$$

The right side becomes zero if $r = \epsilon$, indicating that these three contact conditions are determined by the fact as to whether the axial rake "a" is positive, zero, or negative.

Cutter-Engagement Diagram

For practical use in engineering offices a "cutter-engagement diagram," Fig. 6, has been prepared, based on the foregoing mathematical analysis. It refers primarily to cases where the cutting edge is a straight line and where only a determination of the location of initial contact is desired.

A more general method is described later on in this paper which includes the determination of the progress of engagement and of the magnitude of impact, in addition to the location of impact.

Four fields will be seen in Fig. 6, marked in counterclockwise order; S-contact, T-contact, U-contact, and V-contact. Three smaller rectangles flanked by these four fields refer to three types of line contacts as indicated on the diagram. The two remaining types of contact, S-T line-contact and U-V line-contact are determined by the border lines between the S-field and T-field, and the U-field and V-field, respectively.

The type of contact can easily be determined by tracing along horizontal and vertical straight lines. This is indicated by a dot and dash line marked "example."

The example refers to a case where, at a distance of 2 in., a cutter with a diameter of 12 in. is located on the workside, which is the usual condition with face mills, end mills, etc. The radial rake and the axial rake are assumed 10 deg negative. Following

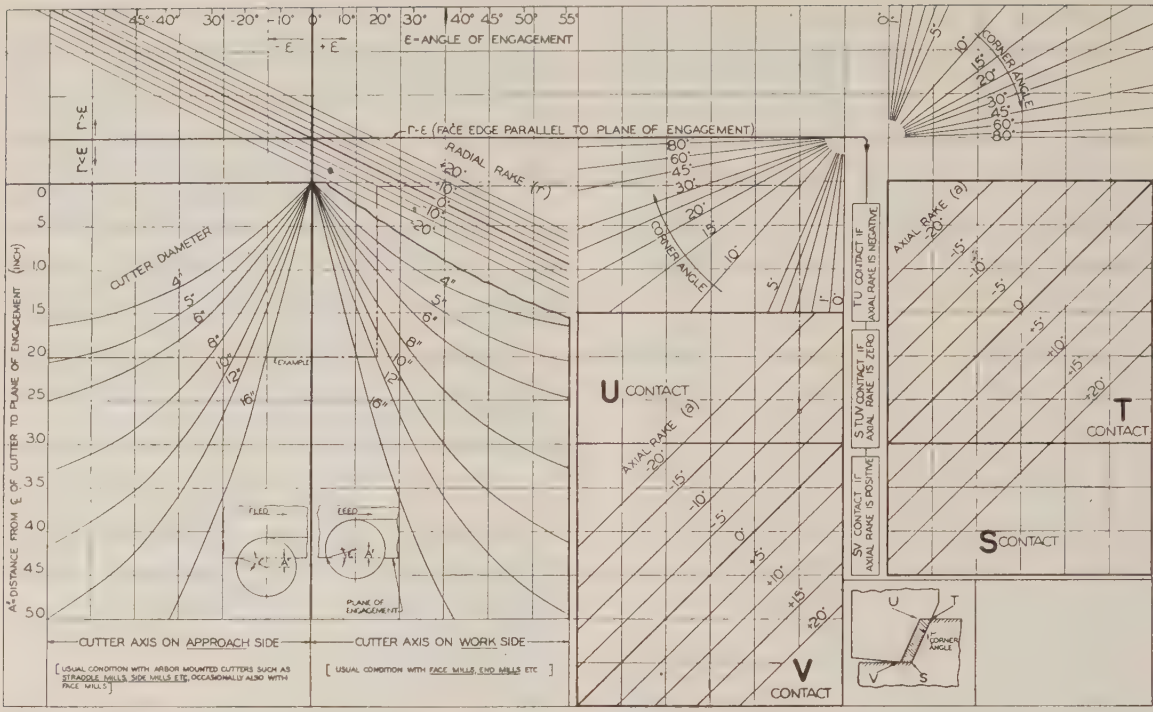


Fig. 6 Cutter-Engagement Diagram

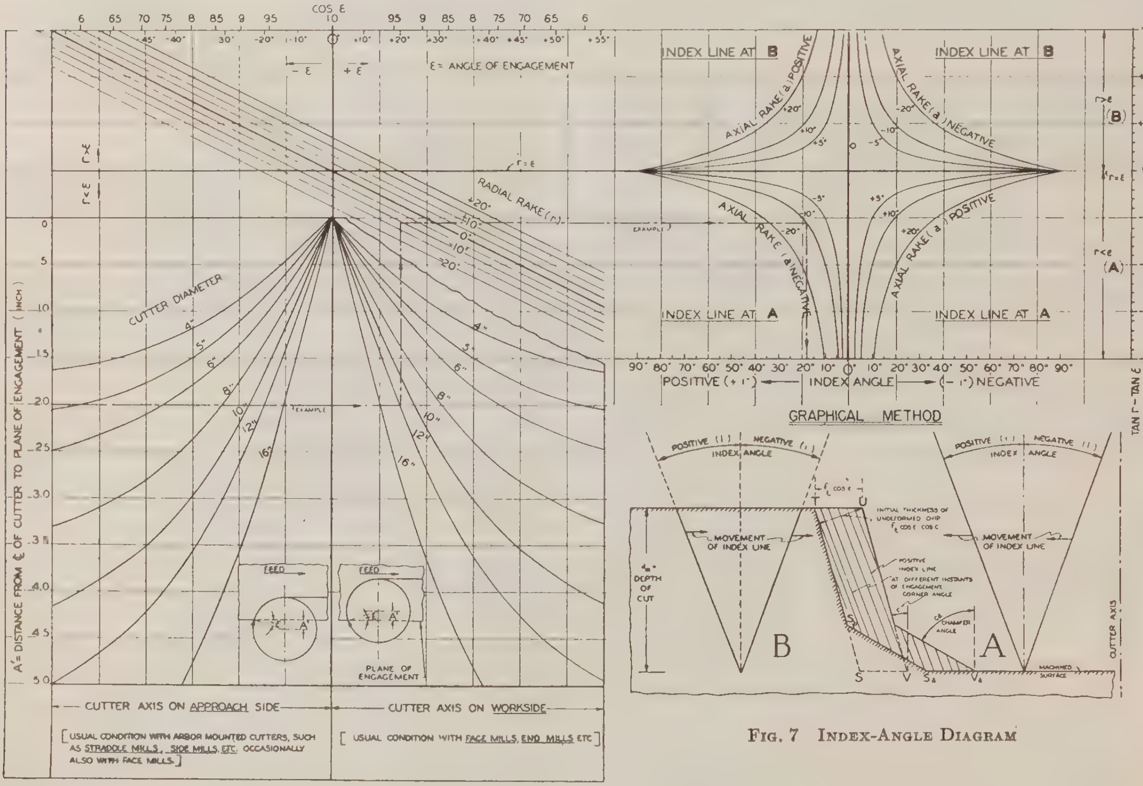


Fig. 7 Index-Angle Diagram

the dot-and-dash line leads to the U-field. Hence U-contact will ensue from this combination of rake angles, corner angle, cutter distance A', and cutter diameter.

The angle of engagement can be read at the top of the cutter-engagement diagram ($\epsilon = 20$ deg); it is however, not necessary to read this angle when using the diagram.

Graphical Solution of Contact Problems for any Shape of Cutting Edge

Complex formulas for the contact conditions result if the cutting edge is chamfered or rounded. However, a graphical method which gives, also, a mental picture of the engagement of cutter and work has been developed for solving such cases on the drawing board. The graphical method covers a broad field in that it includes the progress of engagement and disengagement, and also the magnitude of impact in addition to solving problems of initial contact for any shape of the cutting edge. Exit conditions existing when the cutter leaves the work can likewise be solved with the graphical method.

It has already been stated, in connection with Fig. 3, that the spot on the tool which contacts the work first is indicated by the point where the intersection line first contacts the white parallelogram.

Thus in order to determine graphically the location of initial contact, it is only necessary to draw the chip cross-sectional area (white parallelogram) to scale and to move a line at angle i' toward it. Where this line touches the chip cross-sectional area first, initial contact occurs.

It is, however, more convenient to move a line at angle i (index line) instead of the intersection line at angle i', because the corner angle c can then be used when drawing the chip cross-sectional area instead of the slope angle c'.

Furthermore, referring to Equation [3], it will be seen that the index angle i is independent of the corner angle c and therefore independent of the contour of the cutting edge.

Hence initial contact can be determined graphically by moving a straight line at index angle i toward a chip cross-sectional area of any shape, not only a parallelogram.

The index angle i, which is the compound angle of axial rake, radial rake, and angle of engagement, can either be calculated from Equation [3] or be determined without use of mathematics by means of the "index-angle diagram," Fig. 7. The left-hand portion of this diagram is identical with the left-hand portion of the "cutter-engagement diagram," Fig. 6, while the right-hand part of Fig. 7 is different, having curves for the axial rake, instead of the contact fields. The example in Fig. 7 corresponds to that on the cutter-engagement diagram. Tracing along the dash-dotted example line, however, leads, in Fig. 7, to an index angle of +18 deg and to the instruction that the index line must be placed at A, meaning that it moves away from the cutter axis (because $r < \epsilon$). Compare arrow "A" in Fig. 4.

The graphical solution (see Fig. 8) includes the following:

1 Construct a figure proportional to the initial chip cross section, indicated by the cross-hatched area, where d_m is proportional to the depth of cut, distances TU, SV, S_aV_a proportional to the feed per tooth $\cdot \cos \epsilon$ and where c = corner angle, c_a = chamfer angle.

2 Draw index line at index angle i at A (or B) of cross-hatched area, as determined from diagram. [A is indicated if $r < \epsilon$, and B if $r > \epsilon$.]

3 Move index line toward cross-hatched area. (If $r = \epsilon$ the index line is moved vertically; upward when the axial rake is positive, or downward when the axial rake is negative. No index line exists for face contact, i.e., when $r = \epsilon$ and the axial rake is zero.)

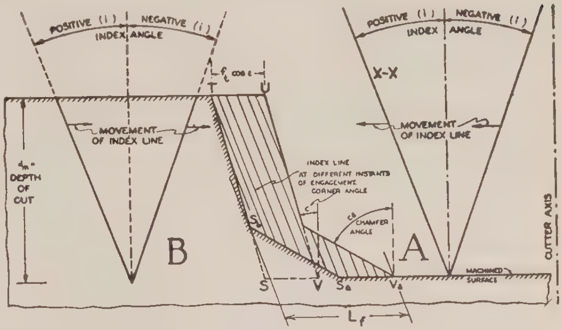


Fig. 8 Graphical Method for Determining Initial Contact of Cutter and Work, and Also the Progress of Their Engagement for Any Shape of the Cutting Edge

4 *Initial contact of cutter and work is indicated by the point or line where the moving index line touches the contour of the cross-hatched area first.*

In the given example when moving an index line x-x at +18 deg the initial contact occurs at point V_a.

5 Progress of engagement: The graphical method, however, permits not only determination of initial contact, but also the progress of engagement. Continuing the movement of the index line, x-x it, will be seen that

2nd point to engage is S_a
3rd point to engage is U
4th point to engage is T
5th point to engage is S_b

A closer inspection, by slowly moving the index line across the cross-hatched area, reveals even more details about the progress of engagement, for example, the fact that the engagement of point U follows very closely the engagement of point S_a. It will also be seen that a chip develops at first by deforming a triangle with the base V_aS_a, while later on a second chip develops from point U. Both chips unite at the instant when the vertex of the obtuse angle (i.e., the point opposite S_b) engages.

If the cutting edge is straight, the chip cross-sectional area is represented by parallelogram S-T-U-V in Fig. 8. Initial contact occurs in the case of $i = +18$ deg at point U instead of at V_a as was the case in the foregoing example for a chamfered cutting edge.

In the case of this example, chamfering of the cutting edge changes the initial contact from U to V_a. Chamfering, however, does not always have this effect. If the conditions for radial rake, axial rake, and angle of engagement are such as to give a negative angle i, initial contact of a tool having a straight cutting edge would occur at V, not at U, and the change due to chamfering would then be only a shifting of the initial contact from V to V_a.

Exit conditions when the cutter leaves the work can likewise be determined by the graphical method. Only one modification is necessary. Instead of using the distance of the cutter axis from the plane of engagement, the distance of the cutter axis from the exit plane must be used, when determining the index angle i in diagram Fig. 7. The terms "cutter axis on approach side" and "cutter axis on workside" are then to be used with reference to the exit plane.

2 Magnitude of Impact

The second problem, namely, the question, "How great is the impact?" is important because it may well be that the magnitude

of impact is a vital factor in cutter life. The magnitude of impact can be determined by further expanding the graphical method, used previously for determining the location and progress of engagement.

From the graphical method, Fig. 8, it will be realized that a chip does not exist at the instant of contact; that is, at the instant when the index line touches the contour of the cross-hatched area. However, as the index line travels across the cross-hatched area, the chip develops gradually, reaching its full size when the index line has moved all the way across the cross-hatched area. At that instant, all points of the tool face that take part in the metal removal have engaged the work.

For instance, in the example where $i = +18$ deg, no chip exists when the index line touches point V_a, but the full chip has been generated when the index line has arrived at point S_b of the cross-hatched area.

Conclusion: During the period of time in which the index line travels across the chip cross-sectional area, the chip has developed from zero to full size. This time interval is called "penetration time."

The shorter the penetration time, i.e., the faster the chip develops, the larger the impact.

The rate of increase of chip cross-sectional area can therefore be used as an "impact factor."

$$\text{Impact factor} = \frac{\text{Chip cross-sectional area}}{\text{Penetration time}} \quad \ldots\ldots\ldots [7]$$

In order to find the impact factor, it is necessary to determine only the penetration time and to divide it into the chip cross-sectional area. The impact factor has the dimension of square inch per sec.

Since "time" is the ratio of "distance traveled" to "velocity," it is possible to determine the penetration time T_p by dividing the distance L_f, which the index line has traveled across the cross-hatched area, by the velocity of this travel v_f.

As shown in Fig. 8, distance L_f can be measured directly from the graph between the contact position and end position of the index line when the graph is drawn to scale.

It remains to find the velocity of travel of the index line.

The velocity of travel v_f can be determined from the model shown in Fig. 9. In the horizontal triangle N-P-M, dimension P-N lies in the direction of cutter rotation and can therefore be taken as the "cutting speed vector." Dimension M-N lies in the direction of travel of the intersection line L-M, and represents therefore the "velocity vector" of the intersection line.

From the geometry of the horizontal triangle N-P-M

$$v'_f = \frac{\tan i'}{-\tan a} v_c \quad \ldots\ldots\ldots [8]$$

The same conversion as before is necessary in order to find the velocity of the index line instead of the intersection line. The result is

$$v_f = \frac{-v_c}{\tan r - \tan \epsilon} \quad \ldots\ldots\ldots [9]$$

It will be seen that the travel velocity v_f of the index line is a function of the cutting speed v_c, the radial rake, and the angle of engagement.

The minus sign in Equations [8] and [9] indicates the direction of travel, i.e., v_f is counted positive when the index line moves away from the cutter axis.

The penetration time in seconds is therefore

$$\boxed{T_p = \frac{5L_f}{v_f} = \pm \frac{5\,L_f\,(\tan r - \tan \epsilon)}{v_c}} \text{ sec} \quad \ldots\ldots [10]$$

where the factor 5 is due to conversion from feet per minute to inches per second.

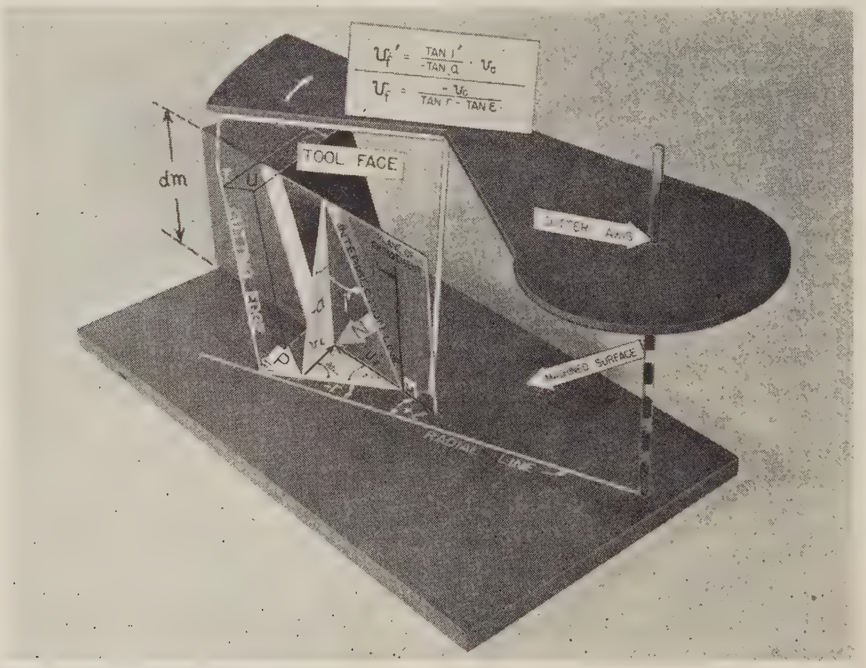


Fig. 9 Model of Cutter and Work Illustrating Relationship Between Cutting Speed and Travel Velocity of Intersection Line

When dimension L_f is very large, it is more convenient to use dimension L_d, Fig. 10, and the velocity (v_d) of the apparent travel in this direction.

Since this velocity, v_d is perpendicular to velocity v_f, their relationship is expressed by

$$v_d = \frac{v_f}{\tan i} = \frac{v_f\,(\tan r - \tan \epsilon)}{\tan a} \quad \ldots\ldots\ldots\ldots [11]$$

Upon substitution of Equation [9], the following formula results for the penetration time

$$\boxed{T_p = \pm\frac{5L_d \cdot \tan a}{v_c}} \quad \ldots [12]$$

When comparing the magnitude of impact for various tool-angle combinations, it is often *not* necessary to determine penetration time and impact factor from Equations [10], [12], and [7], but it is sufficient to compare only the dimensions L_f (or L_d), obtained graphically as the travel distance of the index line across the chip cross-sectional area from contact position to end position.

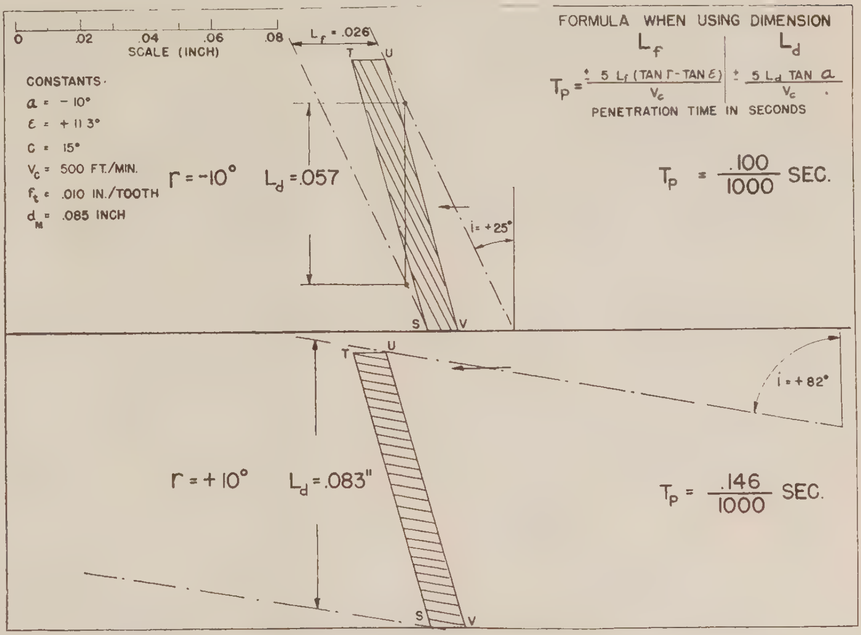


Fig. 10 Determination of Penetration Time and Magnitude of Impact

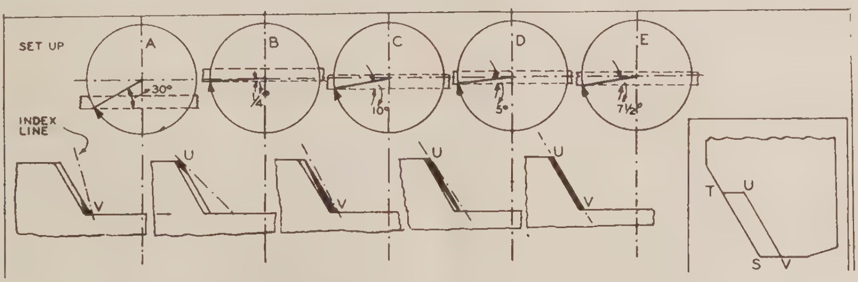


Fig. 11 Setup of Cutter and Work Used in Tool-Wear Tests

Setup	Distance of cutter axis from plane of engagement, in.	Angle of engagement, deg	Location of impact	Magnitude of impact (impact factor) sq in. per sec
A	2.000	30	V	3.4
B	0.020	1/4	U	12.3
C	0.700	10	V	19.5
D	0.360	5	U	24.5
E	0.520	7 1/2	UV	44.5

Constants:

Axial and radial rake, deg	10 negative
Corner angle, deg	30
Feed per tooth, in	0.009
Depth of cut, in	1/8
Cutting speed, fpm	490

The following rules apply:

(*a*) The longer distance L_f, the less the impact, provided that the axial rake is the only variable involved in a comparison of impact.

(*b*) The longer distance L_d, the less the impact, provided that the radial rake or the angle of engagement is the variable involved in the comparison.

(*c*) The longer either distance L_f or L_d, the less the impact, provided that the corner angle c is the only variable involved in the comparison of impact.

The two examples shown in Fig. 10 indicate that the impact will be smaller in the lower case, because L_d is here longer than in the upper case. The L_d comparison is used because the radial rake is the variable in the examples. The ratio of impact is inversely proportional to the distance L_d, i.e., the impact is 46 per cent greater in the upper case than in the lower one.

3 HOW IS TOOL WEAR AFFECTED BY IMPACT?

Tool-life tests have been carried out in order to determine the relationship between impact and tool wear.

A single-tooth cutter was employed for each run, changing the tooth after 13,300 impacts. Tests were also made with 20,000 impacts, and 26,600 impacts.

The location and magnitude of impact were varied by changing the distance of the cutter axis from the plane of engagement, that is, by changing the angle of engagement (ϵ). It was necessary only to adjust the knee of a horizontal milling machine up or down for each setup. Five different setups were tested as indicated in Fig. 11.

It will be noticed that the variation in the angle of engagement (ϵ) was small; this was done in order to keep the initial chip thickness and the exit conditions of the cutter as constant as possible.

The width of the workpiece was only $^3/_4$ in. in order to reduce tool wear due to cutting, in comparison with the wear due to impact.

The average depth of the crater on the tool face, measured

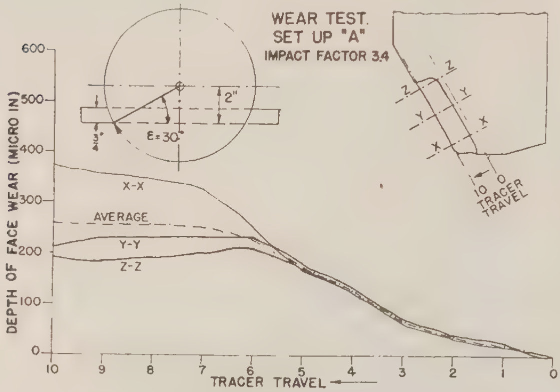


Fig. 12 Wear Test, Setup A; Smallest Impact Factor

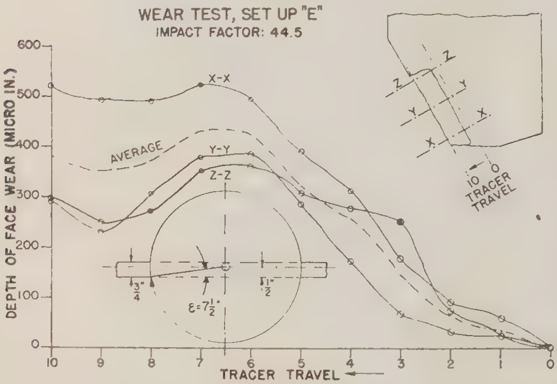


Fig. 13 Wear Test, Setup E; Largest Impact Factor

after the same number of impacts, was taken as the basis for comparison of tool wear. The results are indicated in Figs. 12 to 16, inclusive.

Fig. 12 shows the results of the wear test for setup A where the distance from cutter axis to plane of engagement was 2 in., corresponding to an angle of engagement of 30 deg.

The wear was measured at ten points in each of the three planes

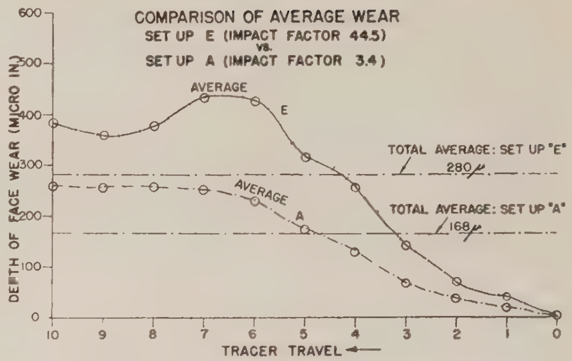


Fig. 14 Comparison of Average Wear; Setup E Versus Setup A

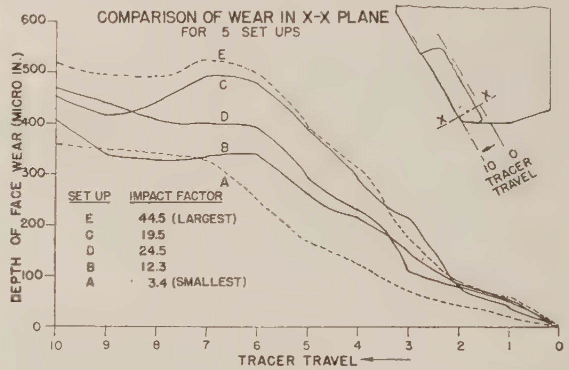


Fig. 15 Comparison of Wear in X-X Plane for Five Setups

marked X-X, Y-Y, and Z-Z, perpendicular to the cutting edge. The wear at each point was measured repeatedly.

It will be seen that the wear in the X-X plane was the greatest, with a maximum at tracer travel point 10 (that is, on the cutting edge). Tracer point 0 indicates the surface on the tool face not touched by the chip.

The dash-dotted line indicates the average of the wear in the three planes X-X, Y-Y, and Z-Z.

Setup A is the case where the impact factor is the smallest of the five setups tested.

Fig. 13 refers to the wear in the case of the largest impact factor (setup E), again measured in three planes perpendicular to the cutting edge. The distance of the cutter axis from the plane of engagement was $^1/_2$ in., the angle ϵ was $7^1/_2$ deg.

The averaged value of the three curves is again indicated by a curve marked "average."

Setup E and setup A are compared in Fig. 14, where the average curves of these two setups only are plotted. It will be seen that the wear in the case of the larger impact (setup E) was always greater than the wear of the smallest impact (setup A). Totaling the wear indicated by each one of the average curves results in a figure called "total average." This figure represents the over-all wear for each setup and is the result of about 200 values.

In the case of setup A, the total average was 168 microinches but as much as 280 microinches in the case of setup E. This appreciable difference of nearly 70 per cent is caused by the slight difference in the relative position of cutter axis to workpiece, and its resultant change in location and magnitude of impact.

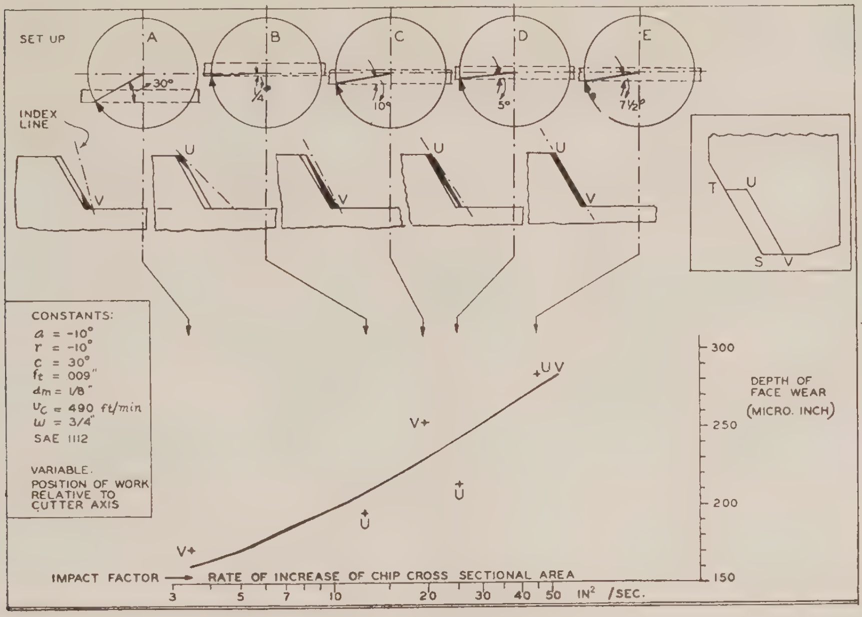


Fig. 16 Summary of Wear Tests

Diagram, Fig. 15, has been prepared in order to compare the wear for all five setups. Wear values in the X-X plane have been selected as an example, but the same trend holds also for the other planes.

It will be seen that the wear curves follow each other in approximately the same order as the impact factors.

Reading from the bottom up, the smallest wear (curve A) occurred in the case of the smallest impact; the second curve (curve B), referring to a larger impact factor, indicates also greater wear, and the greatest wear is found in the case of the largest impact factor.

Only curves C and D are reversed, because D has a smaller wear but a higher impact factor than curve C.

An explanation for this reversal is offered in Fig. 16 which is a summary of the wear tests. The impact factor is plotted logarithmically on the x-axis; the depth of the face wear on the y-axis, which is arithmetical.

Each of the five points is the result of approximately 200 values and represents the total average of the wear for the respective setup, indicated diagrammatically at the top. A definite trend is indicated by the curve, namely, the finding that tool-face wear increases with increasing impact factor, that is, with the rate of increase of chip cross-sectional area. It will furthermore be noticed that the points indicating the wear in case of U-contact are below the trend curve, while those for V-contact are above the curve.

It may well be the case that, within limits, a large impact applied at point U causes less wear than a small impact at point V. This may explain the reversal of the C and D curves in Fig. 15 where the D curve represents the case of a large impact applied at a favorable point of the tool (U-contact), while the C curve refers to a small impact applied at the less favorable point V. This would also indicate that equal impacts will cause less wear when applied at U than at V.

In so far as our tests indicate up to now, it is concluded that a small impact at point U is the most desirable condition, but a large impact at this same point (U) is worse than a small impact at V.

It appears that both the location and the magnitude of impact should be taken into consideration for determining favorable tool-life conditions. Their combined effect is a major factor in tool wear.

There are naturally other factors affecting tool wear; if these other factors cannot be kept constant when varying the location and magnitude of impact, they may change the trend indicated by the curve. The curve may run more horizontally or rise more rapidly. This will be the subject of further research.

Conclusions

With the aid of the graphical method, the following conclusions can readily be drawn from exploring contact conditions of various combinations of axial rake, radial rake, cutter diameter, distance of cutter from work, and for various shapes of the cutting edge:

1 For face mills and end mills, usually located on the work-side of the plane of engagement:

(*a*) The most frequent type is V-contact, that is, at a point on the face edge, remote from the sharp point of the tool by a distance approximately equal to the feed per tooth.

(*b*) U-contact can occur only when the axial rake is negative; the radial rake, however, may be either positive or negative, but must be smaller than the angle of engagement. This latter condition ($r < \epsilon$) is usually satisfied with face mills and end mills, except when the cutter axis is nearer to the plane of engagement than about 20 per cent of the cutter diameter.

(*c*) The larger the corner angle, the less likely is U-contact.

(*d*) Chamfering of a cutting edge renders it more difficult to obtain U-contact.

(*e*) Changing the corner angle c does not always affect initial contact. There is no effect if the axial rake is positive. There may be, however, an effect if the axial rake is negative. (If c is small, initial contact will be at U, if c is large, initial contact will be at V.)

2 For arbor-mounted cutters, such as straddle mills, side mills (and occasionally also for face mills, when only a portion of the width of the workpiece is machined):

(*a*) The most frequent type is *T*-contact, that is, contact at the top point of the cutting edge.

(*b*) The undesirable type of *S*-contact, at the weakest spot of the tool, can be eliminated by grinding a negative axial rake to the tooth.

(*c*) A negative radial rake alone does not necessarily eliminate the undesirable type of *S*-contact.

(*d*) Chamfering of a straight cutting edge having *T*-contact will not change this type of contact and is therefore ineffective as far as contact conditions are concerned.

(*e*) Chamfering of a straight cutting edge, however, having *S*-contact, will eliminate such contact by shifting the location of the contact point either to S_a (along the face edge) or to S_b (out of the plane of the machined surface), depending on the size of the chamfer angle.

(*f*) Changing the corner angle affects contact conditions of arbor-mounted cutters differently from those of face mills and end mills. In the case of arbor-mounted cutters, a change in the corner angle does not affect initial contact, when the axial rake is negative. However, if the axial rake is positive, it may be either *T*-contact or *S*-contact, depending on the relationship of index angle and corner angle.

3 *For all types of milling cutters: Tool angles on milling cutters such as face mills, end mills, straddle mills, side mills alone do not determine any type of initial contact. It is always necessary to consider also the angle of engagement.*

An existing cutter with given rake and corner angles may have different types of initial contact, depending on the position of the workpiece relative to the cutter axis, and on the cutter diameter. Tool life may therefore vary with the setup of cutter and work.

4 Exit conditions:

(*a*) The point of initial contact is not necessarily identical with the point that leaves the work first.

(*b*) The wider the workpiece, the less likely is the identity of the point of initial contact with the point that leaves the work first.

(*c*) Most frequently, the top point *T* of the cutting edge emerges first (*T*-exit).

(*d*) When the top point of the cutting edge contacts the work first (*T*-contact) it also leaves first (*T*-exit).

(*e*) In the case of a positive axial rake: *V*-contact may be followed by either *S*-exit or *T*-exit; *S*-contact may be followed by *T*-exit, depending on the width of the work and the combination of the other tool angles.

(*f*) In the case of a negative axial rake: *V*-contact may be followed by either *U*-exit or *T*-exit; *U*-contact may be followed by *T*-exit, depending on the width of the work and the combination of the other tool angles.

5 Tool wear:

(*a*) The combined effect of location and magnitude of impact is a major factor in the wear of sintered-carbide tools.

(*b*) A small impact at *U* seems to be the most desirable condition, but a large impact at this point is worse than a small impact at *V*.

(*c*) The most desirable condition can be obtained by careful co-ordination of axial rake, radial rake, corner angle, chamfer angle, and angle of engagement.

Acknowledgment

The author wishes to express his appreciation to Mr. Hans Ernst, research director, and to Dr. M. E. Merchant, physicist, for their helpful advice. The able assistance of Mr. Norman Zlatin, research engineer, and Mr. Robert Adams, who conducted the tool-wear tests, is also acknowledged. Mr. Emil Suhr, laboratory assistant, and Miss Jane Alexander assisted in the preparation of the models.

(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until May 10, 1946)

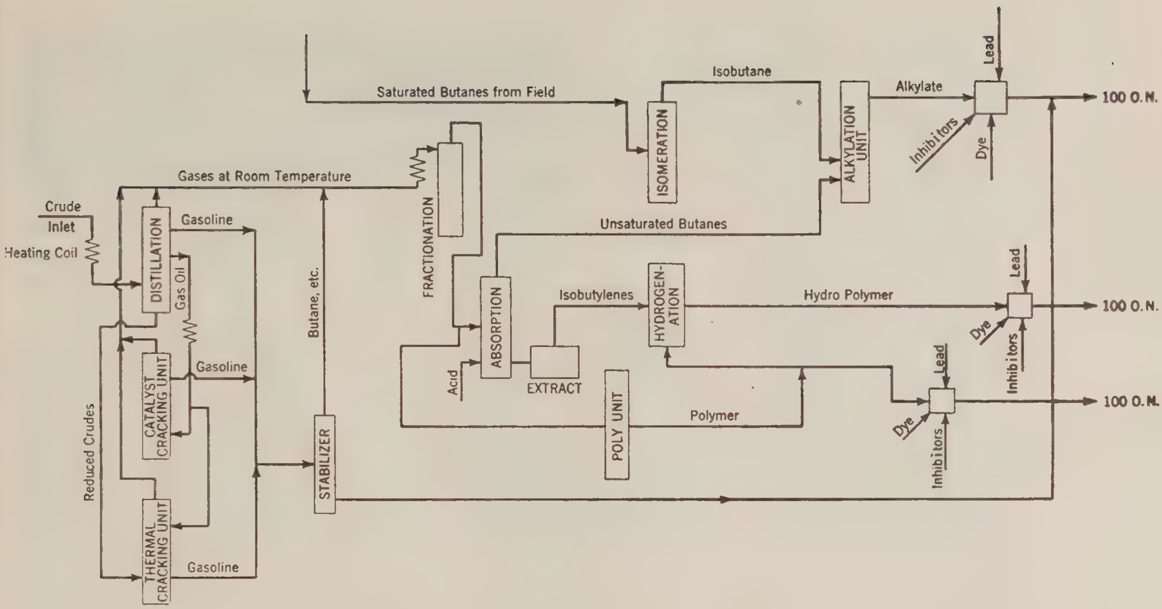


Fig. 1 100-Octane Gasoline Manufacture

Resistance of Nickel-Containing Alloys to Corrosives Encountered in the Petroleum Refineries

By B. B. MORTON,[1] NEW YORK, N. Y.

Production of bulk gasoline in a modern petroleum refinery is accompanied by corrosion during topping and cracking from sulphur and from weak hydrochloric acid which is formed from salts in the crude. Organic acids such as naphthenics are also often present and are quite corrosive. Solutions of refinery corrosion problems are considered in this paper.

The modern petroleum refinery under the stress of producing war essentials, such as high-octane gasoline, rubber, etc., has become a most complex organization.

Fig. 1, which is a rather simplified flow sheet of the processes employed in making 100-octane gasoline, hints at the complexities involved. A view of the towers, tanks, lines, and equipment involved would be more impressive but possibly less instructive.

As a homely comparison a refinery may be likened to a kitchen in which many cooks are turning out a large volume of good soup (standard gasoline). Other cooks, specialists, are busy making a sauce to be added in small parts to the soup to make it superlatively good. This sauce is the additives, made by polymerization, alkylation, hydrogenation, etc. These additives bring up the octane number of the bulk gasoline produced. The final step is the addition of lead compound, the supersauce.

The production of the bulk gasoline is accompanied by corrosion, during topping and cracking, from sulphur and from weak hydrochloric acid which is formed from salts in the crude. Organic acids, as naphthenics, also are often present and are quite corrosive. These acids are apparently becoming more of a corrosive factor during distillation as a result of the crudes used at the present time.

The corrosion problems associated with the formation of the additives are much more pronounced and are more varied than those associated with topping and cracking. This can be understood by considering some of the reagents involved in the manufacture of the additives:

Acids: Hydrochloric, sulphuric, hydrofluoric, phosphoric, etc.
Bases: Sodium hydroxide, potassium hydroxide, ammonia, etc.
Salts: Aluminum chloride, boron chloride, etc.

Corrosion in Refineries

The author has found it convenient to consider refinery corrosion as possessing two different identities, the dividing line being

[1] Development Engineer, Development and Research Division, The International Nickel Company, Inc. Mem. A.S.M.E.

Contributed by the Petroleum Committee of the Process Industries Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

the temperature level at which water is no longer present; an arbitrary figure of 500 F has been taken. Above this temperature "dry" corrosion is considered to take place due to the action of sulphur and also some organic acids; below this temperature "wet" or electrolytic corrosion is assumed to take place.

The nature of electrolytic corrosion has been described in detail in the literature. It is pointed out that a metal tends to go into solution at an anode while hydrogen or another metal plates out at the cathode. It is generally recognized that the areas of the electrodes, the presence of depolarizing agents, and also the relation of the electrodes in a galvanic series, all affect the rates of corrosion. The gradual changes in composition of the film at the face of the electrodes can greatly affect the rate, direction, and nature of the corrosion.

It has been widely pointed out, though not well heeded, that the current passing is the important factor in corrosion by electrolysis rather than the potential between electrodes.

The electrolytic conception of corrosion has provided a working insight into much corrosion. It has also furnished an ingenious but often erroneous explanation of some corrosion. It has awakened a dread of the presence of dissimilar metals, sometimes unwarranted.

A galvanic series of some metals in salt water is provided in Table 1 (1).[2] This series will prove of interest in combating refinery corrosion[3] in condensers, run-down lines, separators, etc. It has been found that galvanic corrosion is greatly suppressed in the absence of oxygen, and this should be considered in evaluating galvanic corrosion in refineries. As light on this subject, Fig. 2 is provided. It will be noted that the nickel plate on the steel sucker rod has failed to produce as serious undercutting as might be expected from the coupling of these two metals. The rod was exposed to the corrosive brine of an Arkansas well. The metal removed is the result of contact with the tubing. The absence of oxygen is assumed to account for the absence of marked galvanic corrosion.

TABLE 1 GALVANIC SERIES OF METALS AND ALLOYS

Corroded End (anodic, or least noble)
Magnesium Magnesium alloys
Zinc Galvanized steel or galvanized wrought iron
Aluminum 52SH Aluminum 4S Aluminum 3S Aluminum 2S Aluminum 53S-T Alclad
Cadmium
Aluminum A17S-T Aluminum 17S-T Aluminum 24S-T
Mild steel Wrought iron Cast iron
Ni-Resist[a]
13% chromium stainless steel type 410 (active)
50-50 lead-tin solder
18-8 stainless steel type 304 (active) 18-8-3 stainless steel type 316 (active)
Lead Tin
Muntz metal Manganese bronze Naval brass
Nickel (active) Inconel (active)
Yellow brass Admiralty brass Aluminum bronze Red brass Copper Silicon bronze Ambrac 70-30 copper nickel Comp. G-bronze Comp. M-bronze
Nickel (passive) Inconel (passive)
Monel
18-8 stainless steel type 304 (passive) 18-8-3 stainless steel type 316 (passive)
Protected End (cathodic, or most noble)

[a] Reg. U. S. Pat. Off.

Fig. 2 Study of Nickel-Plated Steel Sucker Rod From Corrosive Well; Oxygen Absent; ×8
(Note absence of undercutting where plate has worn through in contact with the tubing.)

That highly vigorous attack can be produced under conditions favorable to galvanic attack is shown in Table 2. The data of this table were derived by coupling metals with a carbon-containing material which is used to line vessels against acid attack or to form packing or equipment in vessels. The necessity of insulating to prevent forming a couple of metal with this carbon-containing material can be readily seen. The difficulties of insulating effectively will be great in many cases.

TABLE 2 PRELIMINARY STUDY OF A CARBONACEOUS ELECTRODE COUPLED WITH SOME ALLOYS

Areas {Carbonaceous electrode .... 1.07 dm^2; Coupled metals .... 0.25 dm^2}

Electrolyte	65 per cent sulphuric acid
Temperature	193 C (379 F)
Aeration	None

Couple	Time, hr	Corrosion rates of alloys: M.d.d.	Corrosion rates of alloys: In. per year
Carbon and Hastelloy B[a]	4	59626	9.0+
Carbon and Hastelloy D[a]	16	1266	0.23
Carbon and Monel[b]	4	40413	6.5+

[a] Normally quite resistant to these conditions.
[b] Normally fairly resistant to these conditions.

It must not be assumed that galvanic corrosion is an unmitigated evil. This form of corrosion can be usefully employed and is widely used though often without recognition of the fact. Galvanizing is a nice example of the use of galvanic effects. The iron is protected by the sacrifice of zinc. In general, to use galvanic protection a metal "less noble" than the one to be protected must be coupled with the metal whose life it is desired to prolong. No parsimoniousness can be tolerated in the use of the less noble metal. It must be considered definitely expendable and must be present in suitable mass and area to invite and to sustain attack over the desired period.

The destruction of the less noble metal need not be complete. Some Monel ($^2/_3$ nickel, $^1/_3$ copper) bolts from a condenser, Fig. 3, provide a good example of protection without destruction of the less noble metal. The cast iron in contact with the bolts built up a graphitic layer which protected itself while offering protection to the Monel. The vast area of the cast-iron pipes also tended to offset the attack directed to the pipe by the bolts. The use of greater area of the less noble metal is a common means of

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

[3] It must be recognized that in a different medium the placement of the metals might differ slightly.

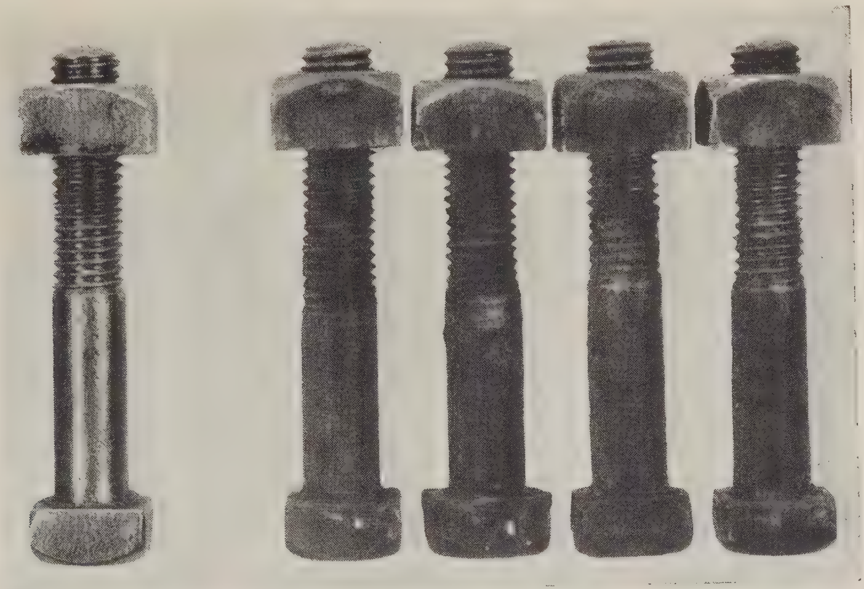


Fig. 3 Monel Bolts Exposed 22 Years in a Condenser Box to Contaminated Salt Water of the Kill van Kull

utilizing galvanic corrosion to protect a part, as a weld or a rivet head, without unduly injuring the less noble material.

General Solution to Refinery Corrosion

A general solution to the two distinct types of corrosion occurring in a petroleum refinery appears as follows:

(a) *Elevated Temperature (Above 500 F)*. Chromium is the metallic element most useful to resist the action of sulphur. The necessary concentration of chromium in the metal appears to depend upon the temperatures used and the concentration of the sulphur compounds; also the nature of these compounds. Metals showing resistance to sulphur which are in wide use are listed in descending order as follows: 18 per cent chromium, 8 per cent nickel; 11 to 14 per cent chrome steels; 4 to 6 per cent chrome steels.

The variety of mechanical properties of these alloys as well as their corrosion resistance dictate the particular choice.

Where naphthenic acids are a source of corrosion at temperatures above 500 F, the following alloys are employed:

Inconel[4] (approximately 80 per cent Ni, 13 per cent Cr)
Type 316 (approximately 18 per cent Cr, 14 per cent Ni, 2–3 per cent Moly)

In some investigations reported, (2) it has been pointed out that 18-8 is more vigorously attacked by naphthenic acid under conditions of elevated temperature than is carbon steel. Both metals were vigorously attacked. Alloy 18-14-3, however, appears quite resistant. Inconel appears to be highly resistant, possibly since nickel itself is, and the 13 per cent chromium present gives protection against sulphur attack which might take place on nickel above 500 F.

Type 316 stainless steel 18-14-3 is finding increasingly wider uses in refineries as they turn toward the manufacture of chemicals from petroleum products. Its resistance to many organic acids is outstanding. It is considered less subject to "weld decay" than is regular 18-8.

The practical application of material to resist corrosion at elevated temperature consists in using tubing of 4–6 per cent chrome or of 18 per cent chromium, 8 per cent nickel alloy, lining equipment with 18-8 or 11 to 15 per cent chromium stainless. Where strip or other welding is used, 18-8, or better, 25 per cent chrome, 20 per cent nickel rod is used.

In some equipment as in certain bubble towers, a wide temperature gradient exists along the axis, say, from 700 to 370 F. In this case a stainless steel is indicated for the temperature region above 500 F, and Monel for the cooler top region Fig. 4. Some very large towers have 18-8 lining, trays, and caps in the high-temperature region and Monel lining, caps, and trays in the top.

Some exchangers exposed to attack from naphthenic acid at temperatures of 700 to 850 F are tubed and equipped with Inconel (approximately 80 per cent nickel 13 per cent chrome).

(b) *Low-Temperature Corrosion (Below 500 F)*. Corrosion occurring within the temperature region below 500 F is considered to be electrolytic in character. It is assumed that liquid water is present to form an electrolyte. The solution to the corrosion of this type is more difficult than in the case of corrosion at elevated temperature, owing to the complex nature of the attacking agents. No single element is effective against corrosion at low temperature as is chromium at elevated temperature against sulphur.

The most satisfactory means for studying corrosion of any type appears to be to install specimens of metals under operating conditions and to observe results. This is particularly true for electrolytic corrosion. Insulating between samples is necessary for this temperature region where water can exist, and this complicates testing. The test spool, Fig. 5 (3), has been developed and improved over a period of years by the author's company. Thousands of specimens have been exposed in various industries, and an impressive mass of data have been accumulated.

From a study of the data it appears that no general rule can be applied to the selection of a material for use in a refinery in order to combat electrolytic corrosion. The selection is best done by studying the exposed samples. Such a study is helpful in picking the most economical material, as the samples will indicate the relative life and the tendency toward pitting. Often the short life of refinery equipment, due to becoming obsolete, dic-

[4] Registered trade-mark name; The International Nickel Company, Inc.

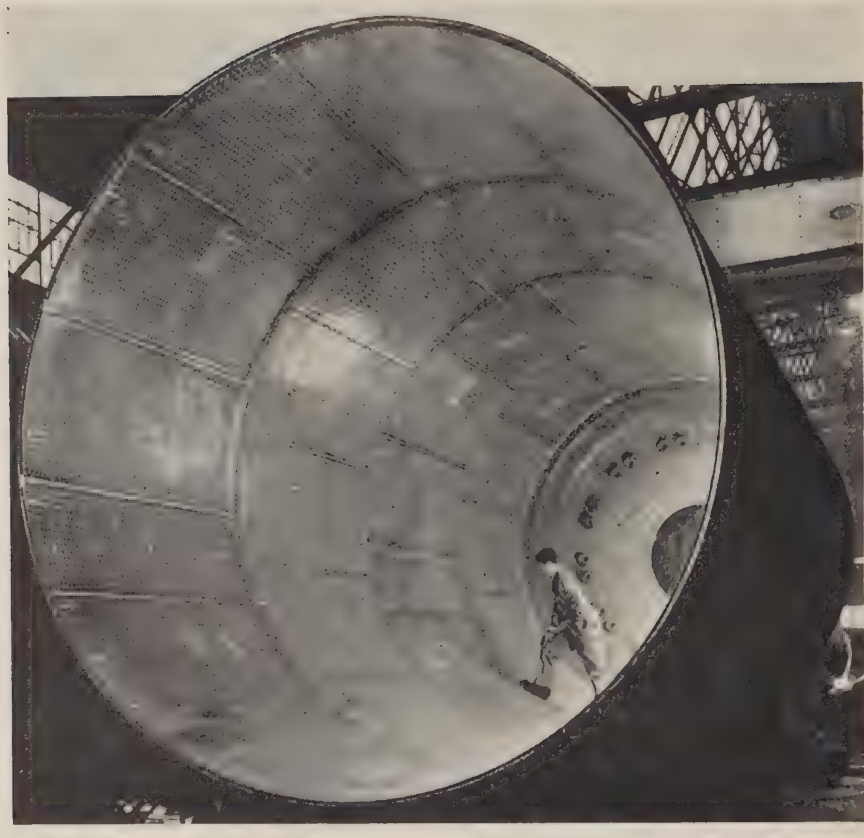


Fig. 4 Section of Large Fractionating Tower the Top Part of Which Is "Smithlined" With Monel Sheet to Withstand Corrosion by Dilute HCL. The Bottom Lining Is of 18-8 to Resist Sulphur

(Monel tower linings are confined to top sections where temperatures do not exceed 500 F.)

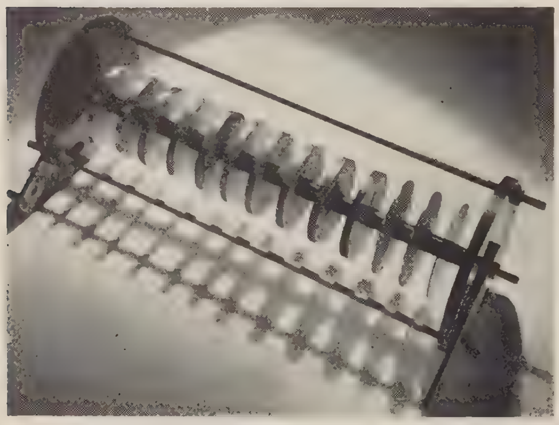


Fig. 5 Spool-Type Specimen Holder Used in Testing Corrosion of Metals

tates a less expensive material than would be selected if indefinite life is desired.

The use of nonferrous alloys to combat electrolytic corrosion is widespread, as will be evident in the consideration of material used in parts of refinery equipment exposed to corrosives.

Condensing Equipment

Condensing equipment suffers much from electrolytic corrosion. As the tendency is increased to improve the efficiency of this equipment the need for better and more corrosion-resistant materials is felt, that is, as the submerged coil of cast-iron pipe is abandoned for shell-and-tube or submerged-tube segments, the need for better material is noted.

Condenser shells are formed either of Monel-clad steel or are protected locally by liners of this alloy, Fig. 6. Where Admiralty tubing is suffering dezincification, the trend before the war was to use 70-30 copper-nickel tubes. Other nickel-containing tubing to thwart dezincification is hinted at as postwar materials.

The 70-30 copper-nickel tube offers greater resistance to impingement attack than does Admiralty tubing.

Isomerization Equipment

The introduction of HCL gas as an atmosphere in an isomerization operation has greatly intensified refinery corrosion problems.

Hastelloy B, an alloy of approximately 60 per cent nickel, 32 per cent molybdenum, is apparently one of the most resistant of commercial alloys to the corrosive conditions evoked by the use of the HCL atmosphere in the isomerization operation. Some data, Table 3, obtained by exposure of samples in an operating unit serve to place Hastelloy B in comparison with some other materials. The position of nickel is also indicated. It has been

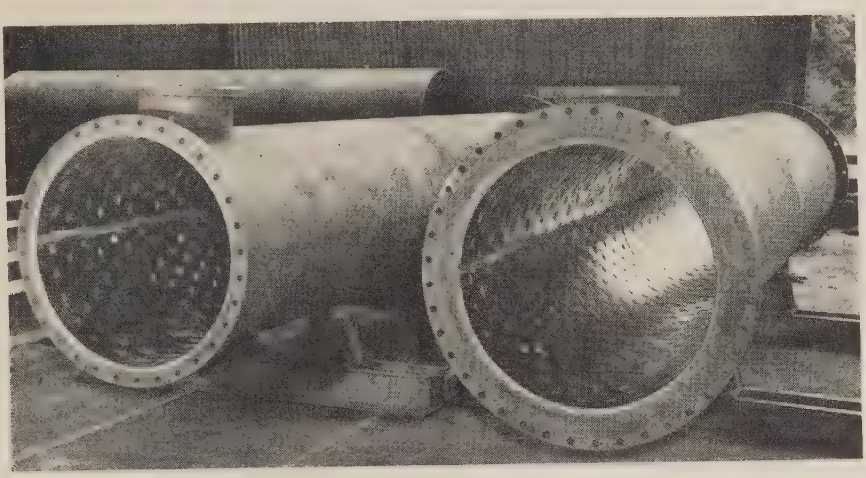


Fig. 6 Monel-Clad Heat-Exchanger Shells

noted in service that nickel is somewhat erratic in its resistance. The cause is not known for the variation noted.

The study of materials for use in isomerization units is proceeding apace. The widespread use of nickel would indicate that certain refinery units understand and can control its rate of corrosion. The success may depend on the efficiency of the dehydrating equipment. High efficiency of these drying units will permit use of the cheaper materials of construction.

Alkylation Units

Alkylation is carried out with the use of hydrofluoric and sulphuric acids as catalysts.

Monel has emerged as possibly the most useful of the commercial materials for parts of units using HF acid for the catalyst. It is now used as liners for vessels; as rings and packing; as valves and fittings, etc. There may be a tendency to use Monel lining with carbon packing. The corrosion engineer so inclined to this use is referred to Table 2 and is warned.

"S"-Monel, a high-silicon alloy (4 per cent Si) (see Table 5) has such excellent resistance to seizing and galling that its use as the plug and other parts of valves was considered for use in HF alkylation units. The vigorousness with which hydrofluoric acid attacks silica caused some concern that the silicon of S-Monel might be attacked. Laboratory results indicate that S-Monel is not attacked with especial vigor by HF, and use is made of this alloy as a plug material.

A welding rod to permit an overlay of S-Monel has been developed by the author's company. Since the rod must of necessity have a silicon content greater than that of S-Monel to compensate losses during the overlaying, the rod was tested in HF acid with results noted in Table 4. The rate of attack on the rod material itself is tolerable, and since the overlay will have the composition of S-Monel, which is apparently adequately resistant, the rod will probably fill a need for a corrosion- and a galling-resistant surface for valve parts.

Sulphuric acid presents many corrosion problems in refineries, especially when it is used as the catalyst in an alkylation unit or as a solvent. This is due to variations in concentrations and temperature. Table 5 lists some materials used in handling sulphuric acid in petroleum refineries.

When the problem involves concentrating acid, Hastelloy D has proved useful and it is used for the elements to introduce heat into the acid.

Pump and valve parts are made from the alloys listed in Table 5, and from similar alloys.

The behavior of sulphuric acid is so influenced by the presence or absence of oxidizing reagents, by the presence of chlorine or other halogens, also by the presence of salts that can form hydrochloric acid, that general recommendations rarely can be made.

TABLE 3 STUDY OF ALLOYS IN ISOMERIZATION UNITS

Corrosive.......Hydrochloric gas and acid
Process.......Isomerization of Butane
Duration.......1050 Hr
Temperature.......240 F avg

	Spool No. 1 (tar)		Spool No. 2 (vapor)	
	Corrosion rate, in. per year (avg)	Pitting during test, in. surface (max)	Corrosion rate, in. per year (avg)	Pitting during test, in. surface (max)
Monel	0.026	....	0.024	0.004
Nickel	0.009	....	0.012	0.010
Inconel	0.010	....	0.012	0.002
18-8 Stainless	....	0.002[a]	.....	0.008
Type 304	0.094	0.002[b]	0.050	0.010
18-8 Mo Stainless	....	....	.....	0.006
Type 316	0.111	....	0.048	0.004
Hastelloy B	0.000	....	0.000	....
Silver	0.047	Perforated	0.065+	Entirely corroded away
Copper	0.13+	Entirely corroded away	0.043	0.003
85-5-5-5 Bronze	0.145	....	0.058	....
Ni-Resist	0.052	....	0.051	....
Mild steel	0.162	0.005[c]	0.081	0.002
Cast iron	0.085	....	0.082	....

[a] 1/2 of specimen corroded away.
[b] 1/3 of specimen corroded away.
[c] 1/8 of specimen corroded away.

TABLE 4 MONEL VERSUS HYDROFLUORIC; COMPARISON OF SILICON-CONTAINING "H" AND "S" MONEL WITH REGULAR MONEL[a]

Duration of tests, air-free.............. 6 Days
Duration of tests, air-saturated........ 1 Day (24 hr)
Velocity........42 to 73 fpm

	Average corrosion rate, in. penetration per year			
	At 86 F		At 176 F	
	Air-free	Air-saturated	Air-free	Air-saturated
Hydrofluoric acid 25 per cent (by weight)				
Monel sheet	0.0002	0.037	0.0024	0.011
Cast Monel	0.0006	0.019	0.0013	0.020
H-Monel	0.0004	0.019	0.0004	0.022
S-Monel	0.0002	0.009	0.0002	0.021
Hydrofluoric acid 50 per cent (by weight)				
Monel sheet	0.0001	0.008	0.0006	0.039
Cast Monel	0.0005	0.006	0.0022	0.037
H-Monel	0.0002	0.007	0.0009	0.044
S-Monel	0.0004	0.003	0.0020	0.046
S-Monel welding rod, 6.6 per cent silicon				0.018
S-Monel welding rod, 5.2 per cent silicon				0.013

Note accelerating effect of oxygen. In refinery service this factor rarely appears.

[a] Laboratory tests in 25 per cent and 50 per cent hydrofluoric acid at 86 F, and 176 F. Specimens were suspended in liquid hydrofluoric-acid solutions in a closed nickel autoclave equipped with nickel reflux condenser. Tests in air-free solutions made by bubbling nitrogen continuously through the acid, and air-saturated tests made by bubbling air continuously through.

TABLE 5 REPRESENTATIVE LIST AND APPROXIMATE COMPOSITIONS OF SOME NICKEL-CONTAINING ALLOYS USED IN CONNECTION WITH SULPHURIC ACID

Material	Nickel, per cent	Copper, per cent	Iron, per cent	Chromium, per cent	Molybdenum, per cent	Aluminum, per cent	Silicon, per cent	Manganese, per cent	Tungsten, per cent	Carbon, per cent
Nickel	99.4[a]	0.1	0.15	...	..	..	0.05	0.2	...	0.1
Monel	67.	30.	1.4	...	..	..	0.1	1.	...	0.15
H-Monel[b]	65.	29.5	1.5	...	..	..	3.	0.9	...	0.1
S-Monel[b]	63.	30.	2.	...	..	..	4.	0.9	...	0.1
K-Monel	66.	29.	0.9	...	..	2.75	0.25	0.4	...	0.15
Inconel	79.5	0.2	6.5	13.	..	..	0.25	0.25	...	0.08
Hastelloy A	53.	..	22.	...	22.	..	1.	2.	...	..
Hastelloy B	60.	..	6.	...	32.	..	1.	1.	...	..
Hastelloy C[b]	51.	..	6.	17.	19.	..	1.	1.	5.	..
Hastelloy D[b]	85.	3.	...	...	..	1.	10.	1.	...	..
Illium G[b]	60.	4.	6.	21.	6.	..	1.	1.	1.	..
Illium R	60.	3.	8.	21.	5.	..	1.	1.	1.	..
LaBour[b]	53.	5.0	8.0	23.	4.0	..	4.0	0.5	2.0	..
Duramet T	22.	1.0	Bal.	19.	3.5	..	...	..	...	0.07
Worthite	24.	1.7	Bal.	20.	3.0	..	3.3	0.6	...	0.07

[a] Including cobalt.
[b] Only as castings.
NOTE: Specific use can be determined by making tests or by seeking suggestions from suppliers.

The action of the presence of copper from copper-alloy fittings in the destruction of otherwise resistant alloys has been reported (2).

The most resistant metallic materials to a wide variety of concentrations of sulphuric acid and to temperature variations and impurities in the acids to come to the author's attention are the high-silicon cast irons, such as "Duriron" and "Corrosiron." The brittleness of these materials somewhat limits their usefulness but when considered in design they are used for many pieces of equipment.

Caustic Regenerators

Caustic is used both to neutralize acid and as a solvent for organic compounds. As an example of the latter consider mercaptan scrubbing and recovery.

Monel appears to be one of the most useful materials with which to line caustic regenerators, and for caps and trays. The tubes of the reboiler are often of Monel which appears quite satisfactory. The service of the tubes is severe since boiling and high attending velocity often injure protective films. An alloy of 70-30 copper nickel appears unsatisfactory for this service of tubes in reboilers in units recovering mercaptans from caustic. The vigorous attack of sulphur upon the copper-rich alloy and subsequent injury of film may account for the failures of 70-30 so far reported.

Before undertaking to line an existing tower that has been in a caustic regenerating unit with Monel, it is well to consider that sulphur on the steel wall may cause embrittlement of plug or other welds. The sulphur and adhering caustic must be entirely removed. Discussion of the proposed lining with the Technical Service group of the author's company might be advisable since observations to date could be presented.

Ni-Resist cast iron, a nickel or nickel-copper cast iron which is austenitic in structure, i.e., nonmagnetic, is useful for pumps and valve parts of equipment handling caustic.

Nickel-clad steel appears to be the favorite material of construction in units employing caustic and an organic compound to act as a solvent for mercaptans.

Solvents

Many solvents find application in petroleum refineries. In fact the use of selective solvents has revolutionized the art of refining.

Many of the solvents are corrosive toward steel. Others not originally corrosive become so after use and contamination.

For such solvents as phenol, furfural, and chlorinated compounds, Monel or 18-8 stainless will be found quite satisfactory. The 18-8 is the proper material to use at temperatures above 500 F. Monel is considered more satisfactory for use with chlorinated solvents at the temperature at which these solvents are used.

Solvents are often used at subzero temperatures of —50 F, —75 F, and —150 F. At these low temperatures corrosion ceases to be a problem but the impact value of many nonheat-treated steels suffers deterioration. Normalized nickel-containing steels retain a large part of the room-temperature impact value at low temperatures when properly made for this service. Nickel steels of $2^1/_2$ per cent are used at temperatures down to —75 F. Nickel steels of $3^1/_2$ per cent, properly deoxidized and aluminum-treated, should provide suitable material for vessels to operate at —150 F. The carbon of the steel should be low, preferably below 0.10 per cent. For uses at lower temperatures one should consider the austenitic stainless steels or Monel.

In the foregoing, plates and large parts capable only of being normalized are considered. Nearly all alloy steels that can be fully hardened by quenching and drawing will give good impact value at very low temperatures.

Hydrogen Attack

Hydrogenation is an important tool in petroleum refineries. Under conditions of temperature and high partial pressure of hydrogen, attack takes place upon steel and other materials in that there is a tendency toward decarburization and mechanical disintegration.

Sulphur attack also is a factor in much of the hydrogenation of petroleum.

The alloy, 18 per cent chrome, 8 per cent nickel, is highly resistant to hydrogen attack. It suffers some temporary loss in mechanical properties due to mechanical presence of hydrogen in the metal. The hydrogen can be removed by heating, and the properties restored.

For other steels the presence of carbide-forming elements as chromium, tungsten, molybdenum, and others are required to prevent decarburization. In general, the more complex the carbide the more resistant is the metal to decarburization under hydrogenation conditions. Unless 12 per cent or more of chromium is present sulphur attack must be expected.

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Creep of Neoprene in Shear Under Static Conditions

By W. N. Keen,[1] Wilmington, Del.

This paper presents some of the mechanical properties, particularly the creep characteristics, of neoprene and natural-rubber vulcanizates. Included along with the creep data are the mechanical properties obtained in both compression and shear on the vulcanizates used in this study.

In an earlier paper Yerzley (1)[2] showed how changing the base composition by the use of different types of softeners and reinforcing fillers influenced the physical properties, particularly the creep characteristics, of neoprene vulcanizates.

This paper shows how the creep in shear of neoprene vulcanizates in the hardness range of 40 to 60 durometer (Shore Type A) is influenced by the following:

1 Degree of vulcanization.
2 Preconditioning in shear at elevated temperatures.
3 Changes in composition.

The long-time static creep-tests discussed herein show that as the degree or state of vulcanization of neoprene compositions is increased, the rate of creep and the set of these vulcanizates are decreased without appreciably altering other important mechanical properties.

Preconditioning at elevated temperatures reduces both the rate and the magnitude of creep to a greater extent for the neoprene vulcanizates than for the natural-rubber vulcanizate tested.

The greatest improvement in creep characteristics has resulted from changes in the composition of the vulcanizates.

Both the composition and the degree of vulcanization must be studied in order to obtain the most desirable combination of properties for a given application. Such a study should be broad enough to cover independent evaluation of the important mechanical properties involved.

Method of Testing Samples

The tests described in this paper were conducted at a controlled temperature of 82 F ± 2 F under a load of 17.75 psi. The sandwich or specimen designed for evaluating vulcanizates in shear on the Yerzley oscillograph (2) was used in this work. Fig. 1 shows the static-creep test stand with specimens under test. The rectangular rubber members of the specimen are each 0.5 × 0.5 × 0.882 in. The schematic sketch, Fig. 2, gives the details of the loading bar showing its knife-edge construction and the connecting linkage between the test specimen and the loading bar. The linkage is readily adjustable so that the bar may be leveled as creep takes place in the specimen. Creep in inches was measured by a dial gage (0.001) equipped with a special jig.

In order to present the data for the several compositions on a comparable basis, they are expressed in terms of relative creep, as defined by Yerzley (1). Relative creep may be expressed mathematically as follows:

$$\text{Relative creep} = \frac{\text{Total deformation} - \text{initial deformation}^*}{\text{Initial deformation}^*}$$

In other words, the relative creep at 5 min is zero in all cases.

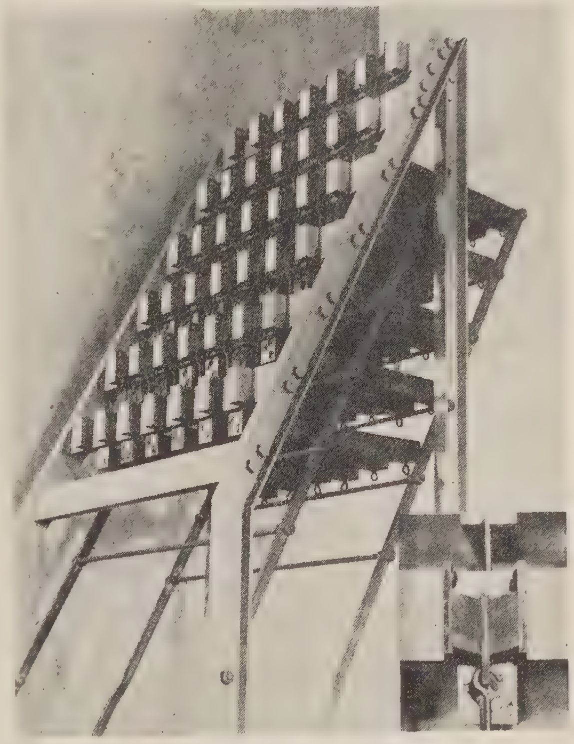

Fig. 1 Static-Creep Test Stand and Detail of Specimen

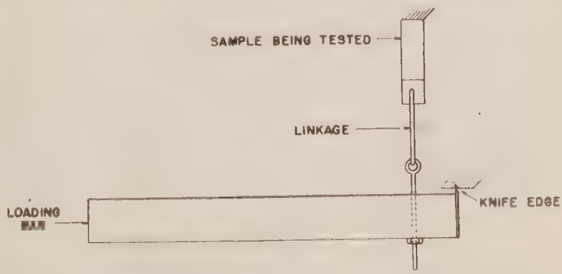

Fig. 2 Schematic Sketch of Test Specimen, Loading Bar, and Linkage

* Deformation obtained 5 min after loading.

[1] Rubber Chemicals Division, Organic Chemicals Department, E. I. du Pont de Nemours & Company. Jun. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Rubber and Plastics Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

TABLE 1 MECHANICAL PROPERTIES OF COMPOSITIONS

Specimen	A.S.T.M. grade no.	Cure min/deg F	Hardness, Shore A	Per cent set, A.S.T.M. methods A	B	Compression characteristics: Resilience at 20 per cent deformation, per cent	Frequency, cycles per min	Static modulus, psi: At 0 per cent deformation	At 20 per cent deformation	Load at 20 per cent deformation psi	Effective dynamic modulus, psi
S-1	SC-425	30/287	44	21	39	87	225	485	650	107	792
S-2	SC-525	60/287	46	13	25	88	227	490	690	117	797
S-3	SC-525	30/307	47	12	24	88	227	470	685	115	857
S-3D	SC-525	30/307	..	..	..	..	...	...	...	...	...
S-4	SC-425	80/307	41	11	18	83	234	435	645	105	847
S-6	SC-420	80/307	43	12	19	64	273	345	390	71	884
S-7	RN-530	15/287	46	11	24	90	193	500	810	130	835
S-7D	RN-530	15/287	..	..	..	..	...	...	...	...	...
S-8	SC-614	40/307	59	10	32	75	...	...	795	160	...
S-9	SC-612	65/307	59	10	18	58	...	...	770	160	...
S-10	SC-614	45/307	59	4	9	77	237	710	870	200	1070

The actual creep in inches is included in order to give the design engineer some idea of the magnitude of the dimensional changes resulting from creep.

As the state or degree of vulcanization of neoprene is increased (S-1 versus S-2), the rate of creep and the set of the vulcanizates decrease. The decrease in the rate of creep with increased state of vulcanization is shown graphically in Fig. 3, where per cent relative creep is plotted against time on log-log paper. The change in rate of creep or slope values B are given in Table 1. It will be noted in Fig. 3 that the magnitude of relative creep is also reduced for this composition as the degree of vulcanization is increased. The numerical values for creep at the end of 3 years and 9 months are given in Table 1.

Table 1 also gives the compression and shear set values determined under constant load and constant deflection. The shear set values were determined in accordance with A.S.T.M. D-395-40T with respect to time and temperature. These values indicate the actual shear strain remaining in the specimen after 30 min recovery at room temperature. As would be expected, there is little difference in the physical properties between S-2 and S-3 since the time and temperature relationship is such that the degree of vulcanization for them is about equal. From the manufacturer's point of view the shorter time of vulcanization at a higher temperature would be more desirable.

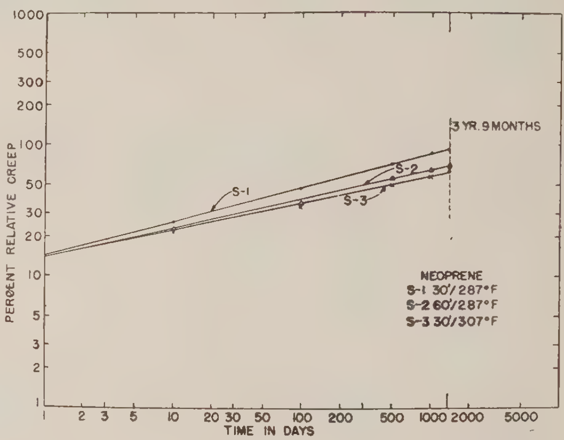

Fig. 3 Influence of Degree of Vulcanization on Static Creep in Shear of Vulcanizates of Same Neoprene Composition

Preconditioning to Reduce Creep

Another way to reduce the amount of creep of some vulcanizates is by preconditioning. Preconditioning of the neoprene (S-3D) and the natural-rubber (S-7D) specimens consisted of straining them 30 per cent in shear and heating in this condition for 24 hr at 158 F. After the heating period, the specimens were removed from the jig and permitted to rest for one week at 82 F before being mounted on the static-creep test stand. Fig. 4 shows how this preconditioning in shear reduces the rate of creep for a limited time for both neoprene and natural-rubber vulcanizates. These curves indicate that neoprene is more receptive to preconditioning as a means of reducing creep than is natural rubber. Deviations from a straight-line function similar to those exhibited by curves S-3D, S-7, and S-7D have been reported by other investigators (3).

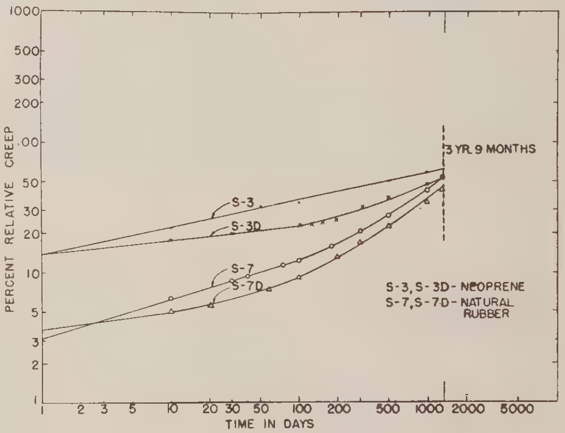

Fig. 4 Influence of Preconditioning, in Shear, on Creep in Shear of Neoprene Vulcanizate, Compared to Natural-Rubber Vulcanizate

Effect of Changing Composition

Fig. 5 presents some idea of how the relative creep may be varied by changing the composition. If this group of curves is considered from the standpoint of slope or rate of change of relative creep, they are all superior to the natural-rubber control except specimens S-4 and S-10. Specimens S-6, S-8, and S-9, would undoubtedly be of greatest interest to the design engineer because of their low rate of creep, as indicated by their slopes of less than 0.20. It will be noted, however, that while all of the curves except S-4 and S-10 have a lower slope than the natural rubber (S-7); all of them except S-10 have a higher per cent relative creep after 1 day. This higher initial creep can readily be taken care of by the engineer in his design of a spring or mounting.

In view of the fact that the relative creep - time curves are linear when plotted on logarithmic co-ordinates, they may be expressed by the equation

$$\text{Relative creep} = At^B$$

where A = Y intercept at 1 day
B = slope of curve
t = time in days

TABLE 1 (*Continued*)

Specimen	Shear characteristics: Per cent set constant load, 17.75 psi	Per cent set constant deflection, 20 per cent	Resilience at 20 per cent deformation, per cent	Frequency, cycles per min	Static modulus, psi: At 0 per cent deformation	Static modulus, psi: At 20 per cent deformation	Load at 20 per cent deformation, psi	Effective dynamic modulus, psi	Static-creep characteristics in shear: Under load of 17.75 psi: Initial deformation, in.	Initial shear, per cent	Per cent relative creep, 3 yr 9 mo	Creep in inches, 3 yr 9 mo	Y intercept A	Slope B
S-1	5.0	3.5	85	167	100	106	20	204	0.0930	18.6	92.3	0.0858	0.158	0.244
S-2	4.2	2.8	86	169	120	107	22	210	0.0858	17.2	69.4	0.0595	0.140	0.222
S-3	4.0	2.7	87	176	115	105	22	227	0.0883	17.7	63.0	0.0556	0.142	0.205
S-3D	...	...	..	...	...	...	..	...	0.0843	16.9	53.6	0.0452	0.140	0.121[a]
S-4	4.6	3.6	83	159	102	95	20	180	0.0935	18.7	42.0	0.0393	0.050	0.295
S-6	6.3	3.8	66	174	91	73	17	224	0.1530	30.6	65.0	0.0995	0.217	0.155
S-7	4.5	4.0	90	161	121	125	25	231	0.0707	14.1	54.2	0.0383	0.031	0.285[a]
S-7D	...	...	..	...	...	...	..	...	0.0717	14.3	46.0	0.0330	...	...
S-8	...	...	80	202	150	145	30	363	0.0591	11.8	104.0	0.0615	0.296	0.174
S-9	...	...	63	193	135	130	28	314	0.0649	12.9	50.6	0.0328	0.194	0.129
S-10	1.8	2.4	81	187	197	173	38	365	0.0465	9.3	22.0	0.0102	0.026	0.294

[a] Slope of straight-line portion of curve.

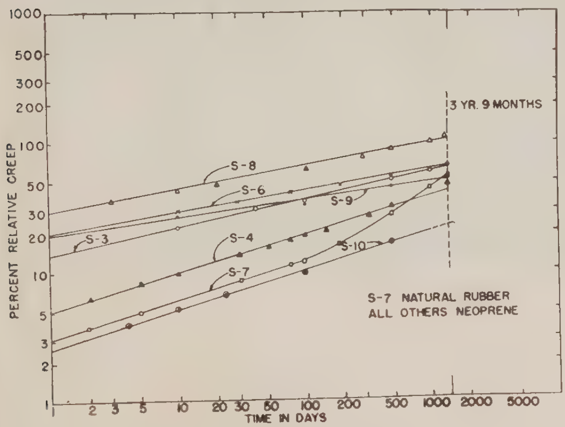

FIG. 5 INFLUENCE OF COMPOSITION ON CREEP IN SHEAR OF NEOPRENE VULCANIZATES

Table 1 gives the values for A and B for all of the curves except S-7D which has a variable slope.

As has been shown, the A.S.T.M. set, and the creep characteristics of neoprene are improved as the state of vulcanization is increased. Likewise some of the other mechanical properties in both compression and shear are increased as the state of vulcanization progresses (see Table 1). The compression characteristics were determined on cylinders 0.75 in. diam and 0.5 in. high, molded from the same composition used in making the shear specimens.

In view of the popular belief that high resilience is indicative of low creep, it is interesting to note that neoprene specimens S-6, S-8, and S-9, which have the lowest rate of creep of the compounds investigated, also have the lowest resilience. This is further evidence to confirm Yerzley's (1) statement that there is no simple correlation between creep rate and resilience.

Change in Properties With Age and Temperature

Since engineers are always interested in knowing how age affects the material with which they are working, the following data were obtained. The static creep test in shear after 3 years was terminated on specimens of S-1, S-4, S-7, and S-10, and the set in shear remaining 24 hr after removing the load was recorded. The change in the mechanical properties of these four specimens and in the specimens of the same four vulcanizates which had been stored at 82 F in an unstressed condition for 3 years were obtained by means of the Yerzley oscillograph. Data given in Table 2 show that natural aging for 3 years has caused a slight stiffening of the neoprene vulcanizates which is reflected by the increase in resilience, frequency, and moduli. There is little difference between samples aged in a stressed condition and those aged in an unstressed condition.

Knowledge of how the mechanical characteristics of a vulcanizate are affected by changes in temperature is for most applications essential to the design engineer. Riesing (4) has shown the effect of temperature on some of the physical properties of neoprene vulcanizates. He presented data showing the functional stability of neoprene at different temperatures by installing neoprene motor mountings, muffler support, and spring-shackle bushings in a car and conducting tests at ambient temperatures of —40 F and +140 F.

Both Forman (5) and Liska (6) have reported that some neoprene vulcanizates show delayed stiffening on long exposure to moderately low temperatures (32 F). This stiffening is the result of crystallization of the elastomer. From a practical consideration crystallization is not as serious as it sounds because the vulcanizate returns to its original state when the ambient temperature is increased or when there is mechanical work done on the vulcanizate. Recent technological advancements in the

TABLE 2 EFFECT OF NATURAL AGING ON MECHANICAL PROPERTIES

Specimen	A.S.T.M. grade no.	Cure Min/deg F	Resilience at 20 per cent deformation, per cent	Frequency, cycles per min	Static modulus, psi: At 0 per cent deformation	Static modulus, psi: At 20 per cent deformation	Load at 20 per cent deformation, psi	Effective dynamic modulus, psi	Creep, in.	Shear strain, per cent	Per cent relative creep	Per cent shear set (24-hr recovery)
					Original shear characteristics							
S-1	SC-425	30/287	85	167	100	106	20	204				
S-4	SC-425	80/307	83	159	102	95	20	180				
S-7	RN-530	15/287	90	161	121	125	25	231				
S-10	SC-614	45/307	81	187	197	173	38	365				
			Shear characteristics after a 3-year static-creep test (under a load in shear of 17.75 psi at 82 F)									
S-1			90	175	115	118	24	225	0.0675	34.7	81.3	18.4
S-4			85	174	128	118	24	224	0.0366	28.6	39.2	13.5
S-7			90	161	133	130	26	232	0.0315	24.0	43.3	9.5
S-10			83	194	220	195	41	396	0.0092	14.5	19.9	6.0
			Shear characteristics after 3 years of aging in an unstressed condition at 82 F									
S-1			91	170	125	110	24	212				
S-4			82	183	115	118	22	246				
S-7			91	163	130	135	27	236				
S-10	No specimen available											

processing of neoprene have led to a practical way of preventing crystallization (7).

Although static-creep tests give valuable information, an obvious disadvantage is the fact that they are time-consuming, and technical advancement often results in a composition becoming obsolete before the testing is completed. This emphasizes the desirability for a more rapid method of testing the creep characteristics of elastomeric materials. Considerable experimentation was carried out in the du Pont rubber laboratory during the past year for the purpose of developing a satisfactory dynamic-creep test. Apparatus has been built and is in operation.

Acknowledgment

The author takes pleasure in acknowledging the important contributions of N. L. Catton and A. M. Neal of the Rubber Chemicals Division of the du Pont Company to the progress of this work. He is also indebted to M. A. Gass, who made most of the measurements.

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Fig. 1 "Plaspreg" Block Serves for Laminating Purposes

Physical Properties of Resin-Impregnated Plaster

By JOHN DELMONTE,[1] LOS ANGELES, CALIF.

While development work on impregnated plaster bodies has been in progress for many years, it was not until 1945 that the process was finally commercialized with low-viscosity furane resins, and extensive applications undertaken in the tooling field, and in general improvements of plaster-of-Paris patterns and forms. The impregnation of plaster of Paris with liquid thermosetting resins and the subsequent conversion of the product with heat, result in the formation of a material which has been called "Plaspreg." In this paper the methods of producing this material, its physical properties, and applications are discussed.

THE impregnation of plaster of Paris with liquid thermosetting resins and the subsequent conversion of the product with heat results in the formation of an entirely new material of construction. The resinified plaster body still retains the crystalline structural network formed on the hydration of $2CaSO_4.H_2O$, although considerably augmented through infusion with synthetic resins. In appearance and feel the impregnated plaster body resembles a cast plastic material more than it does the ordinary plaster-of-Paris bodies, though the improvements which are possible extend much further than the appearance, as all properties are greatly enhanced.

While development work on impregnated plaster bodies has been in progress for many years, it was not until 1945 that the process finally became commercialized with low-viscosity furane resins, and extensive applications undertaken in the tooling field and in general improvements of plaster-of-Paris patterns and forms. While the range of usefulness of resin-impregnated plaster (Plaspreg) forms will be analyzed more completely later in this paper, it will suffice for the moment to point out that among the applications are master models built of plaster of Paris (weighing several hundred pounds) which have been rendered much harder and more permanent through proper impregnation of furane resin; dies for hydropress operation in forming sheet metal; Keller pattern and duplicating dies; core boxes for the foundry; forms for laminating polyester resins; contoured shapes for shaping acrylic plastics; and general improvement in properties of dozens of plaster-of-Paris patterns of industrial and decorative value.

Method of Application

The application of low-viscosity furane resins to the hardening

[1] Consulting Engineer, Furane Plastics and Chemicals Company, Los Angeles, Calif., and Technical Director, Plastics Industries Technical Institute. Mem. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

of plaster-of-Paris bodies is a relatively simple procedure. Plaster of Paris or "hydrocal" forms are prepared in exactly their usual manner, i.e., added to water in certain optimum proportions and then stirred and poured. Depending upon the grade of plaster, the forms set in approximately 15 to 20 min, reproducing with the utmost fidelity the finest details of the mold. The sections can be splash castings $^1/_4$ in. thick or sections measuring several feet in thickness. The hydration process is an exothermic reaction and some heat is developed although temperatures seldom exceed 50 C. After the plaster has firmly set, it may be removed from the mold. There is a considerable amount of free water present which should be evaporated for best results. This can be accomplished by forced drying or leaving in the open for several days. Up to this point the process is standard practice in so far as the plaster industry is concerned. The next step is the application of low-viscosity furane resin which is applied by dipping, brushing, or spraying. The rate of resin diffusion is very rapid, and after thorough impregnation parts are generally cured at temperatures of 140 to 160 F. The final impregnated body is the entirely new structural material which has proved so useful.

The application of organic plastics to plaster-of-Paris bodies is not entirely a new idea, as coatings have been sprayed on the material for many years, largely to seal its surface or to apply some decorative finish. The writer first began experimenting with the impregnation of plaster some 5 years ago and has successfully applied many types of thermosetting urea, melamine, phenolic, and polyester resins and thermoplastic resins. Some of the background leading to the development of furane-resin-impregnated plaster was described in a recent paper before the S.P.I. (1).[2]

The final procedure selected for preparing Plaspreg involved a low-viscosity furane resin whose viscosity could be readily adjusted to meet various application conditions, not by methods of diluting with solvents such as water, alcohol, or acetone, but through proper polymerization conditions. Thus a complete liquid resin capable of conversion to an infusible, insoluble body was developed exclusively for plaster-of-Paris treatment.

A typical application of furane-resin impregnation to plaster-of-Paris bodies is shown in Fig. 1 illustrating a die for laminating purposes. A polyester resin has been laminated about the impregnated form and cured at 250 F. The technique of plaster impregnation was suggested by the author in 1943 (2). Further work upon hydrocal was cited in a more recent publication (3). A further example of furane-resin impregnation is shown in Fig. 2. The many fine details made possible by this impregnation and curing process are emphasized.

Fig. 2 Complicated Design Details, Maintained in Resin-Impregnated Plaster

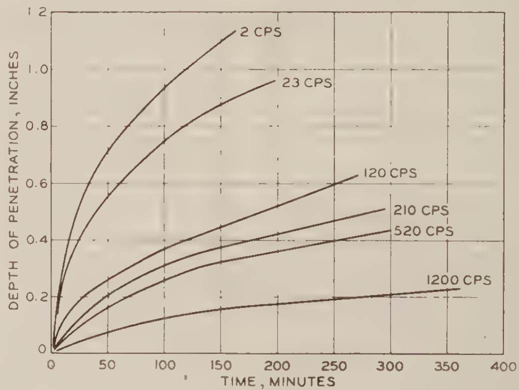

Fig. 3 Amount of Penetration Versus Time for Different Viscosities of the Same Furane Resin

Variables Which Influence Rate and Ease of Impregnation

Throughout all of the work which has been accomplished in the impregnation of plaster of Paris, it has become apparent that best results are obtained through infusion with low-viscosity resins into dry plaster-of-Paris bodies. There are several variables which will influence the rate and ease of impregnation. These are as follows: (*a*) Inherent resin viscosity in the absence of solvent; (*b*) relative porosity of the plaster structure, determined largely by the initial solids-to-water ratio; (*c*) temperature of the plaster-of-Paris body; (*d*) presence of impurities upon the surface. Inasmuch as the correct development of the physical properties of resin-impregnated plaster depends upon these variables, they will be discussed briefly.

(*a*) *Resin Viscosity.* The resin viscosity has a definite influence upon the rate of penetration into the plaster. The higher the viscosity, the higher is the molecular weight and the slower the rate of penetration. The mobility of the resin and its ability to wet surfaces are functions of resin viscosity and, due to the capillary nature of the diffusion process into plaster bodies, resin viscosity is most important. This fact is graphically illustrated in Fig. 3 where the amount of penetration is plotted against time for different viscosities of the same furane resin. The much slower penetration rate of the higher-viscosity resins should be noted. Variations in viscosity were achieved by polymerizing the resin to various degrees, otherwise the material is chemically the same. Another interesting observation which can be made is that the rate of penetration approaches a constant value after a period of time.

While the test method adopted precluded measurements of penetrations much beyond $1^1/_4$ in., it will be noted that complete cessation of penetration did not occur. A relatively simple technique was devised in which the plaster of Paris was poured into a

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

glass vial and then given an opportunity of thorough drying at low temperatures. Because of the slight expansion in setting, a tight fit was insured against the side of the glass vial. The penetration could be observed at the side of the glass vial and measurements made at frequent intervals. These checked completely with any observation upon samples cut open after different intervals of immersion in the resin.

From the data accumulated it was possible to express the rate of penetration by the following equation:

$$r = r_0\left(1 - e^{\frac{-kt}{u}}\right)$$

where

- r is rate of penetration at any time
- r_0 is final steady-state penetration rate
- k constant, depending upon physical factors, such as porosity
- t time
- u viscosity

Thinning the resin with low-viscosity solvents does not produce increased rates of penetration of anything except the solvent, primarily because the plaster-of-Paris body will act as a filtering medium and separate the higher molecular-weight components from their carrier. This means that a thick resin, such as a phenolic casting resin, may presumably be thinned to a low enough viscosity for good penetration. However, during application only the solvent will penetrate, with the high-viscosity phenolic resin remaining near the surface. Other variable factors such as method of application of the resin impregnant, will influence the penetration of the resin. Brushing on the resin and keeping it in motion will aid penetration.

(*b*) *Porosity of Plaster.* The porosity of the plaster has a definite bearing on the amount of resin which can be absorbed. It is seldom possible to produce a workable mix of plaster and water with less than 60 lb of water per 100 lb of plaster. Of the water used, however, only 18.5 lb are required to set the plaster chemically and the balance of 42.5 lb represents free water. This water is evaporated or forced out by oven-drying, leaving voids to be filled by the resin impregnant. While the author has employed a number of water-soluble resins for diffusion into plaster, and in this manner obtained a good penetration followed by subsequent hardening, the total amount of resin pickup is obviously less than when a 100 per cent resin impregnant is employed for application to a plaster in which the free water has been removed.

In Fig. 4 the relationship between the water and plaster ratio and the maximum theoretical pickup is illustrated (calculated on the basis of a 1.18 specific-gravity resin). The resin pickup is based on the total removal of free water after the cast has been made. In addition, the decrease in compressive strength of the plaster is illustrated, indicating that when large amounts of water are employed, the percentage of voids is greater and the effect upon the strength more pronounced. The values for compressive strength of the non-resin-treated plaster are obtained upon dry plaster which is appreciably stronger than wet plaster. At the same time, the compressive strength of resinified plaster is also illustrated. The reinforcing effect of the furane-resin impregnant is quite apparent, as even low-density lightweight plasters are considerably improved. The same improvement is also noted if, during the process of drying or during use, some of the water of crystallization is removed. Resin impregnation overcomes weakness or loss of strength in plaster.

To demonstrate the increased rates of resin impregnation for more porous plasters, a group of test specimens was prepared from 100/60 (solid-to-water ratio) to a rate of 100/100. When these had been thoroughly dried of free water, resin-impregnation tests

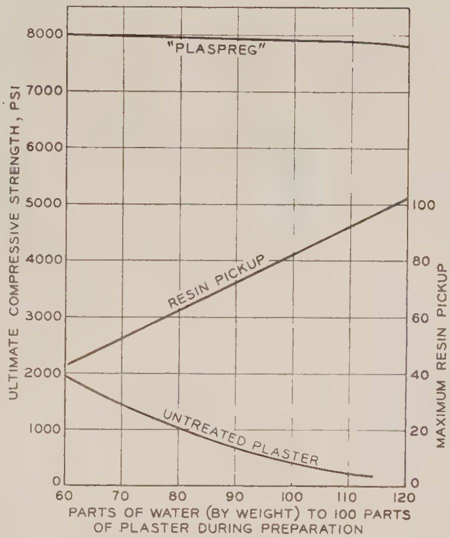

Fig. 4 Relationship Between Water and Plaster Ratio and Maximum Theoretical Pickup of Furane Resin

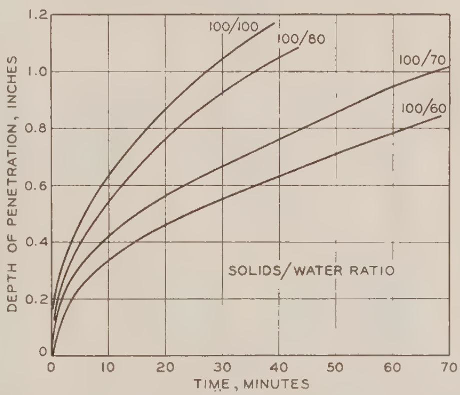

Fig. 5 Effect of Porosity of Plaster on Penetration

were conducted and depth of penetration versus time observed. Averages of several test results are reported in Fig. 5 for a furane resin of initial viscosity of 2 to 3 centipoises. Viscosity measurements were conducted at 25 C, with a "Zahn" viscosimeter cup purchased from the General Electric Company. The increased rates of penetration for more porous plasters are readily indicated upon this curve sheet.

The porosity may also be checked by the water absorbed and in some processes, such as pottery manufacture, this is an important procedure. Aside from this application, low water absorption is an asset. In resinified plaster, where voids are substantially filled with furane resin, water absorption is very low; less than 2 per cent A.S.T.M. water absorption in 24 hr when properly prepared. Herein lies the utility of furane-resin-impregnated plaster tools and articles from a weathering standpoint; they are substantially more satisfactory than untreated plasters. Within a few minutes' immersion, most plasters of the 100/plaster/60 H_2O composition will have a water absorption of 25 per cent. Parts will deteriorate rapidly, while nonporous resinified plaster is infinitely more satisfactory.

(*c*) *Temperature of Plaster.* While not of major consequence

the temperature of the plaster at time of impregnation will influence the rate of penetration of the resin. By raising the temperature of the plaster it is possible at the time of resin application to obtain quicker penetration. From a practical point of view, as forms are removed from the drying oven they can be immediately treated with the resin impregnant without waiting for them to cool. In Fig. 6 is compared the rate of penetration of a 1 to 2-centipoise-viscosity furane resin at room temperature of 75–80 F, 135–140 F, and 180–185 F, respectively. The faster

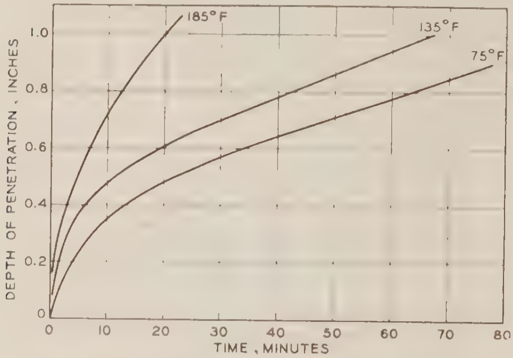

Fig. 6 Effect of Temperature on Penetration Into 100/60 Plaster

rate at 185 F is immediately observed. After a period of time the higher temperature has a retarding action inasmuch as the resin impregnant begins to cure or polymerize, resulting in a decrease in the rate. The plaster used in this experiment and others, unless otherwise indicated, is 100 solids to 60 parts of water.

(*d*) *Impurities on Surface.* There are several types of surface impurities which may influence the rate of penetration of the resin into the plaster. This becomes of practical importance when, for example, treating large surfaces and uniform appearance is necessary. One of the more common surface imperfections may arise from too liberal use of a parting agent in preparing plaster-of-Paris molds and patterns. Stearic acid is largely used for this purpose, as well as various soaps. These materials, if present, would restrict penetration of resin into the surface. However, if removed by light sanding or finishing operations upon the plaster surface, uniform pickup of resin is assured. In applying furane resin to develop resinified plasters, these surface irregularities can be readily observed, and a few spots where an excess amount of lubricant is present may be easily detected. A light touch with very fine sandpaper eliminates this problem.

Another type of surface imperfection finds origin in the mixing of the plaster. This imperfection is known to the plaster trade as a "hard spot." It occurs, generally, on large smooth surfaces, particularly near sharp corners or edges where rapid drying occurs. The hard spots are due simply to portions of plaster not too thoroughly mixed with the balance of the batch. If this is the case, such that a higher solids-to-water ratio is present, the plaster will tend to be harder, and porosity a little less. Once again, however, light sanding over this region will make the pickup of resin more uniform.

Amount of Resin Required

While best physical properties are obtained upon fully impregnated forms, it is also feasible to consider surface penetrations of $^1/_4$ to $^1/_2$ in. without necessarily going through several inches of plaster. For many contour blocks and tools this procedure has been followed for reasons of economy, and a good hard surface obtained without full treatment of the plaster. The amount applied may be controlled to some extent by the number of times the surface is gone over with a brush that has been saturated with liquid furane resin. Absorption of resin into the surface may be likened to the absorption of ink into a piece of blotting paper. Penetration takes place instantaneously upon the initial application, though at a slower rate after 10 or 15 min.

While vacuum and pressure vessels have been employed to obtain even faster and more thorough penetrations, the entire procedure is conducted at atmospheric pressure and temperature. When thin, low-viscosity resins are employed, the amount of build-up on the surface is negligible. However, when high-viscosity impregnating resins are employed, there is some build-up on the surface which must be wiped off before the part is cured. For greatest accuracy of dimensions this is quite important. Likewise, obscuring of surface details is avoided, and the plaster-of-Paris part is unaffected in detail though materially improved in substance.

For quick estimates of the amount of resin required to impregnate completely a plaster-of-Paris body, the article is weighed and approximately 40 per cent of its weight is prepared in impregnating resin. The actual amount required depends, however, on how thorough a penetration is desired, how well the plaster has been dried, and the other variable factors previously described.

Curing Impregnated Plaster

After the plaster has been properly impregnated, it is cured to develop optimum physical properties, which are consistently a 300 to 400 per cent improvement over the straight plaster-of-Paris structure. The curing procedure does not invite any special complications, simply requiring placing the form in an oven and slowly raising the temperature to about 145–150 F. The rate at which temperature is raised or the rate at which it is lowered must be conducted, of course, with concern for thermal expansion and shrinkage characteristics, in order to avoid the development of internal stresses at any position such as may mark the meeting of a thick or thin section of material.

While a very slight initial shrinkage may be observed, of the order of 0.05 per cent, the aftershrinkage of resinified plaster is negligible, and dimensions are held with utmost accuracy. This is a fortunate circumstance when considering the problems brought on by cast liquid resins, and the excessive shrinkages they have incurred, even with a judicious selection of fillers. Even after prolonged heating at 185 to 190 F, the dimensions are unchanged in resinified plaster.

At the same time these tests were made, the thermal-expansion coefficient was determined at 0.000022. This value approximates the expansion coefficient of aluminum alloys. This property has been utilized to advantage in providing for localized reinforcement of resinified plasters at positions of high stress concentration. The insertion of aluminum-alloy strips or corners, folded from sheet metal, gives the end product greater toughness where it is most needed. At first, this reinforcement may be viewed as an obvious procedure. However, the fact remains that cast plastics tooling has not been reinforced with metal strips because the plastics are too unstable and their aftershrinkage characteristics would create conditions for internal stresses and ultimate cracks in the casting. The complete absence of aftershrinkage in resinified plaster and the coincidence of identical thermal-expansion coefficients make a sound combination with aluminum alloy.

In Fig. 7 are curves showing the attainment of cure conditions at different temperatures. There are two stages to the cure: One in which the impregnating resin takes a set and turns to a dark color and the other in which the strength is progressively increased. The criterion proving most useful for this work is the

value of compressive strength, measured upon 1-in. cubes at frequent intervals during cure time. Impact-strength measurements run parallel to these. This set of results is also quite significant and explains in part why resinified plasters are much harder and more chip-resistant than unimpregnated materials. Of course, the values of impact strength indicated will not prevent the part from breaking, although there are various design principles which can be followed to give improved results. These design

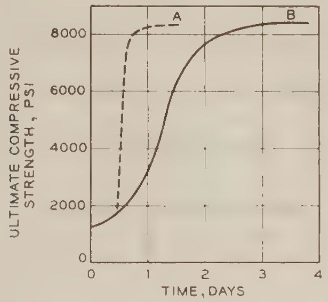

Fig. 7 Attainment of Cure Conditions at Different Temperatures

(*A*—Initial cure at 135 F; followed by 180–190 F. *B*—Cure at 130–140 F. Resin XP, impregnated plaster.)

principles place emphasis upon generously rounded corners and ample sections of materials to withstand load conditions. Impregnated plaster is not flexible, as the modulus of elasticity is quite high and hence when loaded, as in the application to a hydropress forming operation, the base should be perfectly flat and preferably mounted upon a true surface without any "bowing" before load is applied.

Additions of various fillers to the plaster composition to achieve even higher impact strength, will be discussed later in this paper. Comparative physical properties with various methods of reinforcement will also be outlined.

Temperature Stability

One of the most important gains registered in furane-resin-impregnated plasters over unimpregnated plaster is improvement

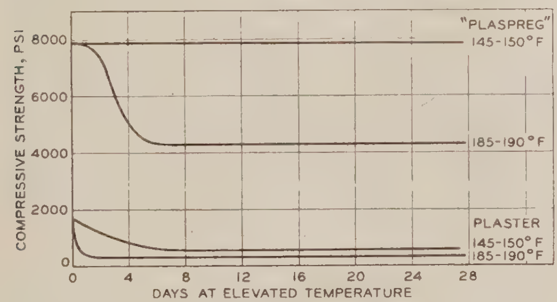

Fig. 8 Effect of Continuous High Temperature on "Plaspreg"

in temperature stability. This has already been indicated in part by pointing out that the material has a negligible aftershrinkage. The study, however, goes deeper than that and in two respects resinified plaster is far superior to ordinary plaster:

1 Maximum continuous operating temperature which it will withstand without suffering any permanent change in physical properties is in the neighborhood of 180 F. This value is appreciably more than straight plaster, the temperature limit of which is about 120 F, because above this it loses water of crystallization and strength. Likewise, the liquid casting resins have temperature limits of about 140 F or thereabouts, because at higher degrees they shrink too excessively.

2 The maximum temperature at which resin-impregnated plaster can operate continuously and still be stronger than the initial plaster, is about 250 F. Above 180 F the strength may fall off slowly to a certain level and then stop, although it will be stronger than the original plaster of Paris. This has been quite important to foundry work where plaster forms and core boxes have not been considered because of the rapidity with which they lose strength when raised in temperature.

The permanency of the physical properties of resin-impregnated plaster is indicated by the retention of its strength over a long period of time. Impregnated bodies prepared a few years ago test as strong as when they were first cured. To accelerate the aging of resinified plaster, a constant-temperature oven at 185–190 F is employed and compressive strength is observed after various intervals of time. Typical results are shown in Fig. 8. For purposes of comparison, straight plaster has also been tested. The loss in strength of plaster is due to loss of strength on removal of water of crystallization at high temperature. Retention of strength after exposure to high temperatures is indicative not only of the permanency of the materials, but also of operating temperature limits. As is the experience with most thermosetting plastics, short exposures to high temperatures in excess of continuously operating temperature limits will not be damaging.

Limiting factors in the temperature resistance of resinified plaster are (*a*) the loss of water of crystallization, and (*b*) the temperature limits of the furane-resin polymer, which lie about 250–300 F. Further improvements in temperature stability can be realized by substituting an inert filler such as powdered silica for part of the plaster of Paris. This can be accomplished without weakening the total cured structure.

Chemical Resistance

The chemical resistance of furane-resin-impregnated plaster is much more satisfactory than is that of untreated plaster. Water-absorption tests at room temperature are shown graphically over a long period of time in Fig. 9. Not only is water absorption greatly reduced because of the presence of furane resins to fill the voids, but furane-resin solids (100 per cent) show an A.S.T.M. water absorption of less than 0.05 per cent in 24 hr. In Fig. 9 sanded surfaces are compared with polished surfaces. For maximum weather protection, resinified-plaster toolings are polished or rubbed by hand with a wax suitable for finishing purposes. This will enable parts to be stored in the open, whereas in the past plaster has fallen apart or was seriously weakened by such exposures.

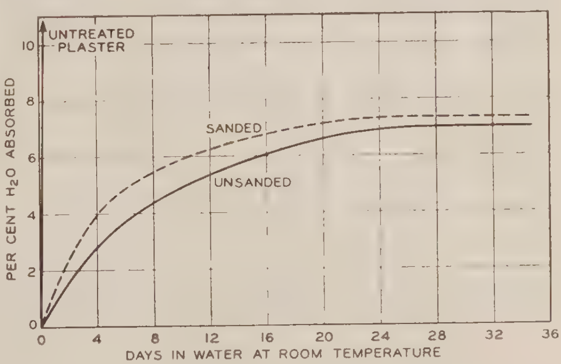

Fig. 9 Water Absorption of Furane-Resin-Impregnated Plaster

Strong acids and alkalies affect resinified plaster only slightly, whereas they cause untreated plaster to decompose. As evidence of their good chemical resistance, various resinified plaster parts have been electroplated after first applying a thin high-polymer film, followed by stannous chloride and then a silvering solution.

For general chemical and temperature resistance, the parts shown in Fig. 1 illustrate the utility of resinified plasters as tooling for low-pressure laminating. The service life of plaster parts is increased quite appreciably owing to the much higher strength. In addition, forms are rendered quite impervious to various polyester resins which must be cured at temperatures as high as 300 F. In the illustration shown, XRS-16631 served as the laminating resin for glass cloth which was cured under vacuum pressure. A vinyl-resin lacquer provides a good release agent for the parts in question.

In another important laminating problem for complicated ducts for aircraft, a removable core of plaster was necessary for each operation, requiring breakage to take the piece out. The plaster cores did not possess sufficient strength to stand up under the laminating operation, although the problem was solved by subsequent treatment with furane resins. By impregnation they acquired sufficient strength to withstand the pressure of the laminating operation. A number of air ducts for fighter airplanes was produced in this manner.

Physical Properties of Resinified Plasters

The discussion so far has centered upon furane-resin-impregnated plaster of Paris. Results are even more noteworthy when various fibrous reinforcements are included in the plaster. Long fibers such as hemp or sisal fiber have been known to strengthen plaster patterns and tooling. They likewise add considerably to the strength of the resin-impregnated plasters. Glass flock, in particular, is effective in forming a strong and tough body which can take much mechanical abuse, and its introduction into plaster before impregnation is an important step.

In Table 1 some of the physical properties of furane-resin-impregnated plaster are noted.

TABLE 1 PHYSICAL PROPERTIES OF RESINIFIED PLASTERS

Material Property:	Unimpregnated plaster	Resinified plaster no filler
Specific gravity	1.15 (dry)	1.65
Compressive strength, psi	1500-1800	7500–8000
Flexural strength, psi	400–500	2500–3500
Impact strength, Charpy unnotched, ft-lb per in	0.15–0.25	0.8–1
Water absorption, per cent in 24 hr.	35	2

All of the physical tests followed the applicable A.S.T.M. standards with the exception of compressive-strength tests which employed 1-in. cubes instead of $^{1}/_{2}$-in. cubes. The unimpregnated plaster and resinified plaster contained 100 parts of solids and 60 of water at the time of preparation. Materials were dried at room temperature for 3 or 4 days and then overnight at 140 F, to remove all traces of free water. They were impregnated with low-viscosity resins and cured about 48 hours at 150 F.

In conclusion, it appears desirable to sum up some of the advantages and disadvantages of resinified plaster. It represents the application of a new thermosetting resin, the 100 per cent furane resin, into the plastics picture and the development of a new and important outlet for plastics in the ceramics industry. A new, low-cost tooling material with negligible shrinkage has made its appearance.

The advantages of resinified plaster are as follows:

Low Cost. Resinified plaster is prepared from a low-cost base material, plaster of Paris, which costs about 1 cent per lb. When fully impregnated, the cost of the treated plaster is appreciably less than that of a thermosetting cast-phenolic resin.

Ease of Fabrication. From the standpoint of ease of fabrication and workability, plaster has few equals. In the resin-impregnation process, plaster of Paris or hydrocal tools, forms, or patterns are prepared in their usual manner and set at room temperature. A very stable structure is prepared with simple tools. Upon resinifying and hardening, this form is rendered permanent.

Physical, Chemical, and Thermal Properties. These properties have been vastly improved over the straight plaster, placing fully resin-impregnated plaster in the same class as liquid-cast resins, with the important advantage of zero aftershrinkage.

Applications. No form is too small or section too big to be handled by this process. Resin penetrations up to several inches have been accomplished at room temperature and atmospheric pressure. Wherever plaster is used resin impregnation may be applied to convert these articles to permanent, stronger materials of construction.

The disadvantages of resinified plaster are as follows:

Drying. For best results the plaster forms should be thoroughly dried out. This necessitates an additional step in preparing plaster and lengthens the time lapse before the tool can be used.

Breaking. While all properties are improved some 300–400 per cent, furane-resin-impregnated forms can still be broken, and one should not expect the treatment to convert plaster into a rubberlike, unbreakable material.

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The Control of Fouling Organisms in Fresh- and Salt-Water Circuits

By JOHN G. DOBSON,[1] SHORT HILLS, N. J.

The difficulties encountered by industrial water users due to fouling organisms are discussed. To control such fouling, a knowledge of the organism's life cycle and reactions to stimulus is necessary. Typical mollusks, Bryozoa, sponges, barnacles, and tunicates are considered. Their control can be effected by heat, change of salinity, change of oxygen content, increased velocity, acids, antifouling paints, screening and poisoning with a number of different poisons. It is concluded that chlorination is the most effective and economical of these methods justifying itself in most plants by improved heat transfer alone. Factors to be considered in designing water systems that may foul are also outlined.

INTRODUCTION

Examples of Problems

THE growth of marine macroorganisms, both plant and animal, has been a continuing source of difficulty to power stations, oil refineries, and other users of industrial water. Although much has been written concerning accumulation of microorganisms of the capsulated and slime-forming types on heat-exchanger surfaces with attendant reduction of heat-transfer efficiency (1, 2, 3),[2] little has been published on the growth and control of the macroorganism in closed water circuits. This paper deals with those fouling organisms visible to the naked eye which cause industrial difficulties.

One of the more important, although perhaps the least obvious of the difficulties resulting from fouling by the larger aquatic organisms, is the reduction in the carrying capacity of a pipe line by reduction of the Hazen and Williams coefficient and also by actual reduction of the pipe-line diameter. In addition to some fresh-water grasses and mats of fresh-water algae in the sunlit portions of the water conduits, the most common organisms responsible for this blocking of fresh water lines are the Bryozoa, notably *Pectinatella magnifica*, and the sponges. In salt-water or brackish-water lines, the various hydroids and Bryozoa, as well as Mollusca and Tunicates, are responsible for such reduction.

Sponges, both of the calcareous and of the siliceous or fiber types have been responsible for occasional fouling of pipe lines. In one southwestern city the Hazen and Williams coefficient has dropped from 145 to 94.5 on a 60-in. iron pipe line due to sponge accumulations. Occasionally large cast-iron pipe lines have become almost completely blocked with various Mollusca, notably *Mytilus edulis* and *Pecten latiauratus* in salt-water circuits and various *Dreisseniidae* in fresh-water circuits. As much as 266 tons of shells have been removed yearly from the circulating tunnels of one New England power station. Another station has an accumulation of dead shells 3 ft to 6 ft deep in a tunnel with a 6 × 11-ft cross section and 400 ft long. Figs. 1 to 4 inclusive, illustrate conditions in a tunnel on October 1, 1945. The tunnel walls and floors had been made broom clean by scraping 4 months before. Where salt water is used in fire service, these same organisms have not only reduced carrying capacity of the line, but shells loosened by sudden rush of water due to fire demands have blocked valves, hydrants or other irregularities, and have completely shut off water supplies with disastrous results.

Similar difficulties have been encountered where salt water is used for displacement of gasoline or fuel oil from underground storage.

Sea-going vessels in harbor waters of temperate climate and most tropical waters have had fire and flushing lines clog with marine growths.

Growths, particularly of shelled organisms, continue to enlarge until their exposed area becomes so great that they are torn from their moorings on the pipe line and are swept into screens, tube sheets, or pumps. In one Gulf Coast refinery, Bryozoa so blocked the tube sheet of divided water-box condensers that the pressure from circulating-water pumps burst through the dividing wall and forced whole sections of the refinery "off stream."

When the lines or tunnels serving shell-and-tube or surface condensers become fouled with these organisms, additional difficulties are encountered. After the organisms break off or die, the shells float up with the circulating water. Many seaboard stations and some stations using river and lake water have had to shut down entire turbogenerator units two or three times a day to permit removal of shells that are blanketing the tube sheets. Some shells in these cases are sure to enter the tubes and cause high impingement velocities with attendant erosion and reduction in tube life (4).

The presence of macroorganisms, as well as that of microorganisms on the surface of metals, cause differential cells and attendant corrosion (5), particularly of steel materials. Attacks on nonmetallic structural materials also occur (6).

Mechanics of Fouling

The exact mechanics of fouling have been studied to only a limited extent. Most of the work has been done on smooth metal or glass plates simulating conditions on a ship's bottom. The first organisms to attach to a newly immersed plate are the unicellular capsulated bacteria, Fig. 5. Whether these organisms attach first merely because of their rapidity of growth, or whether they form a necessary footing for other larger organisms is not clear.

Some "conditioning" of a surface has been thought necessary before fouling begins, even in the case of natural rock (7). This "conditioning" apparently consists of a roughening of the surface and possibly the elimination of the material being leached from the surface.

After that period of "conditioning," first one and later other organisms may become predominant (8). Eventually stable conditions result (9).

History of Control

Early recorded experimental work (10) covering control of

[1] Industrial Division, Wallace & Tiernan. Jun. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Fig. 1 Circulating-Water-Intake Bar Rack
(The rack had been completely cleaned of all growth 2 months before this view was taken.)

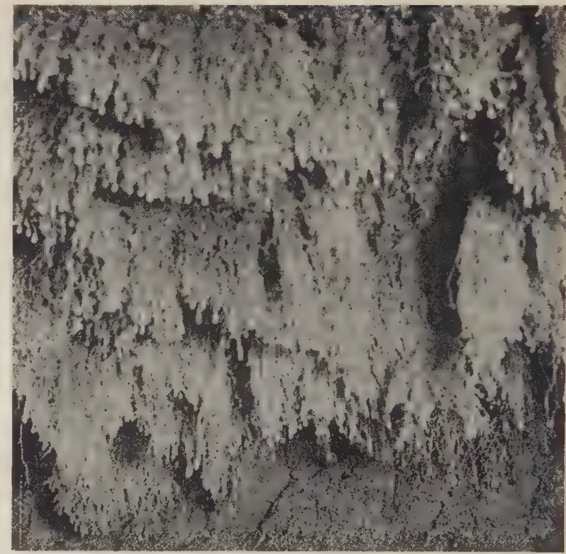

Fig. 2 Ceiling of Concrete Tunnel
(The luxuriant growth of *Tubularia crocea* is 7 in. long and some of it has already dropped off and decomposed on the tunnel floor in the 4 months since the last cleaning.)

Fig. 3 Pump-Suction Bell in Same Tunnel as Fig. 2, With Large Growth of Mytilus
(These mussels are already large enough to blanket tube sheets if they break off in mats or to lodge in a tube if they break off as individuals.)

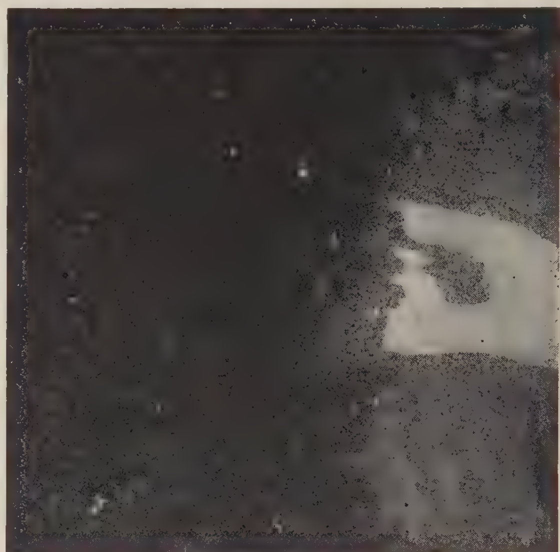

Fig. 4 Close-Up of the Same Pump Bell as Fig. 3
(The area at the right has had the mat of mussels torn off to show depth of growth. The hand merely indicates dimension.)

fouling organisms in closed water circuits was begun when the consulting-engineering firm of Kennedy and Donkin was commissioned early in 1919, by the Electricity Committee of the Edinburgh Town Council to make recommendations regarding the control of marine growths in the circulating tunnel of the Portobello Generation Station, which was then in process of design.

The investigators studied various control methods which had been previously attempted. One private firm at the Leith Docks had attempted to control mussels by annual flushings with high concentrations of sulphuric acid. This control was partly successful but severe corrosion resulted. Others had attempted annual or semiannual cleanings of inlet pipes by use of tight-fitting balls and other cleaning implements being dragged through tunnels; but several pipe lines were severely damaged. At Portsmouth an electrically insulated pipe line had been freed to some extent of mussels by using electrical discharges between the pipe and a central electrode.

Screening to prevent the entrance of small organisms, the use of various antifouling paints, and the use of various mechanical

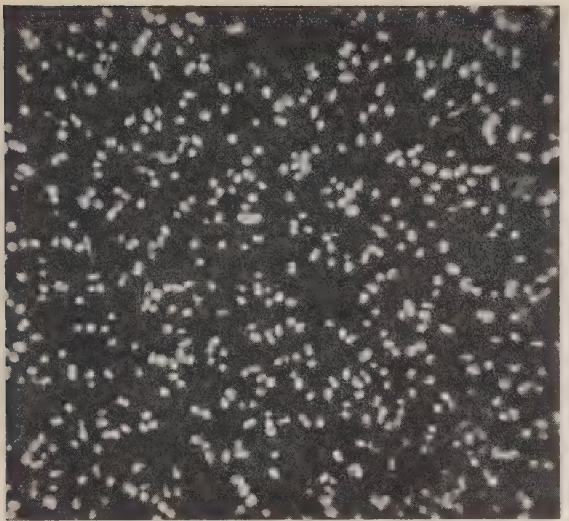

FIG. 5 SMALL GRAM NEGATIVE ENCAPSULATED COCCOBACILLI TYPICAL OF SLIMING MICROORGANISMS IN SALT WATER; ×930

cleaning methods, were considered and rejected as impractical. Chemical means were then attempted.

It was found that sulphuric acid in concentrations of 1 part in 10,000 was unsuccessful; that the larger mussels, after immersion in fresh water even for as long as 120 hr, were unaffected.

Experiments were then tried with heated water. Mussels were removed from sea water at 69 F and placed in salt-water tanks which were gradually heated. Selective killing occurred above 100 F during immersion periods of 25 to 50 min. All mussels failed to revive when returned to normal sea water, after immersion at 106 F for 22 to 63 min.

Based on these experiments, it was recommended in 1921 that the Westbank Station of Portobello be built so as to permit the reversal of flow in the condenser cooling waters, and that that reversal be carried forward at about 4-week intervals. It was also recommended that during this reversal period, the circulating water flow be throttled so as to increase the water temperature on the exit of the condenser to at least 110 F.

Various other attempts were made in the succeeding decade to obtain satisfactory chemical methods of control. These were summarized by the British Electrical and Allied Industries Research Association (11) in 1929. Apparently based on abstracts, the author reported successful control of marine growths by use of chlorine. The major paper upon which the author relied (12) had merely suggested that since chlorination would kill slime-forming and other microorganisms, and since these organisms were the food for mussels, chlorination might control *Mytilus edulis*. No experimental work had been done and since we now know that *Mytilus edulis* does not require living organisms for food, the hypothesis was unsatisfactory.

A German professor (13) has been widely quoted as reporting successful control of the fresh-water mussel *Dreisseniidae polymorpha* in the flumes of a powerhouse on the Glambacksee near Berlin. His report in 1921 indicated that the larvae of this organism died in the laboratory under chlorine dosage but apparently no unchlorinated control was used. No plant-scale work had been carried out. Some confirming laboratory work was done in Germany in 1929 (14). Similar discussions of the use of chlorine were later published (15, 16).

The first work in this field in the United States was started under the direction of W. J. O'Connell. In 1924 experimental work on intermittent treatment of condenser cooling water for the control of slime-forming microorganisms in cooling-water systems had begun in co-operation with Commonwealth Edison Company of Chicago, and the first plant-scale experiments of such intermittent treatment were tried at the Kearny Power Station of the Public Service Electric and Gas Company of New Jersey. The extension of this work to the control of marine growths was started in 1929, at the Northport Station of Long Island Lighting Company with the co-operation of L. H. Curley. The work there and at Corpus Christi indicated that it was possible, by intermittent chlorination, to eliminate completely the growths of the grasses and higher algae, and to reduce effectively the growth of bivalve forms. However, at that time eliminations of the order of 99 per cent were not attempted.

TYPICAL ORGANISMS

The life history, method of reproduction, food, responses to various unfavorable environments, and living habits of fouling organisms must be thoroughly understood before any intelligent plan can be suggested for their control. It is impossible to cover the wealth of material that is available about all the fouling organisms, but representative types can be discussed. However, before commercial installations are started, details regarding the organisms responsible for the particular fouling should be known.

Luckily for the engineer interested in controlling such growths, very few of the approximately 822,000 invertebrate animals which have so far been described by the biologists, cause fouling difficulties.

THE MOLLUSCA

Of the fouling organisms, the members of the phylum *Mollusca*, which includes the snails and other single-shelled *Gastropoda;* and the clams, oysters, mussels, and other bivalve or two-shelled *Lamellibranchiata* are the most troublesome. *Mytilus edulis*, or the edible or black mussel, is probably the most common fouling mollusk throughout the world. This organism has been responsible for most of the fouling difficulties in the British Isles and along the North Atlantic coast of the United States. The closely related *Mytilus californianus* has been responsible for

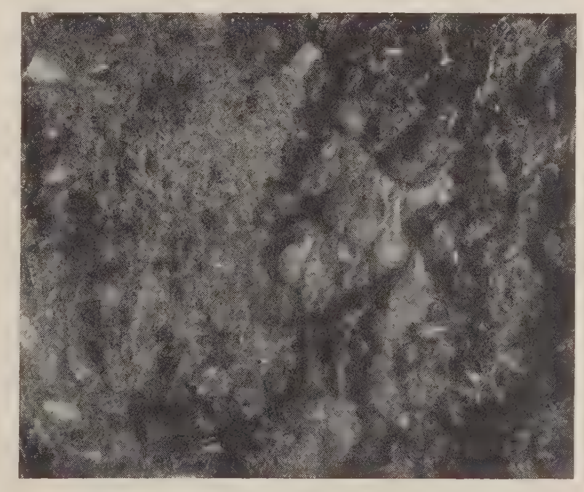

FIG. 6 HYDROIDS GROWING IN LOW-VELOCITY AREA OF A TUNNEL (Young mussels appear at right; dead mussel shells in upper left-hand corner.)

much of the fouling difficulties which have occurred along the western coast of the United States.

The black mussel (17) is familiar to all who have frequented any rocky coast of the northeastern United States. The shell is wedge-shaped, pointed in front and round behind; black or dark brown outside with indistinct circumferential growth rings around the shell. The interior of the shell is pearly with violet margins. Shells vary in length to a maximum of about 4 in. Its life history is typical of many other of the fouling invertebrates (18). Other *Mytilus* have nearly identical life cycles. The sexes are separate, but one individual may be male one year and female the next. The eggs and sperm from near-by individuals are discharged almost simultaneously and the eggs are fertilized in the open water. The sperm has only limited range of locomotion (19). An average female discharges from 5,000,000 to 12,000,000 eggs although a female $3^1/_2$ in. long may release as many as 25,000,000 eggs. Cell division begins immediately after fertilization and is completed within 20 min. The first cilia begin developing at the end of 4 hr (based on 68 F sea-water temperature), and at the end of 5 hr the embryo is completely free-swimming.

Development continues and at the end of 24 hr the organism is a very active swimmer; and by the end of 40 hr the complete digestive tract and the glands for forming the shell are becoming evident. At the end of 44 hr the organism has attached to some stable hard material such as rock, concrete, glass, rubber, or similar solid support, and has begun to develop the first single shell. At the end of 69 hr the elementary single shell or prodissoconch has completely enclosed the fleshy portions. The organism is still capable of rapid swimming and is able to crawl on solid surfaces, vertical walls, and the tops of horizontal tunnels. Some time later the true bivalve shell or dissoconch develops. Contrary to popular belief, the organism, which is now attached by long clear threads, known as byssus threads, (20) is still capable of motion, and if conditions of poor feeding, inadequate oxygen supply, or mechanical or chemical irritation develop, the organism will break the byssus threads from their point of attachment and crawl by a "foot" which projects out of the shell near the hinge. Such "walking" is similar to the locomotion of the familiar terrestrial snail. The mussel can also move by secreting new byssus threads and breaking off older ones.

The greatest growth rate occurs when the organism is completely submerged at all times, although under natural conditions predatory fauna may reduce the number of organisms below the low-water line (21).

Under normal conditions the adult *Mytilus edulis* will leave the shell open about 15 to 20 deg, the cilia on the mantle will vibrate rapidly and water containing food will be forced through the digestive tract. This flow of water amounts to about 10–20 gal per day in a specimen 2 in. long (22, 23) and seems to be constant, provided the organism is not irritated. This water circulation is called "drinking." The rate of digestion, on the other hand, seems to vary with metabolism demands. The shell remains open about three quarters of the total time in a temperature range of 31 to 77 F (24) without any variation with temperature.

As the water is circulated through the digestive tract any suspended solid matter adheres to the mucus surfaces. Digestive juices then digest such material as is usable for food, and this is assimilated. The balance is excreted as fecal pellets. By this process, mussels are capable of clearing cloudy and turbid water.

The exact nature of the food requirements is still little known. It was originally believed that living microorganisms mostly diatoms and algae, Fig. 7, were required, and they undoubtedly make up a large part of the natural food (25, 26). Robinson (12) based his theory of killing of mussels on this postulate. However, more recent work has indicated that while organic matter is necessary as food, decaying animal and vegetable material is as efficient as living organisms (27, 28, 29). It is known that domestic-sewage pollution, provided continuous anaerobic conditions are not set up, encourages growth.

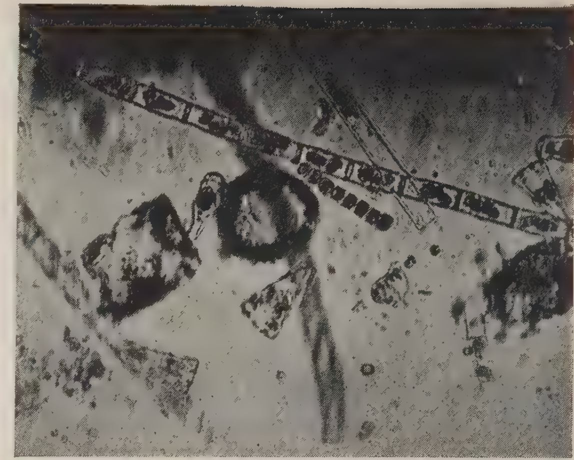

Fig. 7 Algae and Diatoms Which Form Basic Natural Food of Mussels; ×460

Each species has its own preference as to conditions of light. The author has repeatedly placed *Mytilus edulis* and *Modiolus modiolus* in equal quantities in adjoining and connected cells, one light and the other dark. Marked migration of *Mytilus edulis* toward the dark and *Modiolus* toward the light is always noticed. *Mytilus* grows three times as fast in total darkness as in sunlight (30), although a weak light may be still more favorable. Optimum growth occurs at 50 to 68 F, and is seriously inhibited by heating for even a few minutes a day (31).

If any unfavorable conditions develop or anything disturbs the organism, the shell will close within a fraction of a second and only after an extended period will the mussel open the shell slightly and "drink" very cautiously. This ability to close its shell tightly makes all adult *Lamellibranchiata* very difficult to control. All mussels have a cushion pad which forces the shell open as soon as the adductor muscle is out or relaxed. Therefore a gaping shell can be considered as a sign of damage to the organism.

Pecten (scallops) (32) are not usually responsible for fouling, but *Pectens latiauratus*, which attaches with byssus thread reduced an 18-in-diam cast-iron line serving a South Carolina power plant to 6 in. in about 12 months.

The various fresh-water mussels have a life history similar to *Mytilus edulis*. However, important differences exist. Instead of releasing single-celled eggs, the adult female permits the eggs to be fertilized while still within the mantle, and free-swimming "glochidia" about 0.014 in. diam are released (33). A parasitic period on host fish is necessary before metamorphosis into shelled forms can take place. *Dreisseniidae* is limited in range to fresh or brackish water of less than 1 per cent salt content (34).

Dreisseniidae is similar in appearance to *Mytilus edulis* but lacks the pearly shell interior. The organism is common in Germany, having been responsible for actually shutting down a power station on the Glambacksee with a 12-in-thick growth on tunnel walls (13). The American oyster (*Ostrea virginica*) is a common Mollusca responsible for severe fouling of power inlets on our South Atlantic and Gulf coasts, where it is easily recognized by its rough shell and white interior. Its life history, together with that of its European cousin (*Ostrea edulis*) has been carefully

studied because of its importance in the culture of oysters for food. Its life history is similar to the edible mussel (35). However, its spats are apparently considerably more delicate. The period of pelagic or larvae life varies from 13 to 17 days (36), depending upon temperature. The spat prefers a dark place to settle, and metamorphosis can be delayed by failure to find a suitable resting place (37). Spawning can be stimulated out of season by raising the water temperatures from 82 to 86 F (38). It can be induced by the presence of eggs or sperm in the water, not only of its own specie but those of other mollusks, or by various chemicals (39). The season of normal spawning varies with location, being from April to June in some eastern United States waters and from July to August in others (40). In Denmark it occurs from late May to mid July, while in Holland spawning occurs in September.

The daily rate of growth varies with age of the organism and water temperature, but does not vary with the actual size of the organism (41).

The Sponges

The sponges (*Porifera*) are the simplest of the fouling macro-organisms. A "sponge," as normally seen with the naked eye, is a colony of individuals with the large number of "pores" on the outer surface. Through these pores, water containing food material is circulated by flagella, and as it passes through the cloacal cavity, exposed cells absorb food material and transmit it to near-by cells. No truly "specialized" organs exist.

Reproduction is by three separate and distinct methods, i.e., by budding, by the formation of gemmules, and by sexual methods.

Budding takes place within the organisms and results in a single "sponge" increasing in size or even dividing, but does not permit the formation of new groups at a distance from the previous place of attachment.

The formation of gemmules is similar to a formation of "spores" in microorganisms. A living cell migrates to the center of the colony and is covered with a capsule secreted by the adjoining cells. When unfavorable conditions of temperature (in fresh water lakes) or dryness (in tropical lakes) occur, these dormant forms resist those unfavorable conditions and upon the return of conditions conducive to development, burst the capsule and begin development of a new sponge.

Sexual reproduction is in the main responsible for the development of new colonies. "Some sponges are hermaphroditic, others are unisexual. No special sexual organs are present, the ova and sperm developing from ... the inner cells .... The ... egg develops into a ciliated two-layered larva which swims actively about in the water, which finally attaches itself and, after metamorphosis, develops into the adult animal" (42).

It is during the larva stage that the sponge is most vulnerable to attack by poisons. They are relatively sensitive to changes in pH (43).

The Bryozoa

The Bryozoa, or Polyzoa as they are called in England, are minute colonial animals that are often confused with seaweed and sponges both by the layman and the early naturalists. There are about 3000 marine species and 35 fresh-water species. Upon first examination, they may appear as red or white encrusting patches, as groups of dead sticklike organisms, brown or purple in color, projecting from a rock or as branched sticks with plumed ends (*Plumatella*). The familiar jelly balls in fresh-water lakes (*Pectinatella*) are also Bryozoa.

Each colony or "zoarium" increases in size by budding. Sexual reproduction also takes place; the eggs and sperm being released into an inner cavity and developing to a larva stage there. The larva are released either through special openings or by the death and disintegration of the parent. Other specie (notably the fresh water *Plylactoloemata*) are capable of forming "statoblasts" to resist unfavorable conditions similar to those developed by sponges.

Statoblasts of *Pectinatella magnifica* can endure much drying but not absolute desiccation. They are not harmed by brief exposure to temperatures as low as 31 F and as high as 130 F; while the adult form or "polypids" survive only in a range of 50 to 104 F (44). The statoblasts of fresh-water Bryozoa usually germinate at 62 to 66 F. These statoblasts not only provide a means of surviving unfavorable conditions, but also provide a ready form for transportation by wind and birds.

Grave (45) has reported on the development of *Bugula flabellata*, one of the common Bryozoa on the New England coast. The breeding season at Woods Hole, Mass., extends from June 10 to November 15. The young are expelled from the colony as swimming embryos. After a free-swimming period of 4 to 6 hr, the larva attaches, profound metamorphosis takes place involving the loss of some larva organs, and the colony begins development. At the end of a week the colony has increased by budding to 8 or 10 individuals, at the end of 2 weeks to 100 individuals, and by the end of a month the colony is half grown and has reached sexual maturity. Colonies $1^1/_2$ to $1^3/_4$ in. diam are able to hibernate successfully and resume growth in early May. Even more rapid growth occurs in South Atlantic and Gulf Coast waters where the time from first infection to serious difficulty in industrial water circuits may be only a matter of weeks. Colonies of *Bugula neritina* grow as long as 4 in. at Beaufort, N. C. (46). Many fresh-water Bryozoa flourish in dark as well as light, and cause serious fouling in water mains (47).

The Barnacles

The barnacles are probably the best known of the fouling organisms, since they have been responsible for most of the serious fouling of ships' bottoms. Because of their hard shell with several parts and because they seemed to "drink" like the mollusks, they were so classified by the early zoologist. J. V. Thompson in 1830 was the first to show definitely that they were Crustacea, closely related to the crayfish, lobster, and soft-shelled crab. Darwin (48) in 1850, made an extensive study of the group (49).

Moore (50) on the Isle of Man and Grave at Woods Hole, Mass. (51), among others, have made extensive studies of the barnacles. They are mostly hermaphroditic but in a few cases are unisexual. Some species have a male form which is very tiny and lives parasitically within the shell of the female.

The eggs (52) are held within the mantle cavity and the larvae are released as free-swimming forms after hatching. The "nauplei" changes form slightly with each moult into forms known as "meta-nauplei," and then metamorphoses into an entirely different type of organism known as the cypris larvae or the "cyprid." This later stage lasts from 7 days for *B. amphitri* to two weeks for *B. balanoides*. During this stage the cyprid swims with a sudden backward motion of the appendages. The barnacles then settle down on a suitable place of attachment and begin to undergo a second metamorphosis and form a shell. Actually the organisms will make several attempts to find a suitable place to attach and if the one selected is not satisfactory the barnacles will break away even after they have become partially attached. After finally reaching a suitable place, some small motion in an area of about $^1/_2$ in. is still possible (53).

"In immature barnacles tissue growth continues throughout the year but is most rapid in the spring and autumn. In mature barnacles there is a sharp drop in growth rate during the summer

and a slight one during winter associated with the development of the larva." In exposed places, below mean-tide level, the *Chthamalus* reach sexual maturity in the first year and die in their third year. In less exposed places, the organisms reach sexual maturity in their second year and die in their fifth year (54).

The spat will choose its resting place partly on the basis of light intensity, apparently making a choice of optimum light, shying away from too bright and too dark places. The organism apparently can only distinguish light from dark areas. Light has also some control on growth (55, 56), and the adult organisms will close their shells when a shadow passes over them (57).

Barnacles also react to changes in potassium (58), alcohol (59), carbon dioxide (60), calcium (61), salt-water dilution (62), strong electrolytes, urea, glucose, and glycerol (63).

The Tunicates

The highest organisms responsible for fouling in industrial plants are the tunicates. They represent a "subphylum" of the phylum *Chordata* to which also belong the vertebrates including fish, mammals, and man. They derive the name from their thick, hard, cellulose coating and are one of the few animals capable of synthesizing cellulose.

Of these, one of the most common is *Molgula manhattensis*. It consists of a brown sac with two distinct openings at the end of noticeable tubes. While the individual organisms are only about 1 in. in diameter, they form large colonies which break off in mats from tunnel walls and often blanket tube sheets. It is also not uncommon, for example in the Kill Van Kull off Staten Island, for these mats to be broken off by wave action and block inlet screens and sometimes even bar racks. Related specimens with nearly clear tunics have been clogging powerhouse screens in Stamford, Conn., harbor. They are commonly called "squirt balls" because of the habit of ejecting water from their body cavity sometime after they have been lifted from water.

The life history of the *Molgula manhattensis* has been carefully studied by Berrell (64). Eggs of about 0.006 in. diam are released in the water and fertilized there. The time taken from fertilization to hatching varies with temperature, being approximately 120 hr at 46 F, and 40 hr at 62 F. The hatched egg yields a "polliwog" which is free-swimming with an active tail. The tunicates have a heart, elementary circulatory system, digestive tract, liver, a nerve cord, and a gangula which is an early form of the brain. After a period of 160 to 450 hr, which period depends on CO_2 content of water, salinity, pH, and temperature, the "polliwog" attaches to rock, inlet tunnel or tube sheet, or other structure, with three sticky tentacles and begins a retrogressive metamorphosis. Powers of locomotion are lost, the existing digestive tract disappears, and a new one begins to form on the remaining portion of the animal.

Even this high animal can reproduce by "budding" and thus form extended colonies. In order to attain maximum efficiency in control of tunicates, control methods must be timed to attack the organism during the early active larvae stage or during early metamorphosis, before the larvae alimentary canal, through which both food and oxygen are absorbed, is eliminated from use.

Al arge number of standard zoological texts, as well as specialized books and articles, give further details on the life history of the fouling and related organisms. Pratt (42) is a good representative general text for identification of invertebrates; while Ward and Whipple (65) give more detailed treatment covering the fresh-water organism.

A good recent review of the reactions of invertebrates to stimuli has been published by Warden, Jenkins, and Warner (66).

CONTROL METHODS

Numerous methods have been suggested for preventing the settling of larva forms and the killing of adult forms of fouling organisms. Among them are the following:

1 Heating the water.
2 Increasing or decreasing salinity of the water.
3 Creating anaerobic conditions in the water.
4 Increasing water velocity in tunnels.
5 Poisoning with acids.
6 Antifouling paints.
7 Screening the tunnel entrance.
8 Poisoning with miscellaneous poisons.
9 Poisoning with chlorine.

Any theory of killing or method of control is only as valuable as it can be proved in actual practice at plant scale. Heat, increased velocity, surface treatment, acid, and chlorine are the only methods that have been put into practice on a large scale.

In selecting a particular method of operation, it is well to bear in mind the factors to be considered. While there will be differences of opinion as to the relative importance of these various factors, most designers will wish to consider them all.

1 The method should not materially increase the operating cost of the plant. Too often in evaluating the various methods involved, designers are likely to decide against one particular method, because of a direct cost of operation and decide upon another that disproportionately increases pumpage costs or lowers the general efficiency of the plant.

2 The method should have lowest possible first cost consistent with low operating cost. Particular care should be taken in evaluating cost of special equipment for control of marine growths compared with increased cost of general plant equipment to make possible other methods of control.

3 The method should assure that no organism can grow to a size that will be detrimental to plant equipment, and a method should be available for recovering control of the growth if, for any reason, the treatment method must be interrupted.

4 The method should permit operation of the plant at full capacity without interruption.

Control by Heat

The earliest method of control used on plant scale was heating circulating water. One power station throttled the circulating-water flow until the desired heat balance was obtained. The amount of heating required is dependent upon the normal average temperature and normal range of temperatures of the water in which the organism has grown.

Without reporting initial temperature of the water from which the organisms were removed, Henderson (67) found that by raising water temperature gradually and removing specimens to cooler water for recovery, the lethal temperatures for various mollusks were *Leda tenuisulcata*, 88.7 F; *Cardium boreulis*, Conrad, 88.9 F; *Modiolaria discors*, Beck, 89.4; *Saxicava rugosa*, Lamaich, 91.0 F; *Crenella glandula*, Atoms, 91.0 F; *Cardium pinnulatum*, Conrad, 91.8 F; *Astarte undata*, Gould, 92.4 F; *Pandora trilineata*, Say, 92.4 F; *Modiolaria nigra*, Loven, 94.2 F; *Yoldia sapotilla*, Simpson, 94.7 F; *Zirpoea crispate*, Gray, 92.4 F; *Mactra solidissoma*, Chemnitz, 98.6 F; *Mya arenaria*, Linne, 105 F; *Mytilus edulis*, Linne, 105.4 F; *Nacoma fusca*, Atoms, 108.2 F; *Venus mercenaria*, Linne, 113.3 F; and *Ostrea virginica*, Lister, 119.3 F.

It is interesting to note that *Tubularia* in water above 68 or 70 F, remains dormant, although it is able to recover when held for 3 days at 65 F (46, 68). In Scotland successful control of

Mytilus edulis on plant scale has been obtained by heating the circulating water to 120 F.

The cost of operating such a system can be extremely high. Assuming that a power station with initial steam conditions of 400 psi and 500 F, and capacity of 25,000 kw is fully loaded, and assuming that the circulating-water temperature must be raised from 65 to 120 F, the fuel cost alone due to the corresponding loss in vacuum for a 2-hr period will amount to approximately 40 tons or, computed at $2 a ton, $80. The load capacity of the plant is, of course, at the same time reduced.

In addition, increasing the temperature of the condenser may cause leaking ferrules and other leaks due to uneven expansion of tubes and tube sheets. Any operator who has experienced a failure of condenser water supply will appreciate this problem.

The cost of the initial installation of such a heating system for a steam power plant will also be higher than other methods of marine-growth control. This can be readily appreciated when it is realized that the entire circulating-water system, and particularly the pumps must be designed for reverse flow. For plants not having waste heat available, the cost of such heating would be prohibitive.

Control by Changes of Salinity

Fresh water has been used as a means of eliminating marine growth from fouled surfaces since ancient times. Early mariners ran their vessels up fresh-water streams to loosen the ship's fouling load.

Most marine organisms are adversely affected by reductions of salinity, but the absolute value that will support life varies with the species, the salinity of environment from which they are removed (69), and the rate at which the change in salinity occurs. Because of the number of variables, it is difficult to correlate various experimental results. Sudden reductions to 30 per cent of normal salinity have been found to be fatal to *Mytilus* (70). *Ostrea virginana* will not fatten below 20 per cent salinity (71). The critical salinity for maintaining life and for growth varies from individual to individual within a given specie (72).

Most fresh-water invertebrates die in salt concentrations above 5 grams per liter NaCl (73) but *Physa heterostropha* have been known to live actively in 25 per cent sea water if the concentration is slowly increased to that figure (74). Contrary to the average tendency, the oyster drill grows to a larger size in brackish water than in salt water (75). Increase or decrease of salinity from that of normal sea water acts as an anesthetic, causes loss of sense of balance in the motile organisms and paralysis of the motor systems (76, 77).

Practical applications of this method for control of growths in industrial-plant water circuits are difficult, unless brackish streams varying in salinity with rainfall are readily available. Unfortunately even for plants with such streams available the problem is not fully solved for all the fouling organisms are not as sensitive as those just mentioned, and some will thrive under conditions of widely varying salinity.

Plants on such brackish streams are faced with another problem which is acute along the St. Johns and Hillsboro Rivers in Florida. Spring rains almost invariably change these streams from brackish to fresh water. The mussels and barnacles which have set in the circulating-water tunnels are killed, and the shells and debris of the dead organisms float up and clog pipe lines and blanket tube sheets. No operator having tube sheets badly clogged with shells will say that natural variation of salinity is a cure for his marine-growth problems!

Control by Creating Anaerobic Conditions in Water

The reduction of oxygen content of the water below certain critical levels will eliminate many organisms. The exact value of dissolved oxygen which will support life varies with species. All fish and most of the invertebrates require free dissolved oxygen. At least one of the contributing causes of death of oysters in water heavily polluted with paper-mill waste has been the reduction of oxygen content (78, 79).

However, no practical means are available to remove oxygen from industrial-water circuits. The gross pollution which has been so common in some of our tidal streams has effectively done this in certain areas; but the disagreeable odors and prevention of recreational use of the streams are too high a price to pay for elimination of fouling organisms (80, 81). As pollution of streams is reduced with improved waste treatment, these anaerobic conditions will be eliminated. This trend will probably be greatly accelerated in the near future, both by governmental pressure to eliminate such public health hazards and by the realization by industry that many valuable products may be recovered by proper treatment of their industrial wastes. Even in streams having completely anaerobic conditions certain mollusks are still able to survive. *Mya arenaria* (82), *Saxidomus gigantea* (83), and various others (84) have been shown to be capable of acting as facultative anaerobes. By this is meant that they are capable of obtaining energy by the breakdown of organic matter without use of oxygen.

It has not yet been conclusively demonstrated whether the Mollusca can continuously live under these circumstances, or whether they must be subjected at regular intervals or at stated times in their metamorphosis to aerobic conditions.

Sewage pollution will also encourage certain seaweed growth such as sea lettuce (*Ulva latissima*) (85), which also often blocks screens and tube sheets.

Control by Increasing Velocity

Where high velocities in the line can be obtained without unduly increasing the cost of pumping, such increased velocities may be used for the control of aquatic growths. The author has observed that *surface* velocities in excess of 1 fps minimize the set of mussel and barnacle larvae on smooth metal surfaces. However, very much higher *average* velocities must be attained to arrive at surface velocities of 1 fps on rough surfaces; such as, cast-iron pipe and concrete tunnel walls. *Mytilus*, which have already formed bivalve shells have been successfully removed by increasing the water velocity in cast-iron pipes to $13^1/_2$ fps for a period of 1 hr each week. However, such a means of eliminating marine growths is practical only where catch basins or strainers can be economically installed to eliminate the debris before it reaches critical equipment; such as condensers and pumps. The idea of high-velocity flushing is definitely practical for application in small plants. However, in large installations, particularly where large circulating- and cooling-water flows are required, the cost of pumps to attain these velocities, and the cost of power to operate them at the higher friction losses make the method economically impractical.

Control With Acids

The shifting of pH of the water was one of the earliest methods of control of fouling organisms attempted. Plant-scale experiments were carried out with partial success as early as 1915. Extensive studies of the effect of lowered pH on various mollusks were later carried out by Prytherch (86, 87) who was searching for an easier method of opening oysters to expedite commercial shucking. The natural pH of sea water and its buffer capacity varies slightly from place to place in the open sea (88) and varies considerably where industrial pollution is present. Fig. 8 is a typical curve of a buffer capacity of sea water, as measured by glass-electrode titration of samples taken in the open sea off the New Jersey coast.

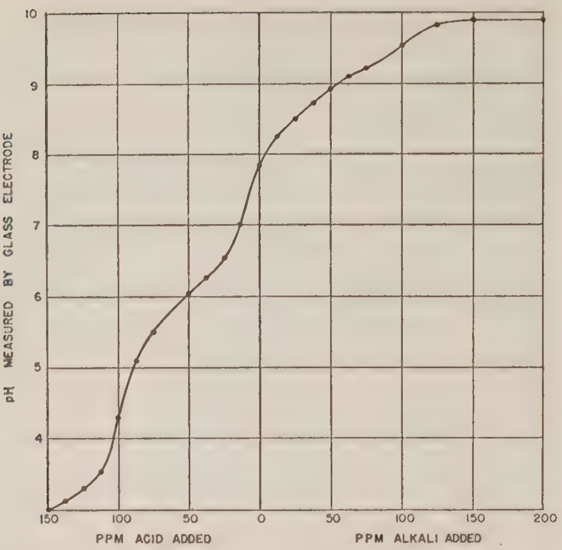

Fig. 8 Buffer Capacity of Sea Water Measured as pH by Glass Electrode With Sulphuric Acid and Sodium Hydroxide
(The curves remain flat at pH 9.9 until all calcium carbonate has been precipitated.)

It has been found that maintenance of pH between 2 and 5 depending upon the organism, and for periods varying from 1 to 3 hours will anesthetize the average *Lamellibranchia* and cause the shell to open sufficiently wide to permit insertion of a knife. Considerably longer periods are required actually to kill the organism. Corrosion, at these pH values, particularly of brass condenser tubes, is serious. As will be noted from the curve, 150 ppm acid is required to obtain a pH of 3. Translated to plant scale on a 25,000-kw electric generation station, it would require 5625 lb of 60-deg Bé sulphuric acid for a 3-hr period. Computed at current prices of $1.25 per cwt, that would amount to $70 per day of treatment. Corresponding increase in cost would, of course, occur in larger plants. Lower concentrations are useless. Ritchie (10) found that 10 ppm of sulphuric acid in water had no effect on mussels after submergence for 240 hr.

Control With Antifouling Paints and Surfaces

A large amount of carefully detailed experimental work and large-scale tests have been carried out by the United States and British navies and by various laboratories and paint manufacturers in an attempt to discover antifouling paints and surfaces for ships' bottoms. Literally thousands of patents have been issued covering various surface treatments to prevent the accumulation of fouling organisms. Paints containing as wide a variety of materials as mercurials, arsenicals, silicates, lead, copper, zinc, strychnine, cyanides, phenols, creosote, resins, asphalt, hair, guano, asafetida, and cow dung have been suggested (89). A smooth surface, while it will discourage fouling, will not prevent it, for even a smooth plate-glass panel will foul. Some authors (90) have suggested that paints can be compounded so as to slough off at high rates and take the load of fouling organisms with it. However, the required rates are high enough to make the method impractical.

Copper and copper-bearing paints have been found to be the most efficient antifouling materials, and recent exhaustive tests (91) have indicated that the antifouling properties of copper-bearing paints are directly related to the leaching rate of the metallic copper. Means of making laboratory determinations of leaching rates which correlate with fouling determinations have been devised (91).

Antifouling paints, which necessarily leach out material and become porous, should never be used to prevent corrosion. Protective primers of known corrosion-preventative ability should always be used under antifouling finishes.

All antifouling finishes eventually lose their efficiency; most of them after a single season. In long cast-iron pipe lines and on cement tunnel surfaces, the surfaces are too rough to make the use of paint economical. This is particularly true in plants where dewatering once a season for the renewal of the painted surface would be extremely expensive, as well as creating a hardship due to the required plant shutdown. On ships' bottoms and other metal surfaces which must be painted at regular intervals to prevent corrosion, it is the best method of preventing fouling yet devised.

Control by Screening

Water inlets of all power stations and other large users of industrial water are screened. However, to have a screen fine enough to catch the larvae forms of the fouling organisms, which are less than $^1/_{100}$ in. at early stages of development, is totally impractical. Any screen which would catch such organisms would quickly clog with microbiological accumulations, and unless made of corrosion-resistant material, would soon become permanently blocked with corrosion products.

Poisoning With Active Poisons

Various active poisons have been suggested as controls for marine growths. Cyanide has been suggested but has been rejected because of extreme danger to human life.

Some plants have used coal oil, gas-oil drip, kerosene, creosote, and similar oily products for control. Sections of the circulating-water tunnels between high and low tide become coated with these materials and become unsatisfactory places of attachment for fouling organisms. Some control is thereby obtained but no control is effected below the low-water line. Any method using a poison which is not dissipated, or consumed by mixing with other sea water, is dangerous in that it is likely to kill near-by oyster and clam beds and to cause public-health hazards at near-by beaches. Release of such poisons could well become the subject for damage suits and injunctive relief by the owners of proprietary oyster beds near the industrial plant using such poisons. Also, in almost all cases the cost of these poisons at the required dosage are prohibitive.

Other chemicals are useful in the control of particular organisms. For instance, quicklime spread over oyster beds at the rate of 400 lb per acre will kill starfish without affecting the oysters. This material burns the soft flesh of the starfish and causes lesions that fail to heal (92).

A patent (93) has been issued covering the use of cyanide in conjunction with chlorine for freeing a ship's bottom of fouling organisms.

Control by Chlorination

Chlorine is the most economical and widely used means of controlling fouling organisms in both salt- and fresh-water circulating systems. It has proved successful in many plants throughout the world. A few plants have experimented with chlorine and have failed to achieve control. These failures have occurred because of lack of appreciation of the need of proper residuals (i.e., active chlorine left in the water), proper treatment periods, proper variations of treatment with breeding periods, and proper control and distribution of chlorine in the circulating water.

Early experiments using chlorine have already been detailed.

Power stations scattered along the Atlantic coast from Massachusetts around the tip of Florida and as far along the Gulf coast as Corpus Christi have chlorinated successfully. The organisms thus controlled have covered a wide range of species.

Cursory details of English experience (94) and the installation at the United States Naval Air Station at Quonset Point, R. I. (95), have already been published.

It has been reported that successful control of *Mytilus edulis* has also been obtained with chlorine at the Corporation Power Station at Belfast and Dublin, Ireland, and Brighton and Southampton, England.

Early attempts using chlorine were based on continuous chlorination. Later short intermittent treatments were found unsuccessful. We have now found that properly spaced intermittent-treatment schedules are more economical than continuous treatment and are completely satisfactory. It will be remembered that all fouling organisms have a spat or larvae stage during their sexual reproduction and that infections in new areas can begin only by the ingress of such larvae. Therefore if the larvae can be killed at regular intervals, before they have completed metamorphosis, no new fouling will occur. In this the design engineer is fortunate, for most organisms during metamorphosis have a higher metabolism than at any other time in their lives and therefore are more vulnerable to poisoning.

It is obvious that if chlorination is carried out periodically so as to kill all the larvae in the tunnel, and is then repeated before any new larvae that enter the tunnel after treatment stops have had time for complete metamorphosis, complete control will be established. Such a statement is, however, an oversimplification of the problem. It is always possible, and will occasionally occur, that some larvae will arrive in the tunnel in an advanced stage of development and will complete metamorphosis almost immediately. When it is realized that a normal fouled shore line may contain a thousand million barnacles per mile and that these will release a million million larvae, or that a single *Mytilus edulis* may release as high as 25,000,000 eggs, it will be realized that only a very small percentage of the larvae need escape to cause serious difficulty.

A distinction should be drawn between complete prevention of fouling and substantial reduction or control of fouling. In all cases where fouling must be completely prevented, continuous chlorination during the fouling season must be used. However, where control only is required, intermittent chlorination may be used more economically.

In those cases where intermittent preventative treatment is to be used the lines should be purged by continuous treatment for short periods at regular intervals. Glass flanges should be inserted in pipe lines or test blocks should be inserted into tunnels and observations made at weekly intervals. As soon as the residual fouling that is not being prevented by the intermittent treatment, reaches the point where the organisms threaten to cause plant operating difficulty, then a corrective treatment should be instituted. This treatment will consist of chlorination at higher residuals and for longer periods than the preventative treatment. The reason for this longer period is obvious, since all of the *Lamellibranchia* are capable of closing and remaining closed for long periods. Chlorination periods must be extended until the organisms are forced to open their shells to obtain food or oxygen.

Observations by the author on experimental troughs set up at the inlet slip of the power station of the New York, New Haven, and Hartford Railroad at Cos Cob, Conn., through the courtesy of Mr. Sidney Withington, Mr. E. F. French, and Mr. John Coolidge, will illustrate the method of killing adult organisms. A large group of shelled *Mytilus edulis* of varying size was removed from the piers near the site of the experiment and placed in untreated wooden troughs with fresh sea water continuously circulated through the troughs. Time was allowed for the organisms to become acclimatized and no treatment was begun until all specimens were fully attached by byssus threads. Chlorination was then begun in different troughs at levels sufficient to yield 2.5, 5, and 10 ppm chlorine residuals.

Fig. 9 illustrates the rate of detachment and rate of killing of these adult organisms. It will be noted that the mussels break their byssus threads and attempt to migrate to more favorable environments before chlorination actually kills them. Killing, in this case, is defined as reaching a state such that when the organism is removed to clear fresh sea water it does not recover and in a few hours or days it has opened its shell and actual decomposition has begun.

Most larvae, as contrasted with adult forms of fouling organisms, are killed by chlorine residuals of 0.5 ppm applied continuously. This has been confirmed by experiments at various places along the Atlantic Coast from Massachusetts to Florida. However, on one plant scale operation in Virginia 0.5 to 0.6 ppm chlorine residuals maintained continuously failed to control mussels and barnacles.

In another case, an electric plant in New England chlorinated

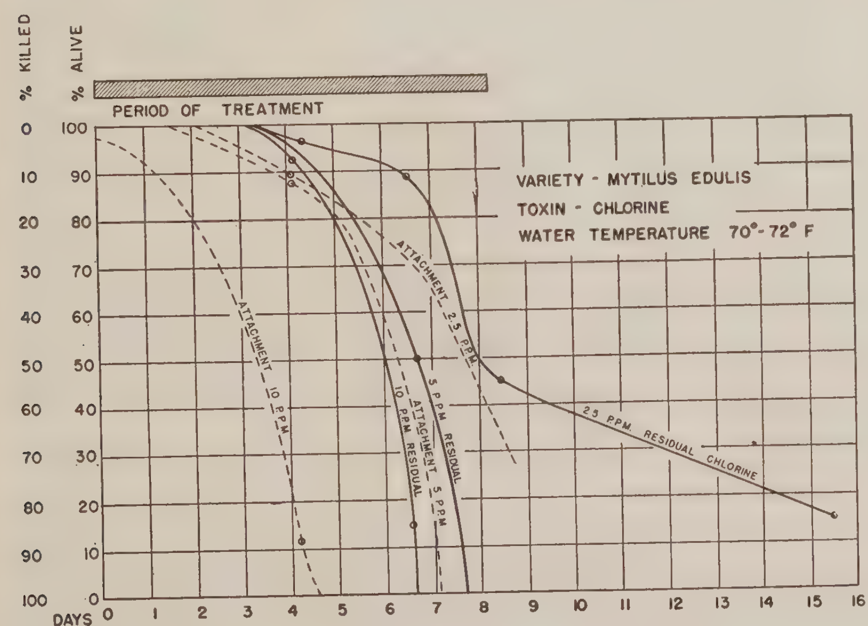

FIG. 9 TIME REQUIRED TO FORCE MUSSELS TO DETACH AND TO KILL MUSSELS IN VARIOUS CONCENTRATIONS OF CHLORINE

during one season at 0.5 ppm chlorine residuals on schedules of 20 min each, six times per day, and obtained complete slime control on condenser tubes, but showed no noticeable reduction in their annual accumulation of 600 cu ft of mussel and barnacle shells. The following season the same plant chlorinated at residuals of 0.7 ppm on a schedule of two 1-hr periods per day. The accumulation that year was 25 cu ft. Further reduction could have been effected.

In June, 1944, the plant of the Montaup Electric Company in Somerset, Mass., removed 20 tons of growth from its tunnels. This was considered a normal annual accumulation. Chlorination was started in January, 1945, and examination in June, 1945, showed less than $^1/_2$ ton of fouling-organism accumulation. Observation indicated that most of this had accumulated prior to the beginning of treatment. Because of physical variations among different plants with regard to water-circulating tunnels and the piping arrangements, as well as predominating fouling organisms, these experimental results and previous plant experiences should be used as a basis of design in other plants only with extreme caution.

Care should also be taken in instituting plant scale operations. For instance, in the beginning of treatment of a circulating-water system that has been in use for some time, a large quantity of organisms, usually sufficient to block tube sheets, and often sufficient to cause serious damage to circulating-water pumps, will be released if curative treatment is immediately applied. To prevent this, chlorine residuals should be held at low levels and chlorination should be short at the beginning, gradually lengthening as less and less fouling organisms remain in the tunnel.

In the preventative treatment, extreme care must be taken that the regularity of intermittent schedule be uninterrupted. The author inspected one tunnel that had been under treatment. Because of failure to obtain chlorine in the required time, the plant had been without treatment for 4 days. No continuous curative treatment was attempted to fill in the deficit. Three months later a full $1^1/_2$ in. thickness of *Pecten* of nearly uniform size had developed. During the 2 years following when no interruption in treatment occurred, no fouling was found.

The chlorination of salt water carried in steel lines has certain distinct difficulties. If continuous treatment is used, the slime film which ordinarily would inhibit salt-water attack on the line is completely removed and corrosion is aggravated. Cases have been observed in which $^1/_4$-in-thick scales have developed on steel lines within a few months. However, if corrosion-resistant alloy or cast-iron lines are used so that the natural aggressive action of salt water is minimized, chlorination will prevent bacteriological corrosion (96, 97, 98) and thus materially increase metal life. Where the water is naturally aggressive and the use of steel lines is indicated, intermittent chlorination, which will permit the reformation of protective slime films during idle periods, will minimize salt-water corrosion and yet maintain macroorganism control and also heat-transfer efficiency. Where the water is not naturally aggressive, such as is normally the case in fresh-water lines, continuous chlorination at properly controlled residuals will reduce rather than increase corrosion rates.

The means of applying chlorine to large circulating-water flows should be carefully considered. Chlorine is only slightly soluble in fresh water and is still less soluble in salt water. For this reason special apparatus for obtaining completely dissolved chlorine in water solution must be provided. In order to obtain marine-growth control in large tunnels and pipe lines, the chlorine solution must be evenly distributed through the circulating-water flow. Unless wide experience is available as to the type of flow that will be found in screen wells and suction chambers, it is likely that a point of application and means of diffusion will be selected which will completely free one area of serious growth and permit uncontrolled growth in other areas. It also should be remembered that while chlorine in the low concentrations required for the killing of these organisms is not corrosive to normal materials of construction, it is highly corrosive in higher concentration and particularly if released as a gas. Therefore, if adequate mixing and dilution are not provided local areas of high concentration may severely corrode pump bells, impellers, and casings.

The chlorination of circulating water will have no adverse effect on oyster or clam beds lying off shore nor on near-by bathing beaches. This can be readily appreciated when it is realized that the "chlorine demand" of normal sea water is such that the mixing of chlorinated water with an equal volume of sea water will eliminate or materially reduce the residual chlorine so as to be no longer toxic to oyster beds. The residuals carried approximate those used in swimming pools, and therefore if pH is maintained as it will be by the natural buffer capacity of sea water, beneficial rather than harmful effects would occur from swimming directly in the discharge water.

Throughout this discussion the term "chlorine residual" has been used to indicate a free chlorine residual or active HOCl as contrasted to a bound or chloramine residual, where the chlorine is bound by reaction with ammonia or amino compounds. Recent work (99, 100, 101) has indicated the wide difference between the killing power of chlorine and chloramine residuals. Chloramine residuals have distinct use for specific purposes since they are more stable in the presence of organic matter. They are therefore highly useful for the sterilization of swimming-pool water, and stock and pulp systems in paper mills. However, where ammonia content is not high, and this is true in most applications where marine fouling by macroorganisms is a problem, the use of free chlorine is recommended. Unfortunately, even if no ammonia is added, chloramine residuals often still occur because of the high ammonia and amino-nitrogen content of water which has been subjected to organic pollution. In those cases a question arises as to whether it is more economical to use "Break Point" chlorination and remove the ammonia, or to take the longer killing periods and higher required residuals of the chloramine. The determination as to whether a particular water with a particular chlorine dosage and contact period is yielding a chlorine or chloramine residual can easily be determined by the Laux-Nickel test (102, 103) or by titration with arsenite (104).

Chlorination should be carried out only at the higher residuals necessary to kill macroorganisms in their larvae form during those periods when the larvae are in the water. Naturally, most industries will desire to chlorinate at lower residuals during the remainder of the year to prevent formation of microorganic slime and thereby maintain heat-transfer efficiency (105).

The cost of marine fouling control with chlorine can usually be economically justified. A 25,000-kw station will usually be able to achieve control for about $3.50 per day during the fouling season. More than this amount will be saved due to improved vacuum as well as reduction in cleaning cost and outage resulting from marine fouling.

The time of year during which fouling will occur and rate of fouling will be dependent upon the fouling organism responsible for the difficulty, upon the water temperature, and upon the particular season. Seasonal variations are in turn dependent upon whether a large or small group of adult organisms succeeded in establishing themselves in the near-by waters during the previous reproductive periods, and whether these organisms survive to the next breeding time. Conditions of tide, particularly as it affects water temperature, will also influence fouling periods and rates. These factors are too numerous and varied to permit accurate forecasting. To obtain this type of informa-

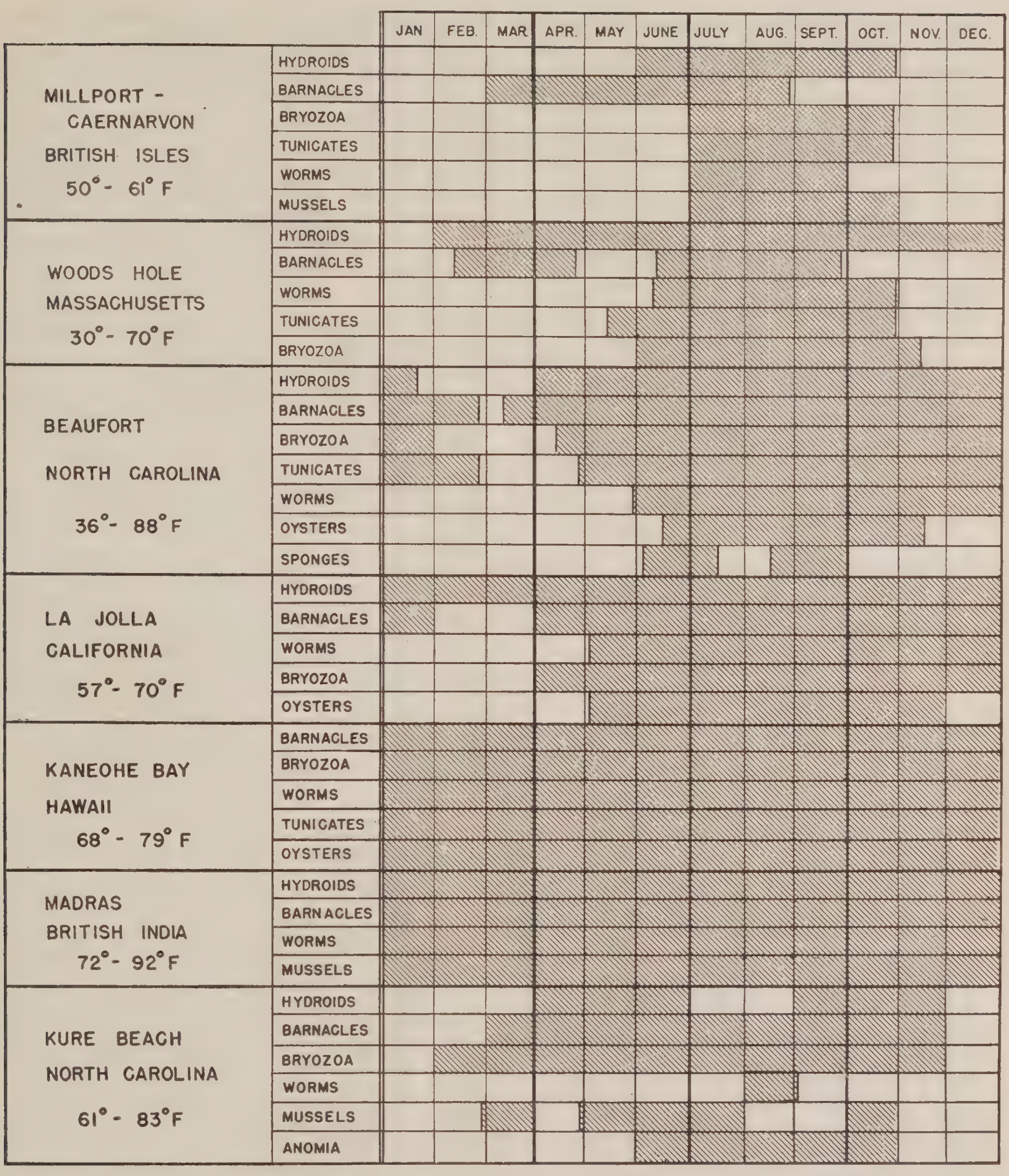

Fig. 10 Fouling Periods in Various Harbors

tion, the William F. Clapp Laboratories of Duxbury, Mass., and the Woods Hole Oceanographic Institution of Woods' Hole, Mass., have maintained for a number of years test blocks and panels at a large group of stations on the Atlantic and Gulf Coasts and elsewhere. These specimens are returned to the laboratories at regular intervals and examined to determine both settling rates and growth rates for the particular station and time. From these examinations much valuable information has been amassed.

The technique of setting such panels in order to obtain reproducible results over a long period is beyond the scope of this paper. Color and material of the panel depth and angle of submergence, type of support, and location of the tested area with respect to contaminating influxes and tidal rips are a few of

the factors to be considered. While results obtained from such test blocks cannot be used to forecast particular conditions at a station unless those tests are taken from the circulating flow of that particular station, some of the published results may be of interest (46). Fig. 10 indicates the fouling periods at widely scattered points.

In shipping or storing specimens for later examination, they may best be preserved by draining surplus water from them and placing in sealed bottles containing 98 per cent alcohol.

Designing for Fouling Conditions

In the design of new plants using fresh or tidal water likely to foul, two questions must be answered: (*a*) What will be the extent and nature of the fouling organism? (*b*) What design steps must be taken to overcome them?

Often forecasts of fouling conditions have been made, based on studies of test panels or of fouling on logs, piers, rocks, and structures near the site of a proposed plant, or on studies of fouling conditions at an adjoining plant.

Such forecasting is extremely dangerous and should never be relied upon. Industrial water supplies taken from large bodies of water usually afford ideal conditions for invertebrate growth in their screen chambers, pump-suction wells, supply pipes, and flumes.

The sessile organisms naturally dependent upon water motion to carry their food to them will thrive and grow at rates unprecedented on the previous shore line. This is naturally due to water velocities usually in the range of $^1/_{10}$ to 2 fps. Differentials of growth rates as high as 3 to 1 have, for instance, been recorded for barnacles in wave-swept waters as compared with barnacles living in comparatively still tidal water. Still faster rates of growth have been noted in industrial tunnels. Where the water taken by the industrial plant is used for cooling an additional factor which cannot be forecast is introduced. The enormous heat output in cooling water from a large power station or refinery is often sufficient to shift the over-all temperature of the body of water as much as 15 deg F. This shift in water temperature may be enough to encourage year-round growth and reproduction of an organism that has previously been dormant a large part of the year, or may permit overwintering and luxurious growth of some organism which has been previously subject to seasonal extermination.

A good example of this occurred in the Southwest during the war. A new plant was to be erected on tidal water and take its cooling-water flow from a brackish stream only a few hundred feet from the inlet of an existing oil refinery. Design engineers, basing their judgment upon the fact that the existing plant had experienced no difficulty, and upon a belief, based upon rather limited experience, that no organism could thrive in varying salt-content water, provided no protection from fouling. Luxuriant Bryozoa growth shut down the plant before a single year of operation had been completed.

For another example, there are two power stations in Rhode Island whose cooling-water inlets are a city block apart. Negligible differences exist between the plants in so far as salinity, water temperature, sewage pollution, and other normal factors are concerned. The plant upstream from the sea has had severe fouling with *Mytilus edulis;* the one nearer the sea has had none. The difference probably exists in variations in river bottoms. Some plants have experienced severe fouling recently after a period of comparative freedom, probably due to changes in industrial and domestic pollution in their water source.

It should be realized that even though conditions are known to be unfavorable to the fouling organisms that exist near the proposed plant site, these same conditions of salinity, temperature, and pollution may encourage luxurious growth of some other organism, not now native, when changed water flow and temperature due to the new operations occur.

It is only sound engineering to design as though fouling will occur. Satisfactory design will require that all portions of the system be accessible for inspection and manual cleaning; and preferably be capable of being readily dewatered; but such manual cleaning should be considered only as an emergency measure. From the standpoint of efficiency, of continuity of operation, and of cost of cleaning, control methods as previously discussed are more satisfactory than manual cleaning. Designing with such control methods in mind from the very inception of plans will often substantially reduce corrective construction and operating costs.

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Discussion

S. P. Ewing.[3] This paper is an excellent condensation of the extensive and scattered information on fouling organisms as related to circulatory salt-water systems, and the Bibliography will be helpful to those who wish to learn more about this interesting subject. The paper is another example of the economic and engineering value of pure science. Certainly the engineer cannot expect to deal effectively with the fouling problem unless he uses the knowledge of the marine biologist.

There is little that the writer can add to the subject of the paper. But it might be worth while to mention a few other instances where the engineer must deal with fouling organisms, and show how in each case the problem is an economic or engineering problem where the marine biologist could contribute to the solution.

The salt-water piping system on ships is similar to the shore cooling system. The fouling problem is in some respects more important because the piping is designed for relatively high velocities, the pipes are relatively small so that comparatively little fouling may greatly reduce the capacity. On combat ships, failure of the piping at a critical time might result in loss of the ship. Hence it is not sufficient to be able to remove the fouling at intervals. The piping and also the hull must be kept free of fouling at all times because nearly perfect performance at all times is essential. It is impossible to design and operate the ship's piping so that the water velocity exceeds the minimum required to prevent fouling. Nor is it possible to close completely parts of the system at regular intervals, and thus kill small attached organisms by suffocation. The use of chlorine or other poisonous chemicals is objectionable on combat ships because of space limitations and the added hazard of these materials in case of battle damage.

The possibilities thus seem to be limited to the use of (1) copper alloys which will prevent fouling, (2) antifouling paints on the interior of the pipes, and (3) chlorine generated electrolytically from sea water. One might think that with a large power plant available, it would be a simple matter to produce enough chlorine from the electrolytic decomposition of sea water to prevent fouling in salt-water pipes. The work that has been done by the Bureau of Ships and the experience on ships indicates that the best solution is to use copper-nickel pipes, which corrode enough to prevent fouling for at least 1 year and then use antifouling paint or abrade the interior of the pipe so as to restore its antifouling characteristics. With steel pipes, the best solution is the use of antifouling paint.

Another well-known instance where marine organisms interfere with the activities of man is the damage to wooden structures caused by marine borers. Here the best defense is to treat the wood so that these organisms are discouraged from boring in it, or to cover it with a material which the borers cannot penetrate. The method used since ancient times to prevent attack on wooden

[3] Bureau of Ships, Navy Department.

Note: The opinions expressed in this discussion are not to be construed as those of the Navy Department or the Naval Service at large.

vessels has been to cover the underwater hull with sheet copper. This method was used extensively on early steel and iron vessels. An elaborate and expensive system of wooden sheathing was used between the copper and steel to control the galvanic couple and thus prevent rapid corrosion of the steel.

There are numerous types of structures where fouling in itself is not particularly objectionable. For example, fouling on a wooden pier does not interfere with the proper functioning of the pier, so long as the organisms do not penetrate the wood and weaken it. Fouling on steel piers, lock gates, buoys, submarine nets, and submerged pipe lines is not particularly objectionable so long as the organisms do not penetrate the organic protective coating and cause corrosion of the steel. On lock gates, buoys, and submarine nets, the added weight may affect buoyancy because the shells are heavier than the water. However, the effect on buoyancy is only about $^1/_4$ of the total weight in air of the fouling growth.

It is well known that when certain barnacles grow on relatively soft bituminous materials and some paints, the shell of the barnacle gradually penetrates the organic material so that after the barnacle reaches its full growth its shell is in intimate contact with the steel over nearly the entire area of its base. Serious corrosion of the metal often occurs in this area, especially if the electrical conditions are favorable for corrosion. This situation has occurred on lock gates and submerged pipe lines. Other unidentified marine organisms are reported to attack asphalt coatings in a somewhat similar manner.

There are several possible ways to prevent damage to the coated metal. An antifouling paint might be applied over the bituminous coating, but this will be effective for only a limited time, since most antifouling paints lose their effectiveness in less than one year. Tests may show that a sufficiently hard coating will prevent penetration. This is certainly the case with barnacles. A sufficient amount of hard mineral filler in the bitumen may be an effective means for discouraging other types of boring or digging animals. If the attack of the marine organisms on the coating is rather slow, and only a few holes are made in a relatively long period, the use of cathodic protection to prevent corrosion at the holes in the coating may be the most economical solution. Each case will have to be studied in order to arrive at the most effective and economical solution; and the solution will probably require identification of the culprit, and a knowledge of its life history.

L. W. Hutchins.[4] This paper fills a gap in the literature of fouling problems. The engineer is presented with an introduction to some of the complex biology involved in the fouling of water circuits, and, indeed, of any fouling. On the other hand, the biologist is offered an excellent review of the practical problems from the engineer's standpoint. The number of new observations recorded and the extensive review of the practical literature, which generally does not come to the attention of biologists, are particularly worthy of praise.

The author has pointed out very well the extent to which the solution of engineering problems must depend on competent biological knowledge. His remarks on the inadequacies of fouling control by screening, on the possible variations of control by heat, and in particular, the variable dosages of chlorine which must be adjusted both to the types of organisms present and the stages of their life histories are good cases in point.

It happens not infrequently that biologists are not able to give the best possible consulting advice about fouling problems simply from ignorance of the practical aspects of the situation. This situation can be alleviated by papers such as this which present both sides of the picture.

[4] Woods Hole Oceanographic Institution, Woods Hole, Mass.

Cases have come to the writer's attention in which plants and designers have refused to allow dissemination of the factual information about the fouling problems they have encountered, in the rather naïve belief that the occurrence of fouling in their setup was a reflection on the abilities of the operator or designer. That view is quite incorrect.

The only sound procedure, as the author points out, is to design with the expectation—a practical certainty—that fouling will occur, and to make suitable provisions accordingly for inspections, cleanouts, and, if desired, attempted control. The failure of any intended control system, moreover, does not necessarily reflect on the engineer, but as often as not, on the inadequacies of the biological knowledge concerning his problem.

Improvement in this general subject can be obtained only by a thorough understanding of both points of view so that the biologist can know the lines which his investigations should follow, and the engineer can have a true picture of the information available for application in designing and operating control measures

The need for more exact biological knowledge has become increasingly clear along two lines. In so far as the economical use of control measures must involve their quantitative relation to the incidence of fouling, it is obviously desirable to know as fully as possible the fouling expectancy and the effects of factors modifying it. The particular fouling population which will develop in a highly localized site appears to depend on the natural populations to which that site is accessible. In harbors and other such locations generally of interest to industry this question of accessibility appears to be highly complicated. The set of local currents may profoundly alter the probable accessibility as judged by simple geography. Pollution, which may be either somewhat beneficial or detrimental to the growth of individual species, depending upon its nature and extent, is another factor believed to be of very great importance in such areas. Far too little is known, however, about the effects of individual contaminants to permit predicting localized harbor fouling conditions with any assurance. There is a need for further facts about all such factors modifying the incidence of fouling.

The second line along which further exploration is indicated is that of the knowledge of toxicity and factors which may be utilized in control of fouling. The investigations of possible toxics other than chlorine and heavy metals can hardly be said to have been prosecuted with any vigor. Knowledge is inadequate even as to possibilities, particularly in the line of organic toxics. Similarly, with reference to chlorination, it would seem worth while to investigate much more thoroughly than has been done the problem of control by intermittent treatment. Even with the intermittent applications now in use, the cost appears sufficiently high so that possible savings would more than justify considerable research aimed at the reduction in chlorine usage. This is a problem in which the biologist must go first, studying the effects of various concentrations and of various time dosages on a wide variety of organisms and attempting thereby to develop generalizations which can be put into practice by the engineer.

The writer feels that the greatest importance of the paper is as an aid to the further profitable co-operation between professional biologists and engineers.

F. L. LaQue.[5] On the basis of some of our investigations in related fields, the writer is able to confirm some of the author's conclusions:

1 We have found that when sufficient chlorine is used to prevent the development of slime films, as well as fouling organisms,

[5] In Charge Corrosion Engineering Section, Development and Research Division, The International Nickel Company, Inc., New York, N. Y.

corrosion of steel pipe lines by sea water moving at high velocity is increased, presumably through the loss of the protective effect of slime films and accumulations of macroorganisms. At the same time, there has been some evidence that small amounts of chlorine in sea water retard rather than accelerate corrosion of some nonferrous alloys, such as Admiralty brass and 70:30 cupro-nickel, perhaps through some reinforcement of protective films. Likewise, the prevention of fouling and perhaps, also, a slight increase in the oxidizing capacity of sea water due to the presence of a small amount of chlorine, have been found to exert a beneficial effect on the performance of stainless steels.

2 Our observations generally confirm those of the author with respect to the influence of water velocity on the fouling of smooth surfaces. We have found that macroorganisms are not likely to become attached at steady flow rates much above about 3 fps, but once attached during periods of lower velocity flow, they may be able to retain a foothold and grow in contact with water moving at much higher velocities, e.g., 10 fps.

3 Examinations of many specimens of steel covered with fouling organisms after exposure to sea water have failed to indicate any significant effect of the organisms in accelerating corrosion through differential cell action as suggested by the author. However, with highly alloyed steels, such as stainless steels and other passive alloys, local attack believed to be due to differential cell action under fouling organisms is common.

Since the possible effects of chlorination on corrosion of equipment are of natural concern to those contemplating the use of chlorine to prevent fouling, it is hoped that more extensive investigations of this phase of the subject, and particularly with respect to intermittent chlorination, will be made and that the results will be made available.

S. T. Powell.[6] The author has given an excellent history of methods undertaken for the control of biological microorganisms responsible for fouling condensers and similar equipment. In addition to this valuable record, he has presented a very clear and concise discussion of the specific types of organisms which have been responsible for such difficulties and has submitted a constructive review of miscellaneous corrective treatments and their application. The paper also includes one of the most complete bibliographies which has been published, and this material alone is an invaluable contribution to this subject.

That portion of the paper dealing with "control by chlorination" is a concise statement of the value and limitation of this type of treatment and should correct many misconceptions as to the application of chlorination.

In general, the paper is a worth-while contribution to the literature and represents an authoritative reference work with a high degree of reliability.

Lester Randall.[7] The writer in support of the facts given in the paper is able to provide some results obtained at the power station of The Connecticut Power Company, Stamford, Conn.

The Stamford station has grown from the early days of the industry so that when an additional unit of 25,000 kw capacity was installed in 1940–1941, a complete new intake system for this unit was necessary. The circulating pumps are propeller type, located in the screen house on the waterfront. The intake tunnels are two 36-in. cast-iron pipe lines from the pumps to the condenser, one line to each side of the divided water box. The combined length of the two lines is 262 ft.

Circulating pumps for the old portion of the plant are of the submerged slow-speed centrifugal type, located in another screen house also on the water front. The intake tunnel for the old plant is also 36-in. cast-iron pipe but laid out as a loop, like a hairpin, with the two ends connected through valves to a header located near the pumps. Take-offs to the condensers are on one leg of the hairpin so that in effect there is a cooling-water header supplying the old condensers, this header being fed from both ends. Division valves make it possible to inspect sections of this line while other sections are still operating. The whole length of this loop, including leads to condensers, is 680 ft, but only 570 ft are in active use, the remaining section being connected to units not regularly operated, and because of a closed division valve very little water circulates through it.

The new unit was put in service during October, 1941. Intermittent chlorine treatment was started in January, 1942, for this unit and in April for the old part of the station. Prior to this time no method of controlling intake fouling had been used. Heavy growths occurred in the pipes and had to be scraped out at least twice a year. The fouling included all sorts of marine animals but mussels predominated.

The new plant has operated as a base-load unit, being shut down over a week end every 6 to 8 weeks for inspections and maintenance. The old plant seldom operates over a week end and when shut down no chlorine is fed, although a small amount of water is kept circulating through the lines.

Circulating water is taken from the harbor, an arm of Long Island Sound. Normal water velocities during the summer in the several pipes are 5 fps for the new plant; 6 fps for a 325-ft section of the old intake piping; and 7.5 fps for another 240 ft.

Until July 20, 1942, the program for chlorination was three 20-min periods per 24 hr, at which time the treatment was increased to four 20-min periods. The following year, 1943, the program was changed to two periods of 60 min each 24 hr. This was started May 12, 1943, and has been in effect most of the time since.

Residuals have been carried at 0.5 ppm, except during July, August, and September, when it has been kept slightly higher, from 0.55 to 0.65 and occasionally, for short periods, to 0.7 ppm. Samples for residual determination are taken from the condenser outlet in the new plant and condenser inlet in the old plant.

Results have been gratifying. Considering the old plant only, the 1942 program of three and four 20-min chlorinating periods per 24 hr reduced the total amount of material to be removed manually by 50 per cent. The two 60-min periods per 24 hr has reduced it by more than 95 per cent when compared with conditions before chlorine treatment.

Results for the new plant are even more interesting. From October, 1941, to August 12, 1945, nearly 4 years, a total of 10.5 cu ft of material, all mussels, has been removed from the 262 ft of 36-in. pipe and 8-in. suction lines to the cooling-water booster pumps. One half of this total was taken out the first year, 1942, when using the 20-min chlorinating period. Probably an important reason for the small amount this first year was the smooth condition of the pipe surfaces, which had been given a shop coat of bitumastic paint.

Prior to chlorine treatment, it was regular operating practice during parts of the year to drain half a condenser at night and pick out the mussels and shells that were wedged in the ends of tubes. Sometimes it was necessary to do it during the noon hour. This operation has been completely eliminated. Rubber plugs are blown through all condenser tubes every 4 weeks to remove such shells as do lodge in the tubes and might cause failures.

The cost of chlorination for the combined plant is 15 tons of chlorine per year; generation for the same time is up to 300,000,000 kwhr.

Considerable time has been given to studying mussels. The beginning and duration of the spawning season and the rate of

[6] Chemical Engineer, Baltimore, Md. Mem. A.S.M.E.

[7] The Connecticut Power Company, Stamford, Conn.

growth are important considerations. Four years of observation has pretty well determined that with the present chlorine treatment, mussels will not begin to appear in the pipes until July 1. It has also been determined that if the pipes are cleaned after October 15, they will remain free of any growth except a scattering of barnacles, for the following 8 months. For removing such mussels as do get started two cleanings are made, the first about August 20, the second about October 20. Using this schedule, mussels that may get started are removed before they get large enough to crowd each other loose from the pipe surface and be carried along by the current.

These results apply to Stamford station. Other stations must determine the treatment best suited for their particular conditions. Some of the factors to consider are nature of tunnel surfaces; velocity of water; size of tunnels; continuity of service and chlorine treatment; nature of fouling organisms.

The author suggests installation of glass inspection plates or hanging test pieces in tunnels. The writer considers that method unreliable, especially for salt-water plants.

In the old plant, during the summer of 1945, a solid blanket of mussels covering 8 ft of 36-in. pipe became established in a section between a condenser take-off and a closed division valve. Also a good-sized patch started in the 24-in. connection at one side of one of the old condensers. Why these patches start in an otherwise clean pipe is difficult to explain. In the new plant, water for the turbine coolers (hydrogen and lubricating oil) is taken through 8-in. suction lines to two 750-gpm booster pumps, one on each main intake pipe. One pump is operated at a time for 1-week intervals. Each summer, mussels completely blanket the insides of these suction lines. Also, in the new plant mussels completely filled the crease in the reinforced rubber expansion joints just below the water boxes.

So far only intake pipes have been mentioned. The pump and screen wells do collect growths on the walls. This is due to a design that does not give good mixing of chlorine with the water in the wells. Chlorine feed pipes are placed just in front of the pump suction, between the pump and the screen, so the chlorine passes directly to the pump and into the pipes.

Chlorine has eliminated all non-shell-bearing animals, slimes, etc., and nearly all mussels from the circulating-water piping. However, barnacles do not seem to be affected, in fact it would seem they like the clean surfaces. In the old-plant intake lines they will nearly cover the surface by midsummer. They are hard to scrape off but because they are so securely fastened to the pipe walls and are so small, they are not a hazard in operation of the plant.

W. D. Bissell.[8] At the time the chlorination system was installed at Montaup Electric Company one of the condensers had been completely retubed the previous year and another condenser was completely retubed at the time the treatment was started.

The selection of sampling points for observation and control was carefully considered and these were located at the discharge elbow of the circulating pumps, each condenser discharge elbow, half the distance along the pump discharge tunnel where house service pumps are located, and also at the combined discharge of all condensers.

As the tunnel system was known to have considerable growth, extreme care was used when treatment was first started in February, 1945, in order to observe the sluffing-off of this growth. A thirty-minute cycle per day was used for about two weeks maintaining a residual of 0.3 ppm at the condenser outlet. The cycle was then changed to fifteen minutes of treatment three times daily and the residual was increased to 0.5 ppm, where it has been maintained since. No difficulty was experienced from accumulated growth dropping off during the initial cleaning period.

It was interesting to observe, during the difficulty which was experienced maintaining chlorine residual during the first month of treatment, that the discharges of the pumps would vary in chlorine residual from day to day. This variation leveled off later so that each pump discharge would indicate the same residual.

It is believed that this variation was due to the uneven deposits in the pump suction and the gradual cleaning of them resulted in the leveling off of residual at all pumps.

The greatest demand of chlorine was along the distance from the river intake screens to the pump discharges which was three or four times greater than through the condensers from the inlet to the outlet. Due to having sampling points at these important locations close observation could easily be obtained. The results were accurately logged which was of great assistance in maintaining machine control and residual control.

When the water temperature reached about 45 deg in the spring of the year the chlorine demand increased very rapidly until 50 deg was reached. From that time to 75 deg, chlorine demand was observed to be quite normal. During the period of warm water operation at temperatures of from 68 deg to 80 deg, an interesting changeable condition would occur in the foam on the surface of the condenser discharge water. This would be very dark brown for a few days at a time and would then change to a light yellow and the chlorine demand would change rapidly.

The control of chlorination was obtained by results as indicated on the condenser tubes, sheets, inlet and discharge water boxes, at regular frequent inspections. When chlorination was first started the growth of so-called moss and marine growth was about $^3/_8$ in. thick on the internal surfaces of the water boxes with some adhering to the tube sheets. After several weeks of treatment this could be easily removed by rubbing with the hand. In about three months this growth was completely removed and the surfaces could be coated with red lead.

Previous to chlorination the condensers were rather foul-smelling and it was extremely difficult for men to make repairs or to stay in them any length of time. Extended periods of working in a condenser would make a man ill. Since chlorination has been used the condensers have not been odorous to any extent, and are not objectionable for men to remain inside for extended periods to make any necessary repairs.

The purpose of using chlorination was to remove slime deposits on the heating surfaces chiefly. Mussels in the intake tunnel were of considerable nuisance and also would collect in various station salt-water lines and valves which caused considerable trouble in the past.

Chlorination has completely removed the slime and reduced the mussel growth so that it is negligible. The station water lines have been free from any growth or deposits and it has not been necessary to clean cooler surfaces, etc. Previous to chlorination such conditions as described in Figs. 2, 3, and 6 were usually found when the tunnel was dewatered.

The use of chlorination has been quite satisfactory in keeping the condensers and the condenser water system clean and we feel that good results have been obtained. However, some points have been raised such as the following:

1 In keeping condenser tubes free from slime and any organic growth by the use of chlorination, will this permit corrosive action caused from oxygen or any corrosive gases in the circulating water to attack the condenser tubes, which may shorten their life?

2 (a) Does the type of bacteria change at different seasons of the year? (b) Do they change at different temperatures? (c)

[8] Engineer, Somerset, Station Montaup Electric Company, Fall River, Mass.

Is it usual to have chlorine demands change rapidly from causes such as mentioned above?

3 In the process of first applying chlorination will chlorine demands change rapidly in initial cleaning periods?

4 Do river-bed deposits change gradually from year to year, thereby affecting the type of growth and gases in the water which may result in changes in the chlorine demand?

5 (*a*) Is it considered advisable to discontinue chlorination during the winter period when temperatures are at 33 deg or thereabouts? (*b*) Should the chlorine cycle be reduced in frequency? (*c*) Should the chlorination residual also be reduced?

6 If marine growth and fouling organisms vary from season to season at various temperatures, is not the best means of control obtained from observation of results which can actually be a guide for correct residuals, thereby obtaining positive control?

7 Since variable conditions of the tide are observed to have an effect on the chlorine residual, can this be explained?

Author's Closure

The author wishes to express his appreciation to the various discussors for their thoughtful and well-prepared contributions. The discussions papers by W. D. Bissell and L. E. Randall are interesting in that they present the viewpoint of the problems as seen by the operating man.

Mr. Bissell has pointed out a familiar observation that the chlorine demand, as indicated by the difference between chlorine dosage in the screen well and chlorine residual in the tailpipe, is made up of two parts: (1) The demand of the water itself during the time of contact permitted, and (2) the demand or absorption of chlorine by the surface accumulations on the tunnels and heat-exchanger tubes. Naturally, when large accumulations exist in a concrete tunnel, considerable chlorine will be absorbed by this organic matter until such time as it has been completely oxidized, and in most cases, has completely sloughed off. Mr. Bissell has raised certain specific questions which the writer will attempt to answer. The question as to whether the chlorination of condenser tubes and the resultant bright tube condition will permit the aggressive action of salt water, of dissolved gases, and of other corrosive fluids to increase the corrosion rate of these tubes over the corrosion rate existing in slime-covered tubes is a subject which will still bear considerable investigation. Mr. LaQue, in his discussion of this paper, seems to feel that such accelerated corrosion does not exist, and that, in fact, the elimination of the slime film and macroorganism fouling will retard the corrosion of nonferrous metals. Further research should be carried out in this field; but suffice to say, from a practical viewpoint, corrosion rates of installed tubes have never been demonstrated to have been increased by chlorination. A few cases are on record where the tube corrosion rate increased simultaneously with the use of chlorine for elimination of slime accumulations, but in all cases where the chlorine was abandoned for a period, the increased corrosion rate continued unabated.

The type of bacteria and other slime-forming organisms responsible for heat-exchanger sliming do not vary in terms of the season. However, the type of organism present at any time is dependent upon water temperature, upon food supply, and upon other factors such as oxygen content in the water. Since these factors, in turn, have seasonal variations, it is possible that specific flora will be more regularly found at certain seasons of the year. Certain microorganisms multiply very rapidly at specific temperatures, but are in practically static conditions in all but a very narrow range of temperature. Certain others are tolerant to much wider temperature conditions, and will thrive throughout the temperature range of normal circulating-water conditions. Mr. Bissell has outlined in his own discussion a very clear demonstration of the high chlorine demand of fouled surfaces during the initial clean-up period of a circulating water system. His experience is quite typical of the conditions in a normal circulating-water tunnel.

The variations of river bottom do not have a particular effect upon the growth of microorganisms in circulating water. In most cases these organisms exist free-floating in the water, and unless the river bottom is contributing changes of water conditions, such as oxygen depletion or hydrogen sulphide production, the condition of the river bottom will not seriously affect the growth of microorganisms on heat-exchanger surfaces. However, the macroorganisms, such as mussels, have only a limited field of locomotion between the point of fertilization of the egg and final point of attachment of the larva animal. For this reason, an invasion of mud over a rocky bottom, which quite frequently occurs in tidal water, may well eliminate the breeding grounds and ultimately the source of infection for macroorganisms for a given tunnel.

The question as to what changes of treatment should be made as water temperatures approach freezing is one that can be answered only after a study of the particular problem. It should be remembered, however, that temperature at which the organism is breeding is not the temperature of the water, but rather the surface temperature of heat-exchanger metal, and for this reason, even at 32 deg water temperature there may be lively growth of the microorganism. As a practical matter, chlorination should be continued at residuals sufficiently high, properly spaced, and of sufficient duration to kill the flora attached to the flume, condenser, or other surfaces.

The answer to Mr. Bissell's sixth question is again a matter of practical application of biological facts. If a station could afford to have continuous studies by an experienced biologist to determine when the spat form of each organism is present in the water, then residuals and dosage periods could easily be varied in accordance with season and water-temperature conditions, so as to yield maximum economies of chlorine use. But the average plant finds it much easier to feed chlorine at residuals sufficiently high to kill the spat form of the macroorganisms during the entire period when water temperature is sufficiently high, or seasonal advance is sufficiently completed to warrant a belief that some of the fouling organisms may be breeding.

Variations of chlorine demand with tidal conditions is quite common. The common condition of a polluted stream moving into a relatively clear estuary will yield high chlorine demands during periods of ebb-tide, due to the preponderance of river water and low chlorine demands while the tide is in flood due to the preponderance of sea water. In any particular harbor the question of what mixing will occur at various tides can become extremely complex.

Mr. Randall's report is an excerpt from the data which he has accumulated during the past six years. These experiments are the most carefully controlled and fully observed plant-scale demonstrations of the control of fouling organisms by chlorination known to the writer. I hope that Mr. Randall will publish his very carefully detailed data in the near future. His experiences have indicated some of the pitfalls which may well trap the unwary operator attempting chlorination for the elimination of fouling organisms. No plant which is circulating any salt water should permit that salt water to circulate for a period of more than twelve hours without treatment. Mr. Randall's difficulty with his dead-ends in the old station is typical of the problems which will occur if this precaution is not taken. In the placing of test panels or plate-glass windows in circulating-water tunnels, the place chosen for test must be representative of conditions throughout the entire station, and if any doubt exists as to whether the place chosen is representative, spot checks throughout the system should be made. The control of barnacles re-

quires slightly higher residuals than those required for the killing of other fouling organisms, and therefore unless barnacles are giving trouble it is uneconomical to chlorinate at these higher residuals.

In the cleaning up of fouled tunnels either when starting to control fouling in existing tunnel, or in the cleaning of a tunnel which has been under treatment and which has been permitted to foul to a limited extent by "preventative treatment" recommended in the original paper, there is always a question of the economy as to whether chlorination or manual cleaning should be used. In all cases where plant shutdown and dewatering is readily possible and access to the tunnel can be made without undue expense, it would seem advisable to choose manual cleaning. Under other circumstances chlorination might easily be the answer.

Dr. Ewing's discussion of problems related to the fouling of circulating-water systems has been very valuable in rounding out the view of the subject. The writer has had some contact with the design work involved in the production of chlorine cells for use on board fighting ships. Certainly considerably more improvement of those cells, and improvement of the antifouling materials for lining the ships' pipes is in order. Such development can best be carried on by the co-operation of all interested parties.

Dr. Hutchins' discussion of this problem from the viewpoint of the biologist will warrant careful study by the designing engineer. His warnings of the unpredictability of fouling in tidal waters should be a warning to all designers faced with this problem. His plea for effective co-operation between the professional biologist and the engineer will, I hope, encourage further publications by operating men and designers who have experienced difficulty in this field.

Mr. LaQue's authoritative discussion on corrosion is much appreciated. It is the writer's hope that further work along these lines will be undertaken within the near future.

All the discussors have been kind enough to point to the gravity of the problem of fouling organisms to circulating-water systems, and the writer trusts that these discussions will lead to a fuller appreciation of the problem and careful consideration by the designers in the planning of future circulating-water tunnels.

Discussion

In 1945, when there was a ban on national meetings, some papers originally scheduled for these meetings were presented before local groups. In the case of these papers the Committee on Publications suspended its rule, which requires simultaneous publication of paper and discussion, and accepted discussion based on the published paper.

Irreversibility in the Theoretical Regenerative Steam Cycle[1]

G. M. Dusinberre.[2] To future workers on this subject the writer recommends the analysis based on 1 lb throttle steam as used by the author. This seems more straightforward than to base on 1 lb through steam.

The author's Fig. 5 is a concise way of presenting the data on this cycle. But if a need really exists for data on the properties of this cycle, the writer feels that neither heat-rate tables nor the entropy-increase diagram is adequate, for two reasons: (1) Reheat cannot be handled except in a roundabout manner, as the author points out. (2) This ideal cycle is based on net work, that is, turbine work less feed-pump work. The practical use of the data would probably be in the study of high pressures where pump work is relatively large. Since there is no necessary relation between turbine and pump efficiencies, the application of any sort of efficiency factor to the ideal heat rate at least makes the fourth significant figure look a little foolish.

The writer concludes that the most useful form in which the data could be presented is (a) to use the divided cycle proposed by Markson,[3] and (b) to tabulate or plot the turbine work, pump work, and through steam, for 1 lb throttle steam: The corresponding heat items are readily obtained from the steam tables.

If the pump work is required, the author's integration is not directly useful as we need values of $(1-w)$ for $\int(1-w)v_f dp$. To get these, the writer has found that $\int \frac{T_f}{H-h_f}\, ds_f$ is more convenient than the function used by Selvey and Knowlton,[4] as the curvature is less.

When data are in graphical form, such as an indicator card, the logical method of integration is to use a planimeter, and not to pick off points for numerical integration. But when the data are in numerical form, as in the present case, it is equally more logical to use some method of numerical integration rather than to introduce the extra step of plotting a curve.

H. G. Elrod, Jr.[5] A completely accurate alternative to the author's Equation [6] can be obtained readily in the following manner.

Consider a regenerative cycle $ABCXZfA$ in the author's Fig. 1. When 1 lb of throttle steam is circulated in this cycle, $(1-w)(s_z-s_f)$ units of entropy are rejected by the system, and (s_c-s_a) entropy units are received. The net entropy increase of both the system and its surroundings is given by

$$\Delta S = (1-w)(s_z-s_f)-(s_c-s_a) \quad [1]$$

The variation of ΔS with exhaust conditions is found by differentiating Equation [1] of this discussion

$$d(\Delta S) = -(1-w)ds_f-(s_z-s_f)dw \quad [2]$$

Using the author's Equation [3] (corrected to include a minus sign), we obtain

$$d(\Delta S) = -(1-w)\left[1-\frac{T_f(s_z-s_f)}{H_z-h_f}\right]ds_f \quad [3]$$

Substituting the value of $(1-w)$ from Equation [1] into Equation [3] of this discussion, we find

$$\frac{d(\Delta S)}{s_c-s_a+\Delta S} = d\ln(s_c-s_a+\Delta S) = -\left[\frac{1}{s_z-s_f}-\frac{T_f}{H_z-h_f}\right]ds_f \quad [4]$$

The left-hand side of Equation [4] is directly integrable. The right-hand side may be evaluated in the manner illustrated by the author.

A comparison of Equation [4] with the author's Equation [6] shows that the author's approximation amounts to the assumption that $\ln(1+x)=x$. Using Equation [4], we obtain for the example cited by the author

$$\ln\left(1+\frac{\Delta S}{s_c-s_a}\right) = 0.0288 \quad [5]$$

or

$$\Delta S = 0.5425 \times 0.02922 = 0.01585 \quad [6]$$

The corresponding heat rate is 6000, which more closely agrees with the accurate value of 6001.2 computed by Shapiro.[6]

Ernest L. Robinson.[7] When the writer first became interested in this subject, there were great gains to be made in station performance. In 1923 he published curves for the efficiency of the extraction cycle which extended up to 1000 psi and 700 F, pressures and temperatures far beyond anything in use at that time.[8] The extraction cycle is now commonplace with steam conditions far above those. The gains looked forward to 20 years ago have been more than realized, and modern discussions are largely confined to the technique of calculation.

The writer has always believed that the best technique of calculation is to formulate the difference from some precisely known relationship and then to calculate that difference with care.

[1] By R. E. Hansen, published in the October, 1945, issue of Trans A.S.M.E., vol. 67, pp. 557–560.

[2] Department of Mechanical Engineering, Virginia Polytechnic Institute, Blacksburg, Va.; now on duty at U. S. Naval Academy, Annapolis, Md. Mem. A.S.M.E.

[3] Discussion by A. A. Markson of "Theoretical Regenerative-Steam-Cycle Heat Rates," by A. M. Selvey and P. H. Knowlton, Trans. A.S.M.E., vol. 66, 1944, pp. 504–505.

[4] Selvey and Knowlton, reference (3) of the paper.[1]

[5] Department of Marine Engineering, U. S. Naval Academy, Annapolis, Md. Jun. A.S.M.E.

[6] Discussion by A. H. Shapiro, of "Theoretical Regenerative-Steam-Cycle Heat Rates," Trans. A.S.M.E., vol. 66, 1944, p. 508.

[7] Turbine Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[8] The Margins of Possible Improvement in the Central Station Steam Plant," by Ernest L. Robinson, Trans. A.S.M.E., vol. 45, 1923, p. 644.

Mr. Hansen has selected the Carnot cycle for his precisely known relationship and has proceeded to formulate the departure of the extraction cycle from perfect reversibility when superheated steam is bled.

This is an excellent method to pursue, as it enables attention to be focused on the magnitude of the departure itself, and thus the errors of calculation become errors of the second order of smallness.

The author takes advantage of this to introduce as an approximation the assumption that the weight flow throughout the cycle is in inverse proportion to the entropy difference between steam and feedwater. The close check with the Selvey and Knowlton tabular values justifies the assumption.

However, since the author uses graphic integration anyway, it does not seem to the writer as if it would complicate his calculation if the correct expression for weight flow were used. When the writer first set down this expression, he used w to represent the weight flow of extracted steam per pound to the condenser. Thus the weight flow in the turbine per pound to the condenser is

$$1 + w = e^{\left[\int_{h_G}^{h} \frac{dh}{H-h}\right]}$$

In evaluating the integral for high pressures it would be desirable, as Selvey and Knowlton pointed out, to allow for the feed-pump work.

Thus

$$1 + w = e^{\left[\int_{h_G}^{h} \frac{1-\frac{vdp}{Jdh}}{H-h}\, dh\right]}$$

The author uses w to represent the weight flow of extracted steam per pound to the throttle. Thus, using his notation, the weight flow in the turbine per pound at the throttle is

$$1 - w = \frac{1}{e^{\left[\int_{S}^{S_A} \frac{Tds}{H-h}\right]}}$$

In conclusion the writer would like once more to call attention to the advantage of weight-flow curves in extraction-cycle analysis. Especially with reference to a fixed exhaust pressure, such as 1 in. Hg, they may be drawn on the Mollier chart as characteristic lines per pound to the condenser for any stage efficiency. With the weight flow throughout the cycle known, performance is easily computed from cycle input and output. This technique is particularly useful for actual efficiencies other than 100 per cent.[9]

Ascher H. Shapiro.[10] The relation between Equations [2], [5], and [7] of this paper, and the first and second laws of thermodynamics is most obscure, due, it is felt, to the only too common practice of not clearly defining the system under consideration. What mass of material, for example, does the dS of Equation [2] refer to? Analysis indicates that it is the increase in entropy in an infinitesimal heater per unit of throttle flow. Referring to Fig. 1 of this discussion, we get for the increase in entropy flux of an infinitesimal open heater.

$$dS_{\text{heater}} = (1 - w + dw)s_f - s_x dw - (s_f - ds_f)(1 - w)$$

After simplification and combination with the "heat balance," Equation [3] of the author's paper, this becomes

$$dS_{\text{heater}} = [H_x - h_f - T_f(s_x - s_f)]\, \frac{dw}{T_f} \quad \text{[7]}$$

which is identical with Equation [2] of the text. Note that in deriving this equation it is unnecessary to allude to fictitious "lost-work" effects.

Thermodynamically, the chief criticism of the writer is that the

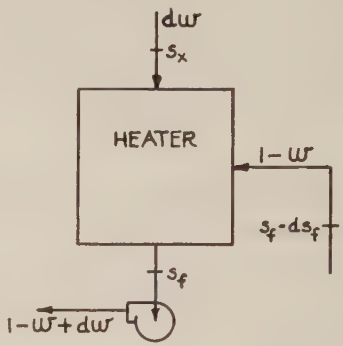


Fig. 1 Flow Diagram for Infinitesimal Heater

author identifies the value of $\Sigma dS_{\text{heater}}$ found from Equation [2] with the ΔS of Equation [7] of the paper. The writer sees no connection between this reasoning and the laws of thermodynamics.

The actual definition of the ΔS of Equation [7] of the paper resides in Equation [7] itself. For the heat rate of the cycle we may write

$$\text{Heat rate} = \frac{3412.75(H_C - h_A)}{H_C - h_A - (1 - w)_{\text{cond.}}\,(H_D - H_G)} \quad \text{[8]}$$

where the flow rate to the condenser (based on unit flow rate to the throttle) is found through integration of Equation [3] of the author's paper, to wit

$$(1 - w)_{\text{cond.}} = e^{-\int_{\text{cond}}^{\text{throttle}} \frac{T_f ds_f}{H_x - h_f}} \quad \text{[9]}$$

If Equations [8] and [9], herewith, are compared with Equation [7] of the author's paper, the exact definition of ΔS is found to be

$$\Delta S_{\text{exact}} = (s_C - s_A)\left[\frac{(s_C - s_G)/(s_C - s_A)}{e^{\int_{\text{cond}}^{\text{throttle}} T_f ds_f/(H_x - h_f)}} - 1\right] \quad \text{[10]}$$

Equation [6], giving the author's formula for ΔS, becomes, after a slight rearrangement

$$\Delta S_{\text{Hansen}} = (s_C - s_A)\left[\log_e \frac{s_C - s_G}{s_C - s_A} - \int_{\text{cond}}^{\text{throttle}} \frac{T_f ds_f}{H_x - h_f}\right] \quad \text{[11]}$$

A comparison of Equation [10] with Equation [11] of this discussion shows that the author's formula for ΔS is correct only when

$$\log_e \frac{s_C - s_G}{s_C - s_A} = \int_{\text{cond}}^{\text{throttle}} \frac{T_f ds_f}{H_x - h_f} \quad \text{[12]}$$

or, in other words, only when ΔS is identically zero.

Equation [12] is not a thermodynamic identity, and can at best be satisfied only approximately. It appears that for the cases given in the author's Table 2, the agreement is fairly good.

[9] "Notes on the Comparison of Steam Turbine Efficiencies," by Ernest L. Robinson, *General Electric Review*, vol. 29, 1926, pp. 503–510.

[10] Assistant Professor of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Jun. A.S.M.E.

Taking, for example, the steam conditions of Table 1, we get

$$\log_e \frac{s_C - s_G}{s_C - s_A} = 1.006; \quad \int_{\text{cond}}^{\text{throttle}} \frac{T_f ds_f}{H_x - h_f} = 0.977^*$$

Since the author's method apparently rests on the fortuitous fact that Equation [12] is approximately satisfied when steam is the working fluid, an important consideration concerns the maximum error which might be incurred through its use. For example, is one likely to encounter considerably greater errors than the largest error in Table 2, viz., 7 Btu per kwhr?

As regards the author's claim that his method requires less labor of computation than that of Selvey and Knowlton, attention is called to the writer's discussion of the Selvey and Knowlton paper. In that discussion it was shown that Selvey and Knowlton's procedure could be improved slightly as regards the thermodynamic analysis itself, and improved in considerable measure as regards computation technique. The method employed in that discussion was essentially equivalent to integrating Equation [9] of the present discussion with the aid of Simpson's rule and then substituting in Equation [8] of the present discussion to solve for the heat rate. With steam conditions of 3200 psia, 1200 F, 1 in. Hg, it was found that with only four calculations similar to those of Table 2 in the present paper, heat rates could be calculated with an accuracy of about 1 or 2 Btu per kwhr. With eight such calculations, the accuracy of the method is of the order of 0.1 or 0.2 Btu per kwhr.

H. Le H. Smith.[11] The author's three-dimensional consideration of temperature-entropy-quantity of steam is of special interest. Some of the general treatment raises questions of rigor, where fluid quantities are spoken of when fluid rates of flow are seemingly meant.

In dealing with adiabatic expansion and the resulting theoretical cycles, the treatment does not come to grips with reality. The author says, "Usually engineers are reluctant to utilize the concept of entropy except when it remains constant, ..." Can these things be?

The problem of the heat-cycle loss when superheated steam is bled for feedwater heating is an important problem, but surely its importance in a realistic sense is in application to real turbines, expanding nonadiabatically.

Toward the end of the paper explicit statements are made concerning the procedures by which the method could be modified to transfer it from the status of dealing with an unrealistic hypothetical turbine to that of dealing with a real turbine.

It would be an enlightening contribution if the author would make this transfer in a supplementary paper.

Author's Closure

The suggestion by Lieutenant Elrod eliminates the approximation to which objection is made in two of the discussions, without adding appreciably to the work involved. However, the error introduced by the approximation is negligible at any temperature likely to be employed in practice for some time to come, as may be shown by solving Equation [5] of the discussion for ΔS, letting A equal the area found graphically or otherwise.

$$\Delta S = (S_C - S_A)(e^A - 1) \ldots\ldots\ldots\ldots [13]$$

The exponential may then be expanded into an infinite series, by means of Maclaurin's theorem.

$$e^A = 1 + A + \frac{A^2}{2!} + \frac{A^3}{3!} + \ldots + \frac{A^n}{n!} \quad [14]$$

The approximation employed in the paper amounts to rejecting all terms beyond the second. If the value of A so found is of the order of 0.03, as in the example worked out in Table 1, the error introduced through letting $(e^A - 1)$ equal A, amounts to 0.00045, which would affect the heat rate by approximately 2 Btu per kwhr. If the value of A is 0.0014, as in the third line of Table 2, the error amounts to 0.000001, which would affect the heat rate by less than 0.01 Btu per kwhr. There is thus no justification for assuming that the discrepancies shown in Table 2 are due to this "approximation." The author has solved Equation [9], using Simpson's rule with eight intervals, and obtained a heat rate at 400 psia 700 F throttle conditions of 8228.3 Btu per kwhr. This indicates that a major part of the discrepancy referred to may be attributable to inaccuracy in the Selvey and Knowlton value of 8222.

Solution either of Equation [6] of the text or of Equation [13] just given is facilitated by the use of Simpson's rule. The following have been computed from data given in Table 1, by plotting the figures in column 10 and reading values at desired intervals.

No. of intervals	A	ΔS	Heat rate, Btu/kwhr
2	0.0284	0.0156	5999
3	0.0288	0.01585	6000.4
4	0.0288	0.01585	6000.4
8	0.0290	0.01595	6001.2

Three intervals[12] are sufficient to obtain an accuracy within one Btu per kwhr, compared with eight necessary if Equation [9] is used. At throttle temperatures below 1000 F, probably two intervals would prove ample. A still rougher approximation would be

$$A = \frac{S_A - S_G}{3}\left[\frac{1}{S_C - S_A} - \frac{T_A}{H_C - h_A}\right] \ldots\ldots [15]$$

The derivation in the paper is predicated on one pound of throttle steam. The time interval involved does not enter the discussion, hence it is unnecessary to employ terms indicating flow rate.

The manner in which reheat may be handled, merely by addition of the ΔS for each expansion, seems to the author exceedingly simple. Commander Dusinberre's observations regarding turbine and pump work are apt. However, it does not appear that theoretical pump work can be of much value, inasmuch as actual feed-pump work usually bears little relation to the theoretical.

It has been pointed out that much remains to be done before a rapid method of estimating real plant heat rates with high accuracy can be made available. The intention in the present paper has not been to indicate exactly how this may be done. It has rather been to clarify the concept of regeneration in the superheat region as an irreversible process, as implied by the title chosen, with the particular purpose of showing how quantitative results may be obtained.

One further step may be taken in the direction of actuality by making an analysis similar to Lieutenant Elrod's, but considering S_X to be variable, and letting feed-pump efficiency be m (assuming motor-driving the feed pump to have 100 per cent efficiency). This leads to the following

$$\Delta S = (S_C - S_A)(e^{(a + b + c)} - 1) \ldots\ldots\ldots [16]$$

* The value 0.977 is taken from the writer's discussion of the Selvey and Knowlton paper, author's reference (3).

[11] Supervisor, Power Department, B.M.T. Division, Board of Transportation, City of New York. Mem. A.S.M.E.

[12] Statement of Simpson's rule given in most handbooks applies only for an even number of intervals; with three intervals, y being the value in Table 1, column 10

$$\text{Area} = \frac{S_A - S_G}{8}(y_0 + 3y_1 + 3y_2 + y_3)$$

in which

$$a = -\int\left[\frac{1}{S_x - S_f} - \frac{T_f}{H_x - h_f}\right] dS_f \qquad [17]$$

$$b = \int \frac{dS_x}{S_x - S_f} \qquad [18]$$

$$c = -\frac{1-m}{m}\int \frac{v_f dp}{H_X - h_f} \qquad [19]$$

All integrals are of course between limits of throttle and condenser pressure. (Statement in paper that "ΔS value because of irreversibility in turbine expansion can be read from the expansion curve" is obviously incorrect, as such a value would not take into account the reduced quantity of steam reaching condenser.) Of the three components of the exponent, only *b* would be comparable in magnitude with the exponent to be evaluated if Equation [9] is employed, being of the order of half of the latter. Using Simpson's rule, not more, and probably fewer, than eight intervals would be required. If the steam flow is wanted, it can be found from the following

$$(1 - w) = \left[\frac{T_G(S_C - S_A)}{H_D - h_G}\right] e^{(a+b+c)} \qquad [20]$$

The inference may be drawn that other factors could be devised to be added to the exponent in Equations [16] and [20], to cover other departures from complete reversibility.

Development of the Lysholm-Smith Torque Converter[1]

Author's Closure

The author regrets that owing to wartime conditions, he has not been able to close the paper before.

Mr. Eksergian agrees with the author that the flow of a multistage reaction converter increases with reduced speed. The influence of this increased flow on the input torque is, however, not correctly interpreted by Mr. Eksergian. By using his theory, the input would increase on both sides on the optimum point, which is not substantiated by the test results. For the converter in question, the last turbine stage acts like a guide to the pump impeller giving a counterrotating vortex. The higher the secondary speed is, the less the circulation of this vortex will be, causing an increased unloading of the impeller. It has also been confirmed by tests made by the author that if a stationary guide is arranged before the pump impeller, the input torque will be substantially constant near the optimum point but, due to the increased flow decreases somewhat at stalling and racing. The dimensions of the various parts of the converter may be obtained from Fig. 1, which is drawn to a scale of 1:6.7.

Mr. Wislicenus comes to the conclusion that by using the equation given below Fig. 2, the actual pressure drop will be twice as great as derived from Equation [1]. This is not correct, as Equation [1] applies both to guide and rotating blades, thus making up for the missing factor of 2. The author regrets that this was not stated in his paper.

The author agrees that it is very difficult to separate losses in a hydraulic converter. The approximate method used when computing Table 1 was to estimate leakage, rotation, and mechanical losses according to usual formulas and try to balance the remaining losses in a reasonable way to correspond to the total losses obtained by the tests.

The carry-over losses are the losses in the open space between turbine and pump, or vice versa.

The author's Equation [2] was taken over from a similar equation used for calculating losses of the Ljungström double-rotation steam turbine. As shown in Fig. 4, this factor is by no means constant, but varies considerably over the range of inlet angles.

The rate of flow indicated in Fig. 2 is valid for constant input horsepower—that is to say, a constant value of $Q\Sigma\Delta h$. The values have been computed from test data for a great number of primary and secondary speeds by means of a Pitot tube fitted after the guide blade *C* in Fig. 1. This characteristic has to be accepted as correct on account of these tests, but it is also possible to confirm the test results by the cut-and-try method described below Fig. 2 in the paper.

A. Lysholm.[2]

[1] By A. Lysholm, published with discussion in the July, 1944, issue of the Transactions of the A.S.M.E. Because of wartime conditions the author was unable to submit a closure.

[2] Consulting Engineer, Stockholm, Sweden.

Experimental Study of the Flow of Coal in Chutes at Riverside Generating Station[1]

R. F. Legget.[2] This lucid account of an exhaustive experimental study into a complex operating problem has proved of unusual interest to the writer since he was working on an allied problem at the University of Toronto, apparently simultaneously with the authors of the paper. His findings, which to some extent supplement those of the authors, will be presented at a future date.

The writer's investigation was concerned with the clogging of bituminous coal in the large reinforced-concrete bunkers of the new steam-generating plant of the Polymer Corporation at Sarnia, Ontario. A model of a bunker was used, one-twelfth full size, but made of wood and not of such a convenient material as pyralin. The grading of the coal used did not correspond with that shown in Fig. 1 of the present paper; however, the coal used for experiments was modified as the authors suggested.

In all the experiments conducted at Toronto, no "scale effect" was noted, and the writer is therefore puzzled by the authors' statement with regard to this matter, especially since it is difficult to see what should cause a difference between the behavior of coal in model and prototype. Possibly the authors will discuss this matter further in their closure.

The comment given on the use of "wetting agents" confirms the writer's experience; he cannot but think, however, that in some way these remarkable chemical products may one day assist in the solution of problems involving the movement of coal.

In view of the differing character of the units being studied, it is the authors' interesting discussion of the physical properties of the coal which provides the closest link between the two investigations. Although the approaches to the study of the properties of coal were somewhat different, the moisture content is paramount in both. Accordingly, it would be helpful if the authors would state how their moisture contents are expressed, i.e., as percentages of the dry or of the wet weights of coal.

[1] By E. E. Wolf and H. L. von Hohenleiten, published in the October, 1945, issue of Trans. A.S.M.E., vol. 67, pp. 585–599.

[2] Associate Professor of Civil Engineering, University of Toronto, Toronto, Ontario, Canada.

Failure of Ductile Metals in Tension[1]

By G. SACHS[2] and J. D. LUBAHN,[3] CLEVELAND, OHIO

The extensive fabrication of aluminum-alloy sheet in the aircraft industries has revealed the lack of fundamental knowledge regarding the phenomena which explain the performance of metals under various stress and strain states. Previous investigations appear to have neglected entirely both the effects of the part geometry and the fact that the strength of the metal and limit of forming are frequently determined by a process of instability, or "necking," rather than by the process of fracturing. In this paper a theory of necking is developed in order to predict the forming limits for various stress or strain states based upon an analysis of the true stress-strain curve in pure tension. A criterion for necking is advanced, based upon the maximum-load (or pressure) conception of instability, and is applied to different geometrical shapes formed at room temperature. Future publications will present, in support of the criterion advanced, experimental analyses of the forming of different geometrical shapes under various loading conditions and stress and strain states.

Nomenclature

The following nomenclature is used in the paper:

A = cross-sectional area

e = ordinary principal strain = $\dfrac{L - L_0}{L_0}$

F = tensile load

F_{max}/A = stress at maximum load point

L = length of longitudinal element

p = internal pressure in tube, sphere, or circular bulge

R = radius of curvature to inside surface of thin-walled tube, sphere, or circular bulge

s = principal stress

h = thickness of element, or wall thickness of a tube or sphere

V = volume of element

w = width of element

ϵ = natural strain or logarithmic strain = $\ln(1 + e)$

0 = subscript denoting conditions before straining

1 = subscript denoting direction of principal stress or strain of greatest algebraic value

2 = subscript denoting direction of principal stress or strain of intermediate algebraic value

3 = subscript denoting direction of principal stress or strain of least algebraic value

Introduction

The general term "forming limit" may cover a variety of fundamentally different metal characteristics. It generally defines the maximum strain, or rather the magnitude and state of strain, which may be achieved with a certain type of equipment. This forming limit may be determined by the "cohesive" strength or "ductility" of the metal, which is often considered to be the only forming limit. However, a process may be limited by the strength of the equipment rather than by the ductility of the metal. In many commercial processes, particularly those of the compression type, the maximum strain is limited only in this manner.

Another rather peculiar type of forming limit results from a condition of loading which causes a local breakdown of the metal. A generally known and much discussed group of phenomena of this type is called "buckling." The buckling of a slender structural member limits its strength in a manner quite comparable to a brittle failure. Besides those phenomena of buckling discussed in textbooks on elasticity, there is also a variety of types of plastic buckling known which limit the forming under certain stress states, such as the wrinkling of deep-drawn cups and the folding of sunk or necked tubing.

The general characteristics of buckling may be defined as a load condition where an increase in strain occurs without an increase or with a decrease in load. Such an instability, therefore, should occur always when the load F goes through a maximum, or when the differential of the load becomes zero

$$F = \text{maximum} \qquad dF = 0$$

This condition determining the forming limit in buckling also applies to another group of instability phenomena, the best known representative of which is the local "necking" of a part strained in tension. When the load passes through a maximum, and the strain increases by a small amount at some location, this location becomes less strong in comparison with the remainder of the part. Then all further strain will occur here while it will discontinue at other locations, thus resulting in a "neck." The necking of a regular tensile-test specimen has attracted considerable interest, and numerous publications deal with stress-strain conditions conducive to necking (1–7).[4]

The condition of necking for pure tension can be written as follows:

$$dF = d(As) = 0$$

$$\frac{ds}{s} + \frac{dA}{A} = 0$$

where s is the true stress corresponding to a cross-sectional area A. This rather lucid formulation of the necking condition relates the instability to the fact that the rate of increase in tensile load by the increase of stress (due to strain-hardening) is no longer able to compensate for the decrease in tensile load be-

[1] This paper is one of a series of reports on the research program on hot-forming of aluminum alloys conducted at Case School of Applied Science under contract with the Office of Production Research and Development of the War Production Board. This research, which was supervised by the War Metallurgy Committee under the "restricted" project NRC-547, is the basis of this paper which has been released for publication by the OPRD.

[2] Professor of Physical Metallurgy, Department of Metallurgical Engineering, Case School of Applied Science. Mem. A.S.M.E.

[3] Department of Metallurgical Engineering, Case School of Applied Science.

Contributed by the Metals Engineering Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

cause of decreasing cross-sectional area. The necking condition in this form has been applied to processes other than uniaxial tension, such as a sheet subjected to biaxial tension (13), a rotating disk, and a tube subjected to internal pressure (8, 9, 11).

In order to develop an instability condition that is applicable to the wide variety of commercial conditions of loading and geometry, the original maximum-load conception must be extended. Already it can be concluded from previous analyses that the instability of a flat sheet and of a tube subjected to the same stress state occur at different strains. Therefore the conception is advanced in this paper that instability occurs, or "necking" begins, if, with increasing strain, one of the external loads passes through a maximum. So far only the tensile load has been considered ($dF = 0$). However, the primary source of stress in parts subjected to internal pressure is this pressure p and consequently instability should then occur when

$$p = \text{maximum} \qquad dp = 0$$

The purpose of this and subsequent papers will be as follows:

1 To develop the conception that under various load and geometrical conditions the forming limit for a homogeneous metal may be determined by instability, and to associate this "necking" with a maximum in either the tensile load or the internal pressure, as discussed previously.

2 To develop a theory of instability which may be applied to the stress-strain curve as experimentally derived for the load conditions and geometrical conditions actually present.

3 To extend this criterion of instability to predict the forming limit for any stress state and condition of geometry considered, using only the stress-strain curve in pure tension.

In the present analysis equations have been developed which apply to actual relations between the stresses and strains that exist during a particular type of test on a particular material. These relations, which are necessary in order to predict the necking strain, may be determined experimentally in a simple manner for certain conditions of geometry and loading, such as a tube subjected simultaneously to internal pressure and longitudinal tension. In this case the stresses and strains are calculated readily from the measured load, the pressure, and the dimensional changes, such as the changes in length and diameter of a tube. However, in other instances of geometry, such as a circular hydraulically formed bulge, a tedious experimental analysis and possibly also a theoretical analysis is necessary to establish the stress-strain relations that are required for the application of the derived necking condition.

Experimentation generally supplies the three principal stresses and the three principal strains for each point of the test, thus defining completely the stress and strain states, respectively. To apply the graphical construction, particular stresses and strains, or functions of these, must be selected to yield a suitable stress-strain relation. It would be desirable that various stress states could be represented by a single, or "universal," stress-strain curve, by plotting a suitable function of the three principal stresses versus a suitable function of the three principal strains. There are several possibilities, and the question as to which is the most suitable is still a matter of considerable argument. In most problems of plasticity the simplest possible condition has been found to yield sufficiently accurate results.

The following analysis applies only to biaxial-tension stress states, defined by the fact that the algebraically smallest principal stress s_3 is zero. In this case the larger of the two tensile stresses s_1, may be plotted versus the numerically largest of the three natural principal strains, as the simplest possible conception that yields a nearly universal stress-strain relation.[5]

Other stress and strain functions yield stress-strain curves for various stress states which agree more closely, but also which require more elaborate calculations. It should be noted that two of the stress states which are considered particularly in this paper, namely, pure tension and balanced biaxial tension, yield identical stress-strain curves, according to any of the recognized theories. The stress state which may be particularly affected regarding the accuracy of the solution by any such assumption would be the condition of plane strain which is intermediate between pure tension and biaxial tension.

The present approach to the problem has the particular advantage that the resulting differential equations allow the direct determination of the necking point on the experimental or equivalent stress-strain curve, using a simple geometrical construction. Also, the equations derived in this paper are basic, and may be extended to include any desired universal stress-strain relation. Finally, the stress may be approximated algebraically as a function of the strain, to permit an algebraic rather than a graphical solution. However, considering the rather large amount of work which such an algebraic analysis requires, it appears that the graphical method of solution is more flexible, is more lucid, and offers less possibility of error; consequently, this will be used in the analyses in subsequent publications.

Regarding the effect of the part geometry on the instability strain for balanced biaxial tension, it will be shown that the results are rather different for the three types of curvature, or changes in curvature, represented by (*a*) a flat sheet, (*b*) a sphere under internal pressure, or tubing under longitudinal tension and internal pressure, and (*c*) a circular hydraulic bulge formed by pressure from flat sheet. Previous experimentation confirms the results of some phases of the theoretical analysis, and additional evidence will be submitted in subsequent publications.

Necking of a Tensile Test Specimen

The well-known necking of a ductile tensile test specimen has been recognized by Considére (1) and Ludwik (2), as a phenomenon of instability. It is associated with a maximum in the load-strain diagram. As long as the load F increases, the specimen stretches approximately uniformly over its length,[6] while a decreasing load indicates necking.

(*a*) If s_1 is the true principal stress, i.e., the load F divided by the actual cross section A of the metal, the beginning of necking is given by the condition (see Sachs, 7)

$$F = s_1 A = \text{maximum}$$

$$dF = 0 = A ds_1 + s_1 dA$$

$$\frac{ds_1}{dA} = -\frac{s_1}{A} \qquad [1]$$

As illustrated in Fig. 1, the point of necking, or maximum load point, is a point in the s_1 versus A diagram, such that the tangent at this point intersects the ordinate at $A = 0$ at a height

[5] This is a special case (for biaxial tension) of the conception that all stress states yield identical stress-strain curves if the maximum shear stress $\left(\frac{s_1 - s_3}{2}\right)$ is plotted versus the maximum (absolute) natural strain.

[6] The "uniform" strain is considered as the presumably constant value of strain encountered at room temperature outside necked areas along the length of a sufficiently long tensile specimen. This value represents the greatest strain the metal will exhibit without necking under ideal conditions.

$\frac{2F_{max}}{A}$, giving a slope value of $-\frac{s_1}{A}$ (see footnote[7]).

This solution becomes particularly simple if the s_1 versus A diagram is a straight line in the vicinity of the maximum load point, as shown by Koerber (4), and pointed out previously by Moellendorff and Czochralski (3); see also Koerber and Rohland (6).

(*b*) The strain may also be expressed by the stretch e_1 of a certain length, $L = L_0\,(1 + e_1)$, determined by the constancy of volume V

$$V = L_0A_0 = LA = L_0\,(1 + e_1)A$$

$$1 + e_1 = \frac{A_0}{A}$$

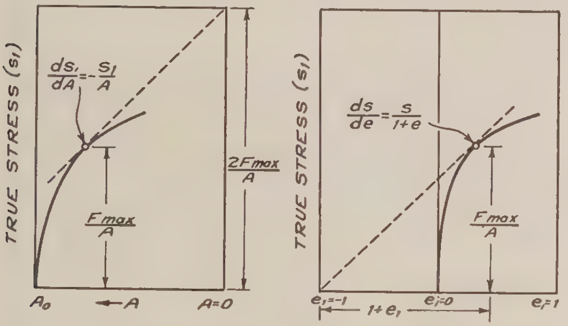
where the subscript zero denotes conditions before straining. The condition of necking then becomes

$$F = A_0\left(\frac{s_1}{1 + e_1}\right) = \text{maximum}$$

$$dF = A_0\left[\frac{ds_1}{1 + e_1} - \frac{s_1de_1}{(1 + e_1)^2}\right] = 0$$

$$\frac{ds_1}{de_1} = \frac{s_1}{1 + e_1} \qquad \text{[1a]}$$

In the s_1 versus e_1 diagram, Fig. 2, the maximum load point is



FIGS. 1 AND 2 CONSTRUCTION OF THEORETICAL NECKING CONDITION IN PURE TENSION FOR s VERSUS A AND s VERSUS e CURVES

determined by the condition that its tangent shall intersect the abscissa at a value $e_1 = -1$, yielding a slope equal to $s_1/(1 + e_1)$, see Considére (1), Nielsen (5), and Gensamer (13).

(*c*) By a third method which will be used generally throughout this paper, the strain can be expressed as the natural strain

$$d\epsilon_1 = \frac{de_1}{1 + e_1}$$

The necking condition then can be written (see Equation [1*a*])

$$\frac{ds_1}{d\epsilon_1} = s_1 \qquad \text{[1b]}$$

As illustrated in Fig. 3, the maximum load point may be determined by the condition that its tangent intersects the abscissa of the s_1 versus ϵ_1 diagram at the value $-(1 - \epsilon_1)$, yieldi :g the slope $s_1/1$.

[7] In the graphical solution for a particular metal, using experimental data, it is more accurate actually to plot the slope (ds_1/dA) and the ratio (s_1/A) against A and note their intersection (see also Fig. 9), rather than use the type of construction indicated in the various figures.

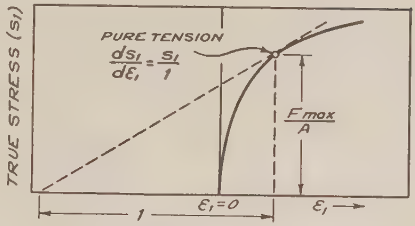


FIG. 3 CONSTRUCTION OF THEORETICAL NECKING CONDITION IN PURE TENSION FOR s VERSUS ϵ CURVE

NECKING OF FLAT SHEET UNDER BIAXIAL TENSION

When a flat sheet is subjected to biaxial tension, the normal stress s_3 is zero, and the maximum shear-stress condition indicates that the largest principal stress s_1 (2 times the largest shear stress) is always the decisive stress.

Under these conditions, the stress-strain curve becomes very similar to that for uniaxial tension if this decisive stress is plotted against the decisive strain, i.e., the numerically largest principal strain. Considering the intermediate natural strain ϵ_2, two general cases of biaxiality arise. When ϵ_2 is less than zero, ϵ_1 remains the decisive strain; when ϵ_2 is positive (but smaller than ϵ_1, according to definition), ϵ_3 becomes the decisive strain.

(*a*) The general condition of instability (necking) for a flat sheet will now be postulated to be the occurrence of a maximum in the force applied in the direction of the maximum tension s_1, this resulting in the same equation as for uniaxial tension Equation [1]

$$F = s_1A = \text{maximum}$$

$$dF = 0 = s_1dA + Ads_1$$

$$\frac{ds_1}{s_1} = -\frac{dA}{A} \qquad \text{[2]}$$

Using the constant-volume condition

$$\left.\begin{aligned} A &= A_0\,(1 + e_2)\,(1 + e_3) \\ dA &= A_0\,(1 + e_3)de_2 + A_0\,(1 + e_2)de_3 \\ \frac{dA}{A} &= \frac{de_2}{1 + e_2} + \frac{de_3}{1 + e_3} = d\epsilon_2 + d\epsilon_3 \end{aligned}\right\} \qquad \text{[3]}$$

The general instability condition, Equation [2], becomes

$$\frac{ds_1}{s_1} = -\,d\epsilon_2 - d\epsilon_3 \qquad \text{[3a]}$$

$$\frac{ds_1}{s_1} = d\epsilon_1 \qquad \text{[3b]}$$

(*b*) Strain ϵ_1 remains decisive for any biaxiality where $0 > \epsilon_2 > -\frac{\epsilon_1}{2}$, the limiting conditions being uniaxial tension ($\epsilon_2 = -\epsilon_1/2$) and plane strain ($\epsilon_2 = 0$). For this whole range of biaxiality, therefore, the same condition Equation [3*b*], applies as that for uniaxial tension

$$\frac{ds_1}{s_1} = d\epsilon_1$$

For the case of plane strain, Baranski (10) has observed previously that the natural necking strain is practically the same as for pure tension.

(*c*) For biaxialities where $\epsilon_1 > \epsilon_2 > 0$, ϵ_3 becomes the decisive

strain, the limiting conditions being plane strain ($\epsilon_2 = 0$) and balanced biaxiality ($\epsilon_1 = \epsilon_2 = -\epsilon_3/2$).

Introducing the quantity n, defined by

$$\epsilon_2 = -n\epsilon_3 \qquad [4]$$

where $1/2 > n > 0$ (for ϵ_3 being decisive), the basic Equation [3a] becomes

$$\frac{ds_1}{s_1} = -d\epsilon_3(1 - n) \qquad [3c]$$

This equation can also be solved graphically.

(*d*) In the special case of balanced biaxial tension, $\epsilon_2 = -\epsilon_3/2$, $n = 1/2$, and the instability condition assumes the following form

$$\frac{ds_1}{s_1} = -\frac{1}{2}\, d\epsilon_3 \qquad [5]$$

or

$$\frac{ds_1}{d\epsilon_3} = -\frac{s_1}{2} \qquad [5a]$$

As shown by Gensamer (13), the graphical solution of this equation can be obtained by the same method as employed in Fig. 3 and illustrated in Fig. 4.

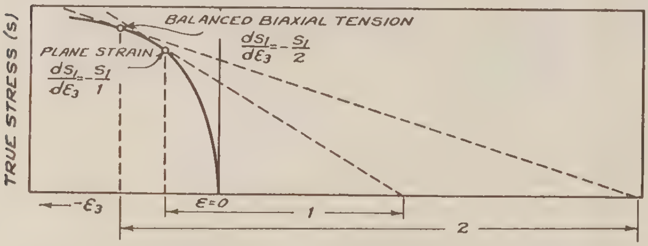


FIG. 4 CONSTRUCTION OF THEORETICAL NECKING CONDITION IN BIAXIAL TENSION FOR s VERSUS ϵ CURVE

(*e*) In the limiting case of plane strain, $\epsilon_2 = 0$, $\epsilon_3 = -\epsilon_2$, $n = 0$, the two solutions, Equations [3b] and [3c], yield identical results (see Figs. 3 and 4)

$$\frac{ds_1}{s_1} = -d\epsilon_3 = d\epsilon_1$$

It can be seen that for identical stress-strain curves, as noted previously, ϵ_3 at the maximum load point of balanced biaxial tension is much larger than ϵ_1 at the maximum load point of uniaxial tension, Fig. 3. Thus the uniform stretch in balanced biaxial tension, $\epsilon_1 = -\epsilon_3/2$, must be greater than one half the uniform stretch in pure tension, and may be either larger or smaller than the uniform stretch in pure tension, depending upon the strain-hardening characteristics of the metal.

INSTABILITY OF THIN-WALLED TUBES SUBJECTED TO INTERNAL PRESSURE

The condition for the necking of a thin-walled tube subjected to internal pressure has not been thoroughly understood up to the present time. Some attempts by Laszlo (8, 9) and Siebel and Kopf (11) indicate that such a tube will neck at considerably smaller strains than a tensile test specimen of the same material.

It appears that the uniform strain of a tube subjected primarily to an internal pressure p, is determined by the condition that this pressure reaches a maximum. For a thin-walled tube having an initial inside diameter $2R_0$ and an initial wall thickness h_0, this pressure is given by the well-known equation

$$p = s_1 \frac{h}{R} = s_1 \frac{h_0\,(1 + e_3)}{R_0\,(1 + e_1)} \qquad [6]$$

where e_1 and e_3 are the circumferential and normal strains, respectively.

This condition should apply for any condition of biaxiality where the circumferential stress is the decisive stress (and possibly also for conditions where the longitudinal stress is the decisive stress).

Regarding the decisive strain, again two conditions arise, where $0 > \epsilon_2 > -\frac{\epsilon_1}{2}$, and where $\epsilon_1 > \epsilon_2 > 0$.

(*a*) The general condition of instability, Equation [6], can now be written as follows

$$dP = 0 = \frac{h_0}{R_0}\left[ds_1 \frac{1 + e_3}{1 + e_1} + s_1 \frac{de_3}{1 + e_1} - s_1 \frac{de_1\,(1 + e_3)}{(1 + e_1)^2}\right]$$

This yields the final equation

$$\frac{ds_1}{s_1} + \frac{de_3}{1 + e_3} - \frac{de_1}{1 + e_1} = 0 \qquad [6a]$$

or

$$\frac{ds_1}{s_1} + d\epsilon_3 - d\epsilon_1 = 0 \qquad [6b]$$

(*b*) For pure (uniaxial) circumferential tension, ϵ_1 is the decisive strain, and $\epsilon_3 = \epsilon_2 = -\frac{\epsilon_1}{2}$. Consequently, the instability condition, Equation [6b], assumes the following form

$$\frac{ds_1}{s_1} = \frac{3}{2}\, d\epsilon_1$$

or

$$\frac{ds_1}{d\epsilon_1} = \frac{s_1}{2/3} \qquad [7]$$

This latter form can again be solved graphically by the method of trial and error, Fig. 5. A comparison with the condition for pure tension for which the same stress-strain curve exists, Equation [1b], shows clearly that the tube would become unstable at a smaller circumferential strain than the (longitudinal) necking strain of a tensile test bar. This difference is due to the change in curvature, rather than to differences in the stress state.

(*c*) If $0 > \epsilon_2 > -\frac{\epsilon_1}{2}$, the decisive strain remains ϵ_1. Introducing the relation

$$\epsilon_2 = -n\epsilon_1;\ \epsilon_3 = -\epsilon_1 - \epsilon_2 = (+n - 1)\epsilon_1 \qquad [8]$$

where $1/2 > n > 0$, the general condition of instability can be written as follows

$$\frac{ds_1}{s_1} = d\epsilon_1\,(2 - n) \qquad [6c]$$

(*d*) For plane strain, $\epsilon_2 = 0$, $n = 0$, and the instability condition then becomes

$$\frac{ds_1}{d\epsilon_1} = \frac{s_1}{1/2} \qquad [9]$$

The graphical solution for this case of biaxiality is also illustrated in Fig. 5. The circumferential strain at which the tube would become unstable, under such conditions, would be still smaller than that for a tube subjected to pure circumferential pressure.

(*e*) If $\epsilon_1 > \epsilon_2 > 0$, the decisive strain becomes ϵ_3. Using the relation of Equations [8]

$$\epsilon_1 = \frac{\epsilon_3}{n - 1} \qquad [8a]$$

where $0 > n > -1$, the general condition of instability can be written as follows (see Equation [6c])

$$\frac{ds_1}{s_1} = d\epsilon_3 \left(\frac{2 - n}{n - 1}\right) \qquad [6d]$$

(*f*) For balanced biaxial tension, $\epsilon_1 = \epsilon_2 = -\frac{\epsilon_3}{2}$, $n = -1$ and the instability condition yields the equation

$$\frac{ds_1}{d\epsilon_3} = -\frac{s_1}{2/3} \qquad [10]$$

The same graphical solution now applies as for pure circumferential tension, Fig. 5, the decisive strain, however, being ϵ_3. The circumferential strain, in the case of balanced biaxial tension, however, would be only one half of that for uniaxial circumferential tension, because of $\epsilon_1 = -\frac{\epsilon_3}{2}$. As noted previously, in balanced biaxial tension, approximately the same curve is obtained by plotting s_1 versus ϵ_3 as that obtained in uniaxial tension by plotting s_1 versus ϵ_1. Furthermore, Equations [10] and [7] apply, respectively, to these two curves in the same manner, yielding as the solutions identical values of necking strain (ϵ_3 and ϵ_1, respectively). In the case of balanced biaxial tension, however, the uniform stretch ϵ_1, which is only one half of the normal instability strain ϵ_3 is therefore also only one half of the uniform stretch in uniaxial circumferential tension.

Instability of Thin-Walled Sphere Subjected to Internal Pressure

The same approach as used for a tube should be valid when applied to a sphere subjected to an internal pressure, p. The pressure for a sphere having an initial diameter $2R_0$ and an initial wall thickness t_0, is given by the well-known equation

$$p = 2s_1 \frac{h}{R} = 2s_1 \frac{h_0}{R_0} \frac{1 + e_3}{1 + e_1}$$

where $s_1 = s_2$ is the tensile stress and $e_1 = e_2$ the tangential strain (in the surface of the sphere), and e_3 is the normal (and decisive) strain.

The instability condition then becomes

$$dp = 0 = 2\frac{h_0}{R_0}\left[ds_1 \frac{1 + e_3}{1 + e_1} + s_1 \left(\frac{de_3}{1 + e_1} - \frac{de_1\,(1 + e_3)}{(1 + e_1)^2}\right)\right]$$

or

$$\frac{ds_1}{s_1} + \frac{de_3}{1 + e_3} - \frac{de_1}{1 + e_1} = 0$$

or

$$\frac{ds_1}{s_1} + d\epsilon_3 - d\epsilon_1 = 0 \qquad [11]$$

And using the relation $\epsilon_1 = -\frac{\epsilon_3}{2}$, the same equation results as for tubing under balanced biaxial tension, Equation [10], the graphical solution of which is illustrated in Fig. 5.

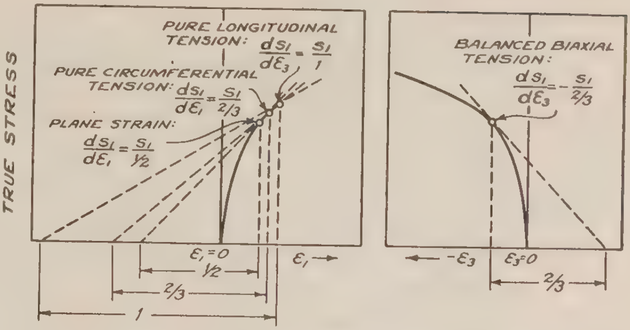


Fig. 5 Construction of Theoretical Necking Condition in Tubing Under Various Stress States

Instability of Circular Hydraulically Formed Bulge

The criterion for necking presented here is limited to uniform stress and strain states over the region in question. This requires that for any section of the specimen both the thickness and radius be uniform at any instant during the test, and that there be no variation of metal properties throughout the specimen.

During the bulging of a thin circular disk clamped at its periphery, neither the thickness nor the radius is uniform over the entire bulge. However, experimentation has shown that at and around the pole there is a rather large area which is close to spherical and where the magnitudes of the stresses and strains are nearly constant. Consequently, an analysis similar to that used for other geometrical shapes should be applicable. The validity of this conception has been verified by experiments which will be presented in a later publication.

Fundamentally, the same condition should determine the strength of a ductile disk, subjected to hydraulic pressure, as that which determines the strength of a sphere subjected to internal pressure, namely, that when the pressure p, reaches a maximum, instability should occur, provided that the metal is sufficiently ductile to "neck" before fracturing. This pressure p is given, as in the case of the sphere, by the equation

$$p = 2s_1 \frac{h}{R} = 2s_1 \frac{h_0(1 + e_3)}{R}$$

where $s_1 = s_2$ is the tension at the pole of the bulge, at which location the highest strains are exhibited, and $e_1 = e_2$ and e_3 are the tangential strains and normal strain, respectively, Fig. 6. The necking condition in terms of the strain e_3 then becomes

$$dp = 0 = 2h_0\left(ds_1 \frac{1 + e_3}{R} + s_1 \frac{de_3}{R} - s_1 dR \frac{1 + e_3}{R^2}\right)$$

or

$$\frac{ds_1}{s_1} = \frac{dR}{R} - \frac{de_3}{1 + e_3}$$

This equation can be written in the logarithmic form[8]

$$d(\ln s_1) = d(\ln R) - d\epsilon_3$$

or

$$\frac{d\,(\ln s_1)}{d\epsilon_3} = \frac{d\,(\ln R)}{d\epsilon_3} - 1 \qquad [12]$$

[8] ln = $\log_e$

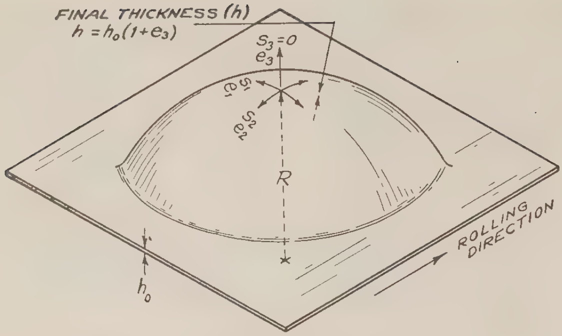


Fig. 6 Stress and Strain State and Geometrical Conditions in a Circular Bulge

This equation differs from that of a sphere because the radius of curvature in this case is an unknown function of the strain. No theory of bulging is yet available which would yield the curvature during bulging if the other quantity were known. Therefore, a graphical solution, based upon experimentally determined values of stress, strain, and curvature, is necessary. The stress-strain curve, s_1 versus ϵ_3, and the radius-strain curve, R versus ϵ_3, are necessary in order to determine the two functions

$$\frac{d\,(\ln s_1)}{d\epsilon_3} \text{ and } \frac{d\,(\ln R)}{d\epsilon_3}$$

The radius is determined experimentally from contour measurements. The normal strain ϵ_3 is calculated from the measured tangential strains $e_1 = e_2$ according to the equation

$$\epsilon_3 = \ln\left[\frac{1}{(1 + e_1)^2}\right]$$

The stress is determined from the experimentally measured quantities, radius, pressure, and tangential strain, related by the following equation

$$s_1 = \frac{pR\,(1 + e_1)^2}{2t_0}$$

These values can be employed in the graphical solution of Equation [12]. Thus, as shown in Figs. 7(*a*) and 7(*b*), ln R and ln s_1 are plotted as functions of ϵ_3, (ϵ_3 being negative). The values of

$$\frac{d\,(\ln R)}{d\epsilon_3} - 1 \quad \text{and} \quad \frac{d\,(\ln s_1)}{d\epsilon_3}$$

are determined from these curves and plotted as functions of ϵ_3, Fig. 7(*c*). The ϵ_3 value at the intersection then determines the strains at the necking point, according to

$$\epsilon_1 = \epsilon_2 = -\frac{\epsilon_3}{2}$$

This strain value is considerably higher than that for the sphere and usually also higher than that for plane biaxial stress, because of the decrease in radius R, with increase in strain.

Acknowledgment

The authors are indebted to Mr. G. Espey and Mr. W. F. Brown, Jr., of the Department of Metallurgical Engineering, Case School of Applied Science, for their valuable assistance and criticism of the original manuscript.

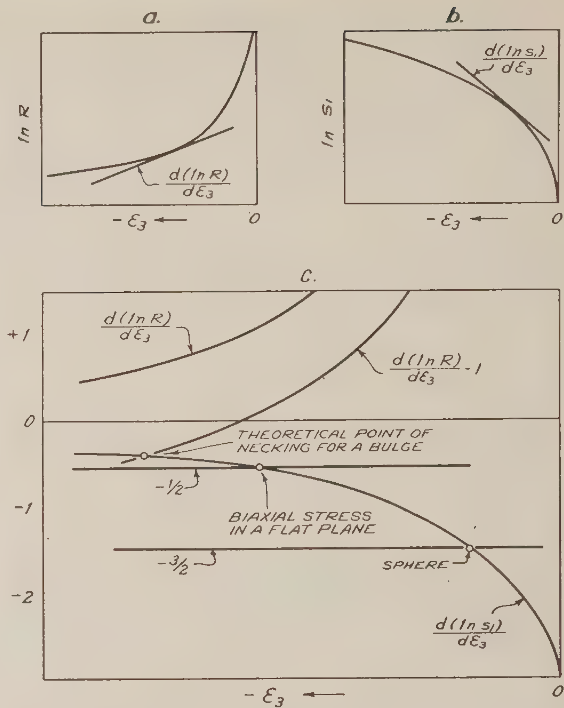


Fig. 7 Steps in Graphical Solution of Equation for Necking of a Circular Bulge [11], and Comparison of the Instability Strain With That for a Sphere and for a Flat Plane Subjected to Balanced Biaxial Tension

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The Necking of Tensile-Test Specimens[1]

By J. D. Lubahn,[2] Cleveland, Ohio

The conventional tensile test is the basis of acceptance tests, and its results are extensively used to evaluate the structural metals regarding their relative performance in forming and their strength in service. However, it appears that the significance of the phenomena encountered in a tensile test are little understood, in spite of a large number of investigations. In this paper, the strains encountered in long rectangular tensile-test bars of the aluminum alloys 24S-O and 24S-T have been investigated in detail. The theoretical conceptions previously advanced,[3] regarding the necking of tensile-test bars, have been confirmed experimentally.

The present theory of necking correlates the occurrence of a maximum in the external load with a change in the strain condition from a uniform strain state to a nonuniform strain state, i.e., local necking. The simplest example of uniform stress and strain conditions throughout the metal should be that encountered in a regular tensile-test bar which is subjected presumably to "pure tension" before necking occurs.

The validity of the necking condition for test bars strained in "pure tension" has been the subject of numerous investigations listed in a previous paper.[3] Recently, however, the existence of a really uniform strain in a tensile test bar has been considered as doubtful. It must be emphasized that commercial metals are not entirely uniform, and therefore their strains in pure tension will be nonuniform to an extent depending upon the variations in metal properties. Frequently, also, test bars and sections that are stretched commercially lack uniformity of cross section along their length and these variations likewise cause the strains to be nonuniform even before instability, or "necking," occurs.[4]

In addition to the foregoing considerations, the stress state of a standard tensile-test specimen at failure is rather complex as a result of two factors: (*a*) The transverse tension near the ends of the gage length, because of enlarged "heads" of the specimen; and (*b*) the transverse tension in the neck itself, which is in effect a notch of large radius of curvature. In a relatively short gage length, such as that of a standard tensile-test specimen, it has been observed that the regions where lateral contraction is restrained, corresponding to the two regions of lateral tension just described, may overlap. This makes it impossible to determine by strain analysis of a standard (American) tensile-test specimen any such strain as the uniform elongation, or the strain under conditions of uniaxial tension at which local necking begins.

Tensile-Test Procedure

In order to study the phenomenon of necking under conditions which avoid the difficulties mentioned, a few tensile tests were made on aluminum-alloy strips 20 in. long, of bare 24S-O and bare 24S-T sheet, Figs. 1 and 2. The strips were reduced in the center portion from a sheared width of $1^1/_2$ in. to a width of 1 in., in order to form a 13-in. gage length of constant width with edges free from nicks and cold work. The edges were prepared by milling to a depth at least equal to the thickness, followed by polishing on a belt sander. The longitudinal direction[5] of the sheet was selected as the direction of testing, in order to obtain the most nearly constant thickness and the most nearly uniform metal properties along the length of the specimen.

The specimens were subjected to tension loads that continuously increased at a very small rate. During tests, measurements of width and thickness were made at 24 stations spaced $^1/_2$ in. apart. By a process of interpolation, the measurements, and thus also the strains, were computed for the same point of time at all the stations. Thus in Figs. 1 and 2, each curve represents the strain distribution at one particular instant during the test. Such a representation illustrates the gradual change in shape of the strain-distribution curve as the test proceeds.

It is apparent from Figs. 1 and 2 that the strains are nearly constant with position along the length of the specimen up to a strain of approximately 8 per cent. With further increasing strain, small nonuniformities appear which are not local necks in the sense of an instability effect, because they do not represent points where the strain continues to increase in the absence of strain at other points. There is a certain maximum strain, however, which has been denoted as the necking strain, above which the strains continued to increase in only one localized region, called the "neck," Figs. 1 and 2. It can be seen that this necking strain was approximately 18.5 per cent for the investigated 0.040-in. 24S-O bare specimen, and 17.5 per cent for the 0.125-in. 24S-T bare specimen. Apparently, for 24S-T bare, the strain at a load of 8980 lb was the maximum-load strain, Fig. 2, since the strain after fracture differed only by the amount of elastic recovery that is to be expected for this alloy.

The curves of (true) stress versus reduction of cross-sectional area for the two materials were obtained by plotting (for one particular station) the cross-sectional areas corresponding to the various strain curves in Figs. 1 and 2 versus the corresponding stresses. The same stress-strain curve was obtained for various stations, thus indicating the correctness of the interpolation process for obtaining the strain-distribution curves. For the purpose of this investigation, the stress-strain curves were extended only to the largest strain encountered outside the region of fracture, to avoid introducing the effects of transverse tension in the neck. The curves so obtained were found satisfactory for the following analysis, in that they exceeded by a few per cent the theoretical point of necking.

[1] This paper is one of a series of reports on the research program on hot-forming of aluminum alloys conducted at Case School of Applied Science under contract with the Office of Production Research and Development of the War Production Board. This research, which was supervised by the War Metallurgy Committee under the "restricted" Project NRC-547, is the basis of this paper, which has been released for publication by the OPRD. The project was directed by G. Sachs, Case School of Applied Science.

[2] Department of Metallurgical Engineering, Case School of Applied Science.

[3] "Failure of Ductile Metals in Tension," by G. Sachs and J. D. Lubahn, published on pages 271–276 of this issue of the Transactions.

[4] The results of recent tests show that these variations are large, even under the most accurate conditions of material selection and machining. Variations in strain along the length of the specimen may be correlated directly with variations in initial cross section.

Contributed by the Metals Engineering Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[5] Tests on transverse specimens were abandoned because of excessive variations in thickness across the width of the rolled sheet.

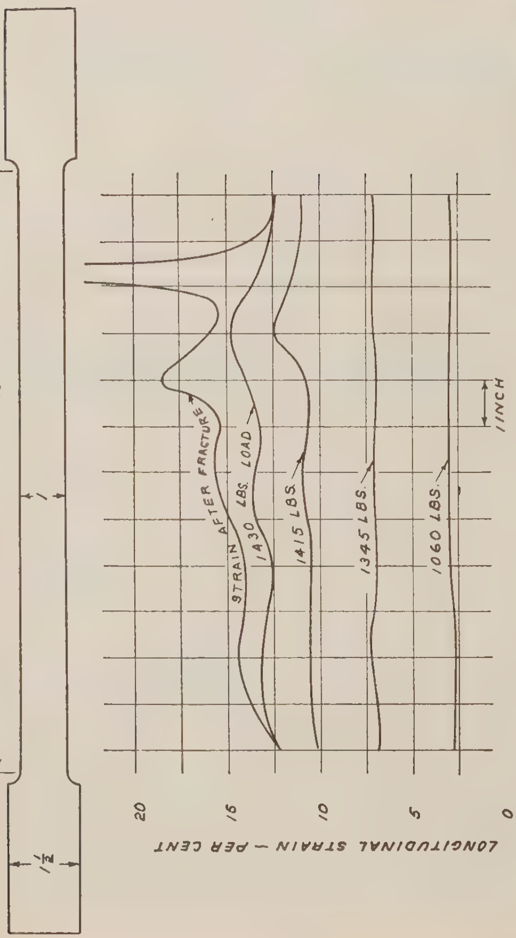


Fig. 1 Progressive Changes in Distribution of Longitudinal Strains in Strip Tensile Specimen Cut Longitudinally From 0.040-In. Bare 24S-O Sheet

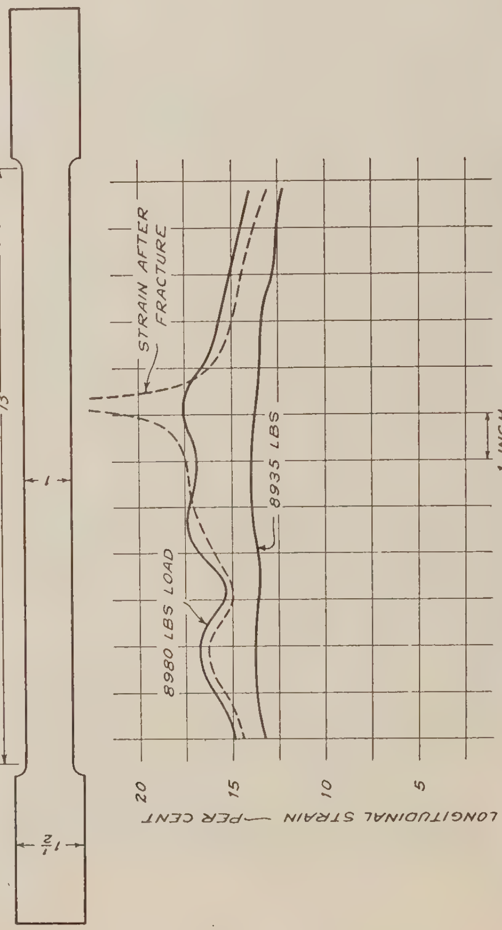


Fig. 2 Progressive Changes in Distribution of Longitudinal Strains in Strip Tensile Specimen Cut Longitudinally From 0.125-In. Bare 24S-T Sheet

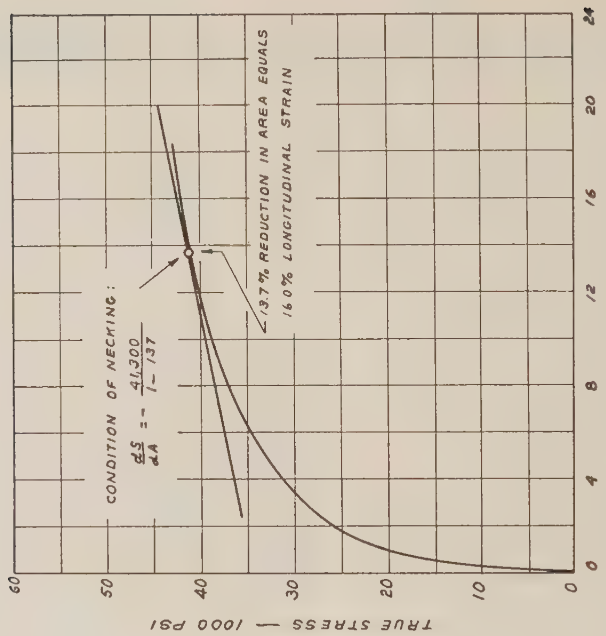


Fig. 3 Stress-Strain Curve for 0.040-In. Bare 24S-O Sheet, Showing Theoretical Condition for Necking

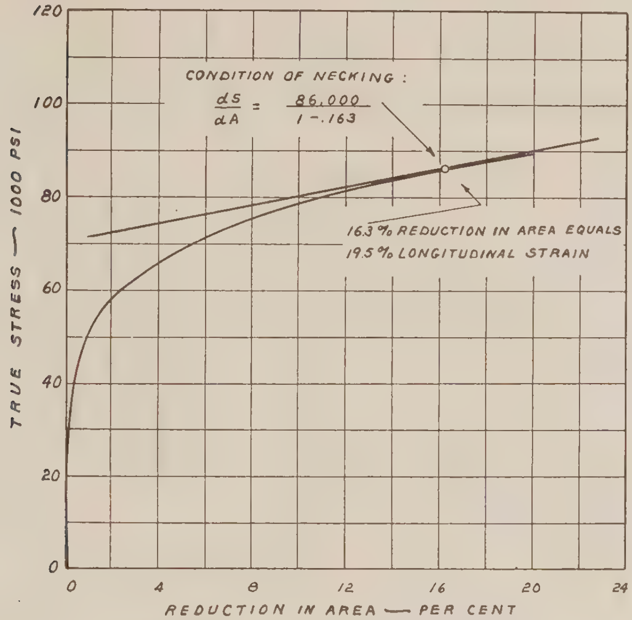


Fig. 4 Stress-Strain Curve for 0.125-In. Bare 24S-T Sheet, Showing Theoretical Condition for Necking

Analysis of Results

The stress versus reduction (of area) curves thus obtained are shown in Figs. 3 and 4, together with the construction lines for determining the strain at necking, according to the theory developed previously.[3] The values of longitudinal strains obtained from the necking theory for 24S-O bare, and 24S-T bare, are 16 and 19.5 per cent, respectively. These values are in good agreement[6] with the corresponding experimentally determined uniform strains of 18.5 and 17.5 per cent, respectively, Figs. 1 and 2.

The accuracy with which the strain at necking can be determined from a stress-reduction curve, such as that in Fig. 4, depends primarily upon the accuracy with which the slope of the stress-area curve is measured. In order to illustrate the probable extent of the scattering of the slope measurements, the necking condition has been solved, in Fig. 5, by intersecting the two functions s_1/A and $-ds_1/dA$. The curve shows that typical errors of slope measurements may result in errors in the strain at necking of approximately plus or minus 1 per cent.

[6] A larger number of subsequent tests show that the necking strain determined theoretically from the stress-strain curve is 1 or 2 per cent above that observed in the strain-distribution curve, and this phenomenon has not been explained as yet. It should be observed, however, that this difference is smaller than the rather large experimental errors associated with instability phenomena.

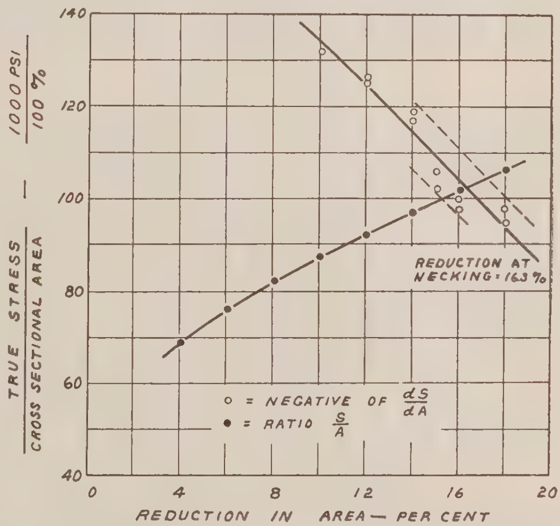


Fig. 5 Condition for Necking of 0.125-In. Bare 24S-T Sheet, Showing Equality of ds/dA and s/A, and Indicating Possible Errors Resulting From Inaccurate Slope Measurements

Instability of Thin-Walled Tubes Subjected to Internal Pressure[1]

By G. ESPEY,[2] CLEVELAND, OHIO

A preceding paper advances the criterion that the maximum-load or pressure conception of instability is applicable to geometrical shapes other than the tensile test. This paper presents experimental analysis to support the theoretical conception advanced for the necking of tubing subjected to pure longitudinal tension, to balanced biaxial tension, and to plane strain with the major strain being circumferential. It also shows that the stress-strain curve in uniaxial tension can be used to predict the necking strain for tubing under various stress states.

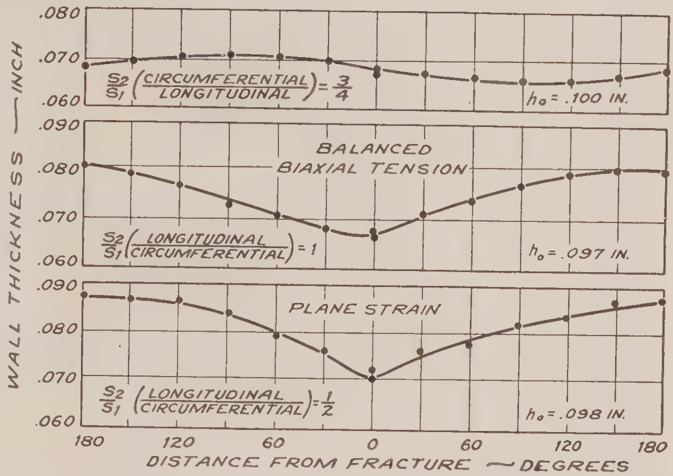


Fig. 1 Distribution of Wall Thickness With Circumference at the Center of Tubes Tested Under Various Stress Ratios and Having Longitudinal Fractures

Load conditions other than pure tension which can be readily investigated are those encountered in tubing which is simultaneously subjected to a major circumferential tension and a minor longitudinal tension.[3]

The following analysis should reveal the validity of the conception advanced[4] that a tube subjected to the stress state mentioned, if it does not fracture in a brittle manner, fails by instability when the internal pressure reaches a maximum. This failure should be of much the same type as that of a tensile specimen which fails by local instability or "necking" when the tensile load passes through a maximum.[5]

Some results have been published recently by Davis,[6] which present a series of tests suitable for a comparison of the theoretical instability strains with experimentally derived forming limits for various stress states. Cylindrical copper tubes 1.45 in. OD, 1.25 in. ID, and having an 8-in. length were machined from solid rod. These were subjected to various ratios of longitudinal and circumferential tension which were kept approximately constant in any one test but which were in different proportions for different tests. Both stresses were progressively increased during a test by increasing the internal pressure and the longitudinal tension. The strains were carefully measured during these tests, and for an investigation of the instability condition the final dimensions of the fractured specimens were also secured later for the purposes of this analysis (lot B of reference 6). Of the tubes tested under combined tensions, the stress and strain conditions of balanced biaxial tension $\left(\frac{s_2}{s_1} = 1, \frac{\epsilon_2}{\epsilon_1} = 1\right)$ and plane strain $\left(\frac{s_2}{s_1} = 1/2, \epsilon_2 = 0\right)$ were available for the analysis.

Correlating Test Results

To correlate the necking performance of tubes under uniaxial tension with those under biaxial tension, a tube tested in pure tension was also investigated. The individual (true) stress-strain curves obtained in the various tests were used as the basis for the determination of theoretical stress and strain conditions of instability under plane strain and balanced biaxial tension.

As might be expected, the tubes which had been subjected primarily to internal pressure failed by splitting longitudinally. The fracture appeared to be rather brittle; furthermore, the occurrence of a longitudinal fracture in the specimen tested with a ratio of circumferential stress to longitudinal stress of 3:4 indicated that the ductility in the circumferential direction was considerably less than in the longitudinal direction and possibly low enough to cause brittle fracture. Measurements of the distribution of thickness h, around the circumference at the longi-

[1] This paper is one of a series of reports on the research program on hot-forming of aluminum alloys conducted at Case School of Applied Science under contract with the Office of Production Research and Development of the War Production Board. This research, which was supervised by The War Metallurgy Committee under the "restricted" Project NRC-547, is the basis of this paper which has been released for publication by the OPRD. The project was directed by G. Sachs, Case School of Applied Science.

[2] Department of Metallurgical Engineering, Case School of Applied Science.

[3] As yet, the condition or combination of conditions determining instability for tubing under a major longitudinal tension and a minor circumferential tension is not clearly understood. However, the limiting conditions of pure longitudinal tension and balanced biaxial tension are known and discussed. The results of tests in the region between these two limits are later reported in Fig. 9.

[4] "Failure of Ductile Metals in Tension," by G. Sachs and J. D. Lubahn, published on pages 271–276, of this issue of the Transactions.

[5] "The Necking of Tensile-Test Specimens," by J. D. Lubahn, published on pages 277–279, of this issue of the Transactions.

[6] "Increase of Stress With Permanent Strain and Stress-Strain Relations in the Plastic State for Copper Under Combined Stresses," by E. A. Davis, Trans. A.S.M.E., vol. 65, 1943, p. A-187

Contributed by the Metals Engineering Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

tudinal center of fracture for this tube showed no definite evidence of local necking, Fig. 1. This therefore indicates that if any instability took place it occurred at a strain very near the limit of ductility.

The instability condition can be applied only if the ductility of the metal is sufficiently high to exceed that required by the necking condition. Measurements of the thickness distribution for tubes tested under balanced biaxial stress ($s_1/s_2 = 1$) and in plane strain ($s_2/s_1 = 1/2$), Fig. 1, reveal the presence of necks. The presumably uniform normal strain under both of these load conditions was considerably less than the normal strain at the fracture, which was approximately the same for all three tubes that failed longitudinally.

Thus it appears that the two tubes representing balanced biaxial tension and plane strain were suitable for a comparison of the theoretical with the experimental instability conditions.

First, however, the validity of the instability condition for the tube tested in pure longitudinal tension was confirmed. The load reached a maximum, Fig. 2, and from measurements of the thickness (radial strain, ϵ_3) and diameter (circumferential strain, ϵ_2) of the fractured specimen, a neck was found to extend over approximately 1 in. on each side of the failure. The remainder of the tube was strained rather uniformly, by a value of natural longitudinal strain, $\epsilon_1 = -2\epsilon_2 = 0.39$. This is practically the same value as that obtained by graphical solution, Fig. 3.

The pressure versus strain curves obtained for the tubes subjected to internal pressure show the decrease in slope which generally precedes a maximum, Figs. 5 and 7. However, no decrease in load was observed because of the explosive failure which occurs. Such failure is expected when the forces become unbalanced, i.e., when instability starts.

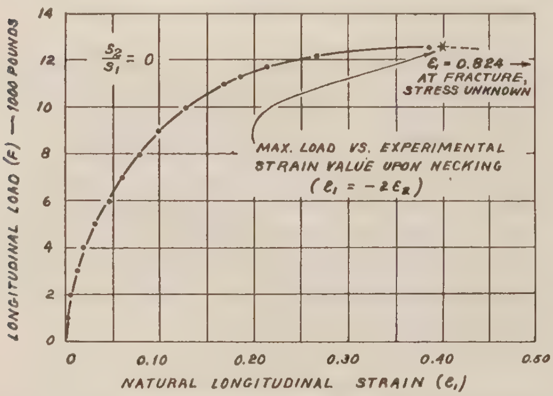


FIG. 2 LOAD-STRAIN CURVE FOR COPPER TUBING IN PURE TENSION

For the two combined tension tests made under balanced biaxial tension and plane strain, the thickness measurements taken around the circumference of the tubes at the longitudinal center of the fracture also showed no pronounced distinction between uniform and local straining, as the thickness gradually decreased toward the fractures, Fig. 1. This type of nonuniformity of strain which may be caused to some extent by slight variations in wall thickness of the original specimens, makes it difficult to determine definitely the circumferential strain at the moment of instability.

However, in the case of perfectly balanced biaxial tension, the uniform strain can be derived indirectly from the condition that before instability occurs the circumferential strain will equal the longitudinal strain; and since the fracture occurs in the longitudinal direction, the longitudinal strain should discontinue as soon

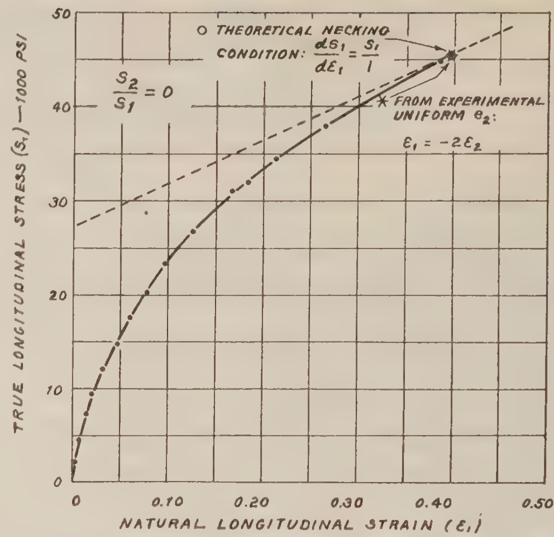


FIG. 3 TRUE STRESS-STRAIN CURVE FOR COPPER TUBING IN PURE TENSION WITH A COMPARISON OF THEORETICAL AND EXPERIMENTAL POINTS OF INSTABILITY

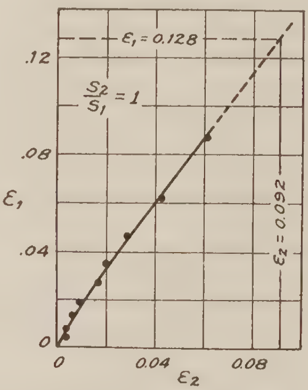


FIG. 4 EXTRAPOLATION OF ϵ_1 VERSUS ϵ_2 TO OBTAIN THE VALUE OF ϵ_1 UPON NECKING FROM THE CORRESPONDING VALUE OF ϵ_2

(Balanced biaxial tension test.)

as instability is initiated. Therefore the uniform strain in the longitudinal direction should be the same as the circumferential strain present at the start of instability. However, for the case considered, the circumferential stress (s_1) was approximately 13 per cent higher than the longitudinal stress when instability started; consequently there was a corresponding difference between the longitudinal and circumferential strains, Fig. 4. A value of circumferential strain equal to 0.128 at the point of instability was obtained by extrapolation of the curve in Fig. 4 to the measured value of uniform longitudinal strain of 0.092. The corresponding value of normal strain is

$$\epsilon_3 = -\epsilon_1 - \epsilon_2 = -0.092 - 0.128 = -0.22$$

The strain values obtained by graphical solution of the necking condition, Fig. 6, were $\epsilon_1 = 0.13$ and $\epsilon_3 = 0.25$. They are again in good agreement with the experimental values.

In the case of plane strain, Fig. 7, the lack of any change in length ($\epsilon_2 = 0$) of the specimen makes it impossible to use this strain as a criterion of the point of instability; and the lack of a

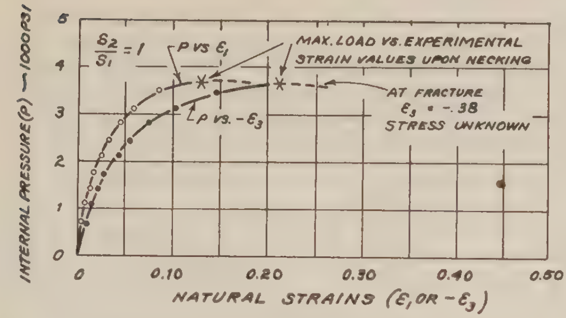


FIG. 5 PRESSURE-STRAIN CURVES FOR COPPER TUBING IN BALANCED BIAXIAL TENSION

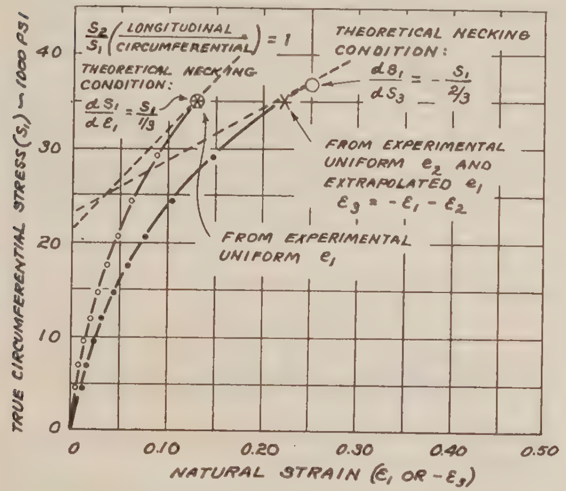


FIG. 6 TRUE STRESS-STRAIN CURVES FOR COPPER IN BALANCED BIAXIAL TENSION WITH A COMPARISON OF THEORETICAL AND EXPERIMENTAL POINTS OF INSTABILITY

region of uniform thickness about the circumference after fracture, Fig. 1, eliminates a second possible method of determining the uniform strain. However, the measured average circumferential strain may be used as the experimental strain at instability, although it is somewhat inaccurate because of the small effect on the average value of the larger strain at the neck. Measurement of the outside diameter after fracture at points $^1/_2$ in. beyond the ends of the fracture yielded a mean value of 1.699 in. This value may be corrected by the corresponding thickness (0.081 in.), obtained by extrapolation of the outside-diameter versus thickness curve, to yield a value of uniform natural circumferential strain of $\epsilon_1 = 0.175$.

As in the case of pure tension, the accuracy of the "theoretical necking strains" for the tubes investigated depends upon the accuracy of the slope measurements. Also, there is likely to be some additional error because of the scarcity of experimental points in the vicinity of the region of necking, with the corresponding uncertainty in the shape, and thus the slope of the curve.

The graphical solution of the instability condition for plane strain yields a strain value $\epsilon_1 = -\epsilon_3 = 0.19$ (approximately) which is again in good agreement with the experimental value $\epsilon_1 = 0.175$, Fig. 8.

Thus the analysis of tubes subjected to internal pressure confirms the conception that a maximum in the internal-pressure curve of a sufficiently ductile metal is associated with a localized

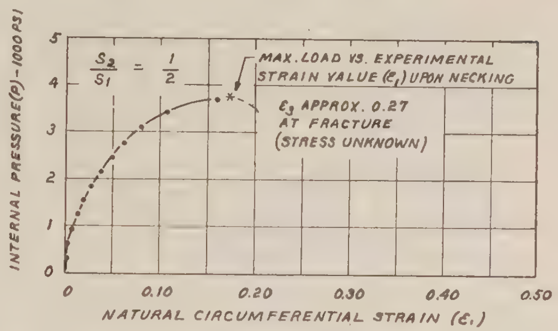


FIG. 7 PRESSURE-STRAIN CURVE FOR COPPER TUBING IN PLANE STRAIN

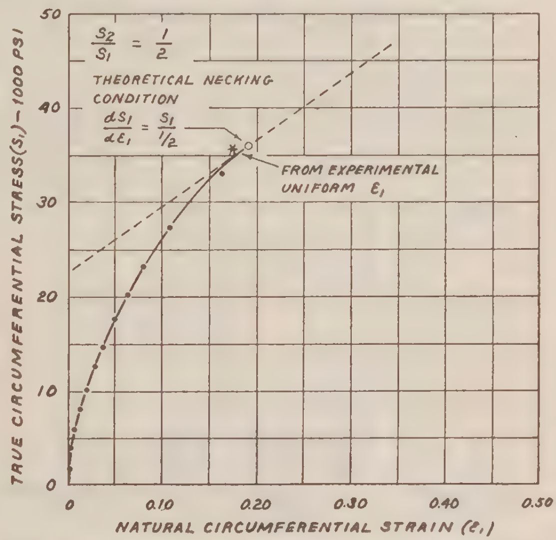


FIG. 8 TRUE STRESS-STRAIN CURVE FOR COPPER TUBING UNDER CONDITIONS OF PLANE STRAIN, I.E., LONGITUDINAL STRAIN = 0, WITH A COMPARISON OF THEORETICAL AND EXPERIMENTAL POINTS OF INSTABILITY

strain region or "neck" in the circumferential direction, and a lack of any further strain in other parts of the specimen.

The investigation of the various load conditions also reveals that in accordance with the theoretical analysis, a thin-walled tube subjected either to "pure" internal pressure (plane strain) or to balanced biaxial tension will fail by instability at a considerably smaller circumferential strain than the longitudinal strain which occurs on necking of a tensile test specimen made from the same metal, Figs. 3, 6, and 8. A tube subjected to pure internal pressure will become unstable at only approximately one half of the circumferential strain as a tube subjected to pure longitudinal tension. It can be shown that the conventional strength is a function of these strains; and the bursting strength of tubes, therefore, should be considerably lower than the tensile strength, the difference depending upon the strain-hardening characteristics of the metal and the degree of biaxiality.

In Fig 9 the three principal strains at the point of instability are represented for the tubes tested by Davis,[6] as obtained both experimentally and by theoretical analysis. The experimentation also covered four tubes not analyzed, which were subjected to a major longitudinal and to a minor circumferential tension. As previously discussed, the experimental evidence indicates

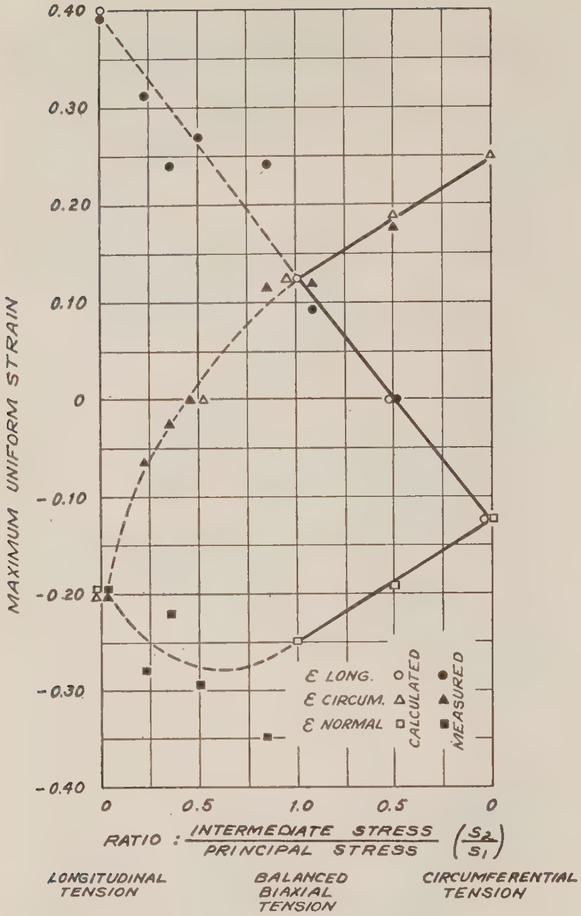


Fig. 9 Maximum Uniform Strains for Various Stress Ratios Existing in Tubing When Necking Starts

that the longitudinal strain at which a circumferential neck occurs should decrease as the internal pressure superimposed on the longitudinal tension increases, Fig. 9. The condition that the longitudinal load reaches a maximum, obviously does not account for the observed changes in longitudinal necking strain with increasing internal pressure, until balanced biaxial tension is

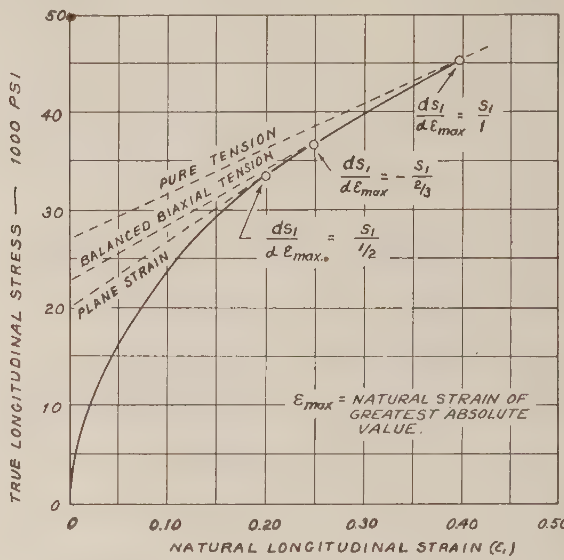


Fig. 11 Necking Condition for Tubing Under Various Stress States as Theoretically Determined, i.e., Graphically From True Stress-Strain Curve in Pure Tension

reached. The effect of the internal pressure in this range has not been accurately investigated as yet.

Prediction of Instability Strains From Stress-Strain Curve in Pure Tension

The instability condition has been developed in such a manner[4] that it may be used to predict the forming limits by necking from

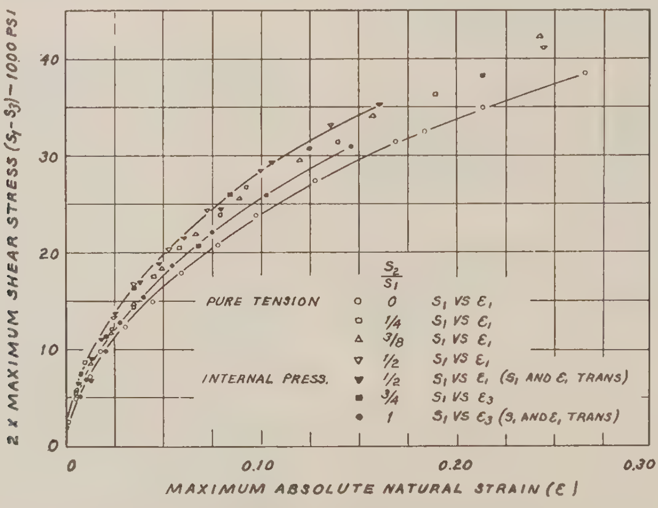


Fig. 10 Curves of Maximum Shear Stress Versus Maximum Absolute Strain for Copper Tubing Tested Under Various Stress States

the true stress-strain curve in pure tension for various conditions of geometry and stress ratios.

The data on copper tubing offer the opportunity to investigate the possibility of predicting the instability strain in this manner since the relation between the stresses and strains are known for this condition of geometry. As was previously mentioned, such application involves the assumption that true stress-strain curves for various stress states will be similar (particularly regarding slope), if the maximum shear stress is plotted against the largest absolute strain. While this maximum shear-stress criteria for flow may lead to considerable error in stress value, it can be shown that no considerable error in predicted strains will result.

In this paper only cases of uniaxial ($s_2 = s_3 = 0$) or biaxial tension ($s_3 = 0$) are considered and therefore the curves for various stress states should be similar if the largest stress s_1 is plotted against ϵ_{max}.

Fig. 10 shows such a plot for copper tubing in various stress states from $s_2/s_1 = 0$ to $s_2/s_1 = 1$. While these curves are not identical, it must be remembered that the slope of the curves is of greater importance than the ordinate value in determining the necking condition and that for a given abscissa value, the slopes for the various curves are nearly identical.

The instability theory has been applied to the true stress-strain curve in pure tension to predict the necking strain for copper tubes in plane strain and balanced biaxial tension, as shown in Fig. 11. Table 1 shows these values together with those obtained by the instability theory from the stress-strain curve for the particular stress state considered and also for the actually measured forming limit. These strain values obtained from the stress-strain curve in uniaxial tension are in good agreement with those derived experimentally.

TABLE 1 STRAIN VALUES FROM STRESS-STRAIN CURVES AND MEASURED NECKING STRAIN

Stress ratio in tubing	Necking strain from stress-strain curve corresponding to stress state	Necking strain from stress-strain curve in pure tension	Measured necking strain
$\frac{s_2}{s_1} = 0$	$\epsilon_1 = 0.40$	$\epsilon_1 = 0.40$	$\epsilon_1 = 0.39$
$\frac{s_2}{s_1} = 1$	$\epsilon_1 = 0.13$	$\epsilon_1 = 0.125$	$\epsilon_1 = 0.11$ $(= -\epsilon_3/2)$*
$\frac{s_2}{s_1} = 1/2$	$\epsilon_1 = 0.19$	$\epsilon_1 = 0.20$	$\epsilon_1 = 0.175$

* Corrected to apply to the stress ratio of $s_2/s_1 = 1$.

Acknowledgment

The author is indebted to Dr. A. Nadai and Dr. E. A. Davis, Westinghouse Research Laboratory, East Pittsburgh, Pa., for permission to measure the special dimensions of the test specimens after failure, and for the original data on pressure, load, and strain values.

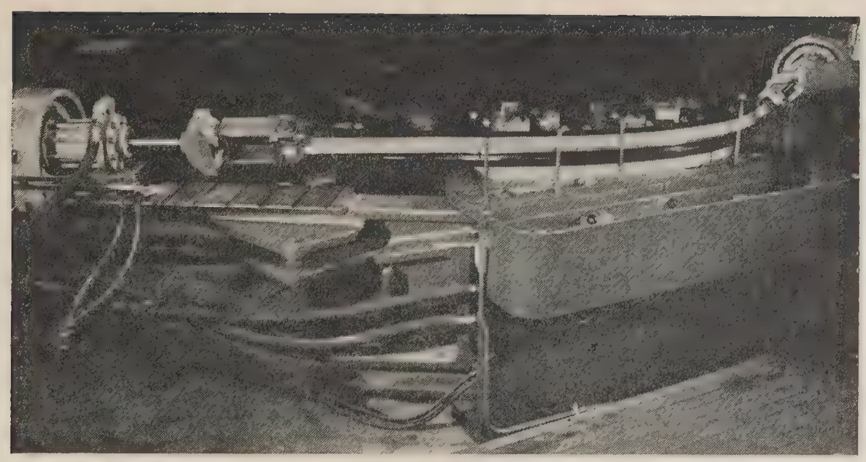


FIG. 1 TYPICAL CONTOUR-FORMING MACHINE IN WHICH TENSION IS COMBINED WITH BENDING

Distortion Due to Contour-Forming of Extrusions and Preformed Sheet-Metal Sections

BY WILLIAM SCHROEDER,[1] BURBANK, CALIF.

The effect of springback on the radius of curvature due to contour-forming of extrusions or preformed sheet-metal sections is analyzed. A number of the commonly used methods of forming are considered. Equations for computing values for change in radius are derived and sample calculations are compared.

NOMENCLATURE

The following nomenclature is used in the paper:

a = area
b = width
c = distance from innermost fiber to neutral axis of bending
d = depth of section
e = strain
e_1 and e_2 = strains due to forming; outer and inner fibers, respectively
ϵ_1 and ϵ_2 = components of elastic strain due to springback that cause a change in radius; outer and inner fibers, respectively
ϵ_x and ϵ_y = components of springback strain at point (x, y) due to elastic bending about the X- and Y-axes, respectively
F = force
I_x and I_y = moments of inertia about the X- and Y-axes, respectively
I_{xy} = product of inertia about the X- and Y-axes
$K = \dfrac{r}{s_0 R_p}$
m = first moment of area
M = moment of force
q_x and q_y = parameters
r = modulus of strain-hardening or slope of true stress-strain curve
R_p = radius of part
R_d = radius of die
R' = transverse radius of curvature due to springback distortion
s = true stress
s_0 = yield strength of material or s_t whichever is greater, see Fig. 5
x and y = co-ordinates with reference to X- and Y-axes
$\bar{x}$ = distance from vertical reference in section to centroid of area
$\bar{y}$ = distance from inner fiber to centroid of area
y_a = distance from inner fiber to axis, parallel to neutral axis, which divides area of section equally

INTRODUCTION

Several methods for computing springback in contour-forming have appeared in the technical literature.[2,3] These, however, apply only to one type of forming, namely, simple bending, and are trial-and-error methods which are very laborious. Many of the present methods of contour-forming have not previously been analyzed.

The springback resulting from contour-forming extrusions and preformed sheet-metal sections may conveniently be classed as

[1] Research Engineer, Lockheed Aircraft Corporation. Mem. A.S.M.E.

Contributed by the Metals Engineering Division and presented at the Aviation War Conference, Los Angeles, Calif., June 11–14, 1945, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[2] "Data on Springback of Metals," by R. E. Oestreich, *Aero Digest*, vol. 45, 1944, pp. 77–81 and 138.

[3] "Determining Springback," by R. G. Sturm and B. J. Fletcher, *Product Engineering*, vol. 12, 1941, pp. 526–528 and 590–594.

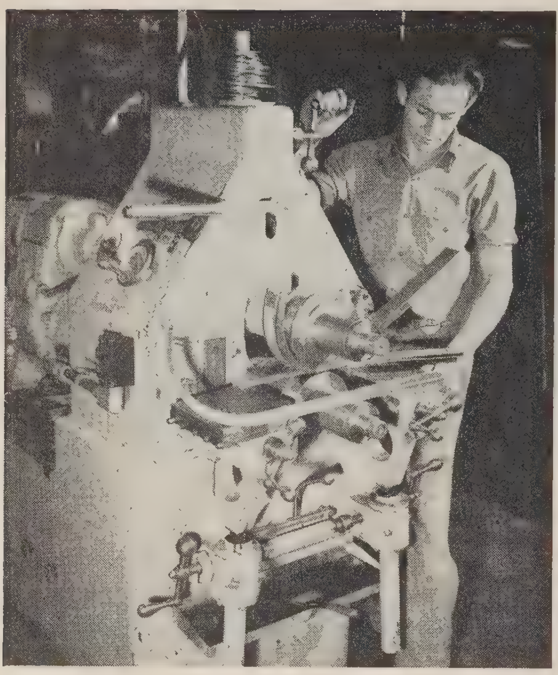


Fig. 2 Typical Contour-Forming Machine Operating on Principle of Simple Bending

(The part is bent by the rolls to the contour of the curved steel bar.)

consisting of a change in the over-all length or a change in radius of curvature of the part. A change in length of a part is generally of little significance because parts are normally trimmed to the desired lengths after forming; however, a change in curvature usually produces a significant error in contour that must be corrected. For this reason, changes in radius due to springback will be treated as important and changes in length as incidental.

In the hand-forming methods of forming stringer sections, the effect of springback may be corrected in the process of forming. When contour-forming machines[4] of various kinds, requiring dies or form blocks, are used, it becomes desirable to be able to predict the change in curvature due to springback in order to reduce the developmental costs of the dies.

The amount of radius change depends upon the forming method employed. Simple bending methods produce relatively large changes in radius while contour-stretching methods produce much less.[5] Many parts can be formed with these contour-stretching machines without excessive springback; nevertheless, under certain conditions springback allowance in the dies must still be made.

SPRINGBACK DUE TO CONTOUR-FORMING

The phenomenon of springback may be explained in terms of the stress-strain relationship for a bar loaded in tension and compression, as shown in Fig. 3. If the loading is interrupted at some point as at a and the load removed, the stress and strain decrease on the line a to c. The elastic recovery or decrease in strain is from b to c. In a similar manner, unloading the bar from point a' in compression changes the strain from b' to c'. The amount of plastic recovery is indicated for a number of values of strain in order to show the approximate manner of variation with strain.

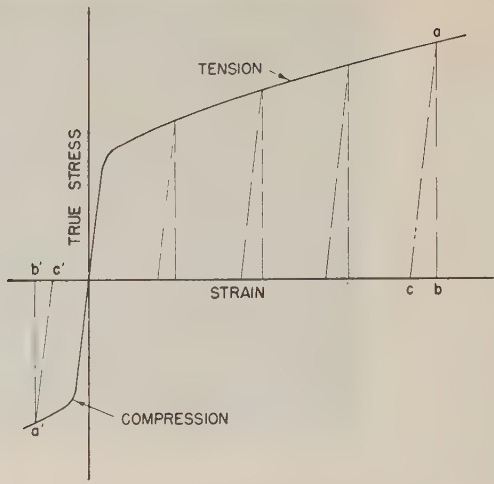


Fig. 3 Stress-Strain Curve and Springback in Tension and Compression

It has already been mentioned that springback from contour-forming depends upon the method and technique of forming. The amount and nature of springback furthermore depend upon the shape of the section. For the purpose of discussion, a number of typical cases will be discussed.

Method I—Forming to Approximate Contour, Heat-Treating, and Stretching to Final Contour

One method which may be expected to give a negligible amount of change in radius due to springback is to form the part approximately to contour by any of the contour-forming methods, follow with a solution heat-treatment or anneal, and stretch the part between 1 to 3 per cent over a block or die. This produces approximately the result illustrated in Fig. 4.

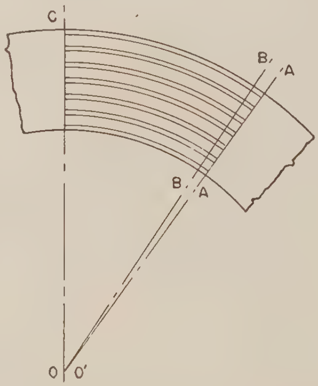


Fig. 4 Effect of Springback by Forming Method I

Owing to the heat-treatment following the forming, all fibers have the same temper at the time of stretching. Furthermore, all fibers are subjected approximately to the same stretching stress and strain since the part is already formed nearly to contour prior

[4] "Methods of Forming Aluminum Alloy Extrusions and Preformed Sheet Metal Sections," by William Schroeder, *Automotive and Aviation Industries*, vol. 90, 1944, pp. 28–31 and 100.

[5] "Elastic Theory in Sheet-Metal Forming Problems," by F. R. Shanley, *Journal of Aeronautical Sciences*, vol. 9, 1942, pp. 313–333.

to stretching. The elastic recovery is therefore proportional to the curved length of the fiber. Hence the fibers terminating on the plane A-A, while under stretching load, spring back to the position in plane B-B. If the line B-B is extended to intersect with line C-O, it is found that the new center of curvature O', coincides with the center of curvature O, before springback. Hence the radius of curvature remains unchanged even though the curved length has decreased.

This result applies equally to all sizes and shapes of cross sections. There is, furthermore, no tendency for the part to distort out of the plane of bending or become twisted.

Inconsistencies in this and all the following methods may result from nonhomogeneity of strength distribution over the cross section of the part. Such nonhomogeneity is sometimes associated with a nonhomogeneous grain structure.

Method II—Forming to Approximate Contour and Stretching to Final Contour

When parts are formed to approximately the correct contour and subsequently stretched without intermediate heat-treatment, a result somewhat different from Method I is obtained. The material becomes strain-hardened nonuniformly in the forming operation, the strain-hardening being greatest in the outer and inner radius fibers and decreasing toward the middle. Springback from stretching a part with such nonuniform strain-hardening will generally produce slight distortion. The amount and type of distortion will depend on the shape of the cross section and the method of contouring. The part will also distort from the plane of bending if the cross section or distribution of strain-hardening is not symmetrical about an axis parallel to the plane of bending. (The plane of bending corresponds to the Y-axis n the figures of this paper.)

Since a wide variety of conditions for this method can exist, each leading to a different result, no attempt to solve any specific case has been made.

Method III Simultaneous Wrap and Stretch-Forming With Constant Force

One rather common method of contour-forming consists of bending and stretching progressively. By this method the part is usually first loaded in tension to a stress corresponding to a very small permanent set. The part is then wrapped around the die. This wrapping action is basically one of bending. It differs from pure bending by the fact that the bending is done under the influence of the tensile force, Fig. 5. Simple bending may be considered as a special case in which the force $T = 0$.

Neutral Axis of Bending With Tensile Force. As a step in analyzing the effect of springback on the radius, the neutral axis of bending will be considered first.

From the geometry of the bend, Fig. 5, it may be seen that

$$d\theta = \frac{dL_1}{R_d + d} = \frac{dL_2}{R_d} \qquad [1]$$

By definition

$$\frac{dL_1}{dL_0} = 1 + e_1 \qquad [2]$$

and

$$\frac{dL_2}{dL_0} = 1 + e_2 \qquad [3]$$

where dL_0 = unstrained length of dL_1 and dL_2.

Therefore

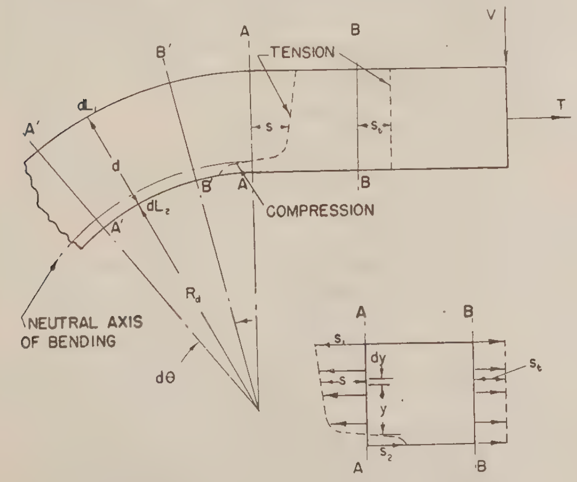
$$\frac{1 + e_1}{R_d + d} = \frac{1 + e_2}{R_d} \qquad [4]$$

or

$$\frac{d}{R_d} = \frac{e_1 - e_2}{1 + e_2} \qquad [5]$$

Stretching of the straight portion of the part may result from the original application of the tensile force; however, the main forming action takes place in the region adjoining section A-A, Fig. 5. The stress conditions are approximately as in Figs. 5(b) and 6. The outside fibers elongate and increase in strength according to the stress-strain behavior of the material. In order for the material between sections A-A and B-B to be in equilibrium, it is necessary that the net force represented by the area under the stress-distribution curves for sections A-A and B-B be equal. This condition is expressed by

$$-\int bsdy + as_t = 0 \qquad [6]$$



(a) Geometry of bend (b) Free body diagram

Fig. 5 Contour-Forming an Extrusion, or Preformed Section, With Constant Tensile Force

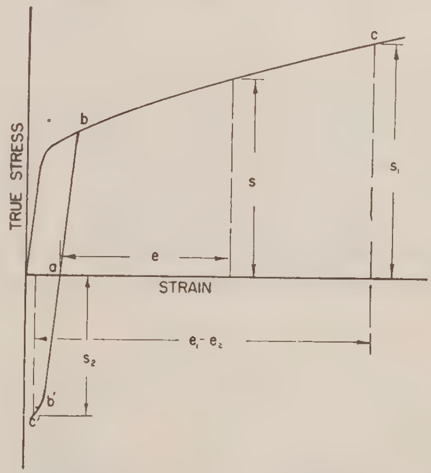


Fig. 6 Stress-Strain Relation for Contour-Forming With Constant Tensile Force

It is assumed that the strain imposed by bending is proportional to the distance from the neutral axis. In this case, the neutral axis of bending is defined as the fiber in the section that has zero stress.

The solution of Equation [6] depends on the relation between s and y, which in turn depends upon the stress-strain diagram. An exact solution is difficult to obtain; however, a useful solution may be obtained by assuming that the stress-strain diagram a-b-c, Fig. 6, may be expressed by

$$s = s_0 + re \quad \ldots\ldots [7]$$

and a-b'-c' by

$$s = -s_0 + re \quad \ldots\ldots [7a]$$

In this expression

$$e \simeq \frac{y}{R_d} \quad \ldots\ldots [8]$$

where y is measured from the neutral axis of bending.

For all cases except $s_t = 0$ (pure bending), it may be assumed that $s_0 = s_t$. For $s_t = 0$, s_0 equals approximately the yield strength of the material.

Substituting Equations [7], [7a], and [8] into [6] gives

$$-\int_0^{d-c} bs_0dy + \int_{-c}^{0} bs_0dy - \int_{-c}^{d-c} \frac{b}{R_d} rydy + as_0 = 0. \, [9]$$

or

$$-s_0(a_a - a_b) - \frac{r}{R_d} a(\bar{y} - c) + as_0 = 0 \quad \ldots\ldots [9a]$$

where

$$a_a = \int_0^{d-c} bdy = \text{area above neutral axis}$$

and

$$a_b = \int_{-c}^{0} bdy = \text{area below neutral axis}$$

Equation [9a] will be evaluated for cross sections composed of rectangular components as in Fig. 7. It may be demonstrated that $(a_a - a_b)$ is equal to twice the shaded area between y_a and c, where y_a is the distance from the inner fibers to an axis, parallel to the neutral axis, that divides the cross-sectional area into two equal parts. Several special cases have been evaluated as follows:

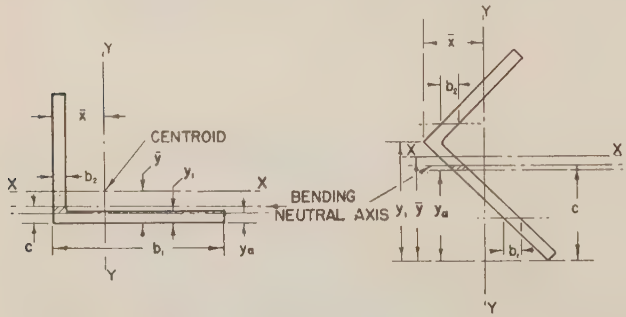


Fig. 7 Dimensions Required for Computing Springback Due to Contour-Forming

Case (a): When $y_a \leqq y_1$, Fig. 7. For this case, Equation [9a] becomes

$$-2s_0b_1(y_a - c) - \frac{r}{R_d} a(\bar{y} - c) + as_0 = 0 \quad \ldots\ldots [10]$$

and

$$c = \frac{2b_1y_a + Ka\bar{y} - a}{2b_1 + Ka} \quad \ldots\ldots [11]$$

where

$$K = \frac{r}{s_0R_d}$$

Case (b): $y_a > y_1 > c$, Fig. 7. Substituting into Equation [4a]

$$-2s_0b_2(y_a - y_1) - 2s_0b_1(y_1 - c) - \frac{ra}{R_d}(\bar{y} - c) + as_0 = 0 \, . . [12]$$

and

$$c = \frac{2b_2(y_a - y_1) + 2b_1y_1 + Ka\bar{y}}{2b_1 + Ka} \quad \ldots\ldots [13]$$

Neutral Axis in Simple Bending:

(*a*) For simple bending the value of c lies between y_a and $\bar{y}$; and $T = 0$. When $c \leqq y_1$ and $y_a < y_1$

$$c = \frac{2b_1y_a + Ka\bar{y}}{2b_1 + Ka} \quad \ldots\ldots [14]$$

(*b*) For $y_a < y_1$ and $c > y_1$

$$c = \frac{2b_1(y_a - y_1) + 2b_2y_1 + Ka\bar{y}}{2b_2 + Ka} \quad \ldots\ldots [15]$$

(*c*) For $c > y_1$ and $y_a \geqq y_1$

$$c = \frac{2b_2y_a + Ka\bar{y}}{2b_2 + Ka} \quad \ldots\ldots [15a]$$

Determination of Part Radius After Springback

The springback in contour-forming with a constant tensile force may be analyzed by considering the moments required for bending and the change in moments due to springback. The bending moment depends upon the shape of the section and the plastic properties of the material, while the moment due to springback depends upon the elastic properties of the section. The following equilibrium conditions must be satisfied

$$\Sigma M_x = 0 \quad \ldots\ldots [16]$$

$$\Sigma M_y = 0 \quad \ldots\ldots [17]$$

$$\Sigma F_n = 0 \quad \ldots\ldots [18]$$

where

M_x and M_y = moments about the X- and Y-axes, respectively

F_n = force normal to X-Y plane

In this case, the X- and Y-axes are taken through the centroid of the section with the Y-axis in the plane of bending.

Since springback is a purely elastic phenomenon, the principle of superposition may be applied and the changes in stress and strain may be resolved into components as due to bending about each of the two axes and a uniform change in strain normal to the X-Y plane. The uniform

strain component produces no distortion, as previously explained, and can therefore be neglected in the following discussion. The other elastic components are related as follows

$$\epsilon = \epsilon_x + \epsilon_y \quad \text{[19]}$$

Since Equation [18] yields information only on the uniform component of strain, it may be omitted from further discussion.

Based on Fig. 5, the moments due to bending are expressed by

$$M_{1x} = \int_s syda \quad \text{[20]}$$

and

$$M_{1y} = \int_s sxda \quad \text{[21]}$$

where $\int_s$ means a surface integral over the cross-sectional area and da refers to an area element with dimensions dy and dx. The changes in moment due to springback are

$$M_{2x} = \int_s E\epsilon yda \quad \text{[22]}$$

and

$$M_{2y} = \int_s E\epsilon xda \quad \text{[23]}$$

Substituting into Equations [16] and [17]

$$\int_s syda + \int_s E\epsilon yda \quad \text{[24]}$$

and

$$\int_s sxda + \int_s E\epsilon xda \quad \text{[25]}$$

The solution of Equations [24] and [25] depends mainly upon the stress-strain curve between points c' and c, Fig. 6. An approximate but useful result is again obtained by assuming

$$s = s_0 + re \text{ and } s = -s_0 + re$$

The elastic bending strain ϵ may furthermore be assumed to be composed of ϵ_x and ϵ_y, where ϵ_x is due to bending about the X-axis, and ϵ_y due to bending about the Y-axis. Assuming that planes before forming remain planes after forming

$$\epsilon_x = \frac{\epsilon_{x1} - \epsilon_{x2}}{d} y = q_x y \quad \text{[26]}$$

and

$$\epsilon_y = \frac{\epsilon_{y1} - \epsilon_{y2}}{d} x = q_y x \quad \text{[27]}$$

For the condition after springback, Equation [5] becomes

$$\frac{d}{R_p} = \frac{(e_1 - \epsilon_{x1}) - (e_2 - \epsilon_{x2})}{1 + e_2 + \epsilon_{x2}} \quad \text{[28]}$$

and

$$\frac{d}{R_p} \simeq \frac{d}{R_d} + \frac{\epsilon_{x1} - \epsilon_{x2}}{1 + e_2} = \frac{d}{R_d} + \frac{q_x d}{1 + e_2} \quad \text{[29]}$$

This may be rearranged to give

$$R_p = \frac{R_d}{1 + \dfrac{q_x R_d}{1 + e_2}} \quad \text{[30]}$$

The distortion arising from springback tending to bend the extrusion from the plane of bending may be expressed in terms of

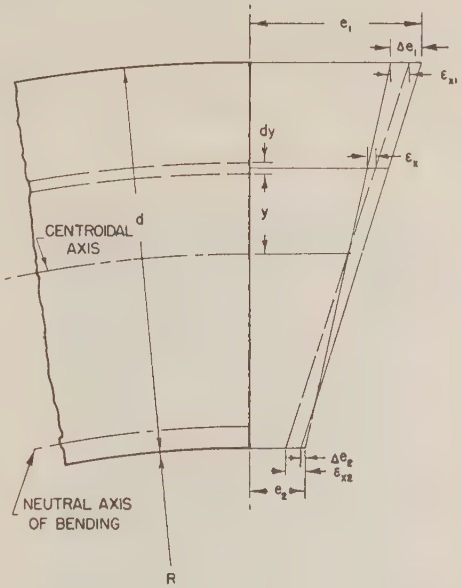


Fig. 8 Strain Relations in Centroidal Plane About X-Axis Due to Forming and Springback

q_y. The transverse radius of curvature R' is obtained by substituting ϵ_{y1} and ϵ_{y2} for e_1 and e_2, respectively, in Equation [5]; thus

$$\frac{d}{R'} = \frac{\epsilon_{y1} - \epsilon_{y2}}{1 + \epsilon_{y2}} \simeq \epsilon_{y1} - \epsilon_{y2} \quad \text{[31]}$$

Combining Equations [31] with [26]

$$R' = \frac{1}{q_y} \quad \text{[32]}$$

Values for q_x and q_y may be obtained by solving Equations [24] and [25]. After substituting for s and ϵ, Equation [24] becomes

$$s_0 \int_{c-\bar{y}}^{d-\bar{y}} yda - s_0 \int_{-\bar{y}}^{c-\bar{y}} yda + \frac{r}{R_d} \int_s y^2 da + E \int_s (q_x y + q_y x) yda = 0 \quad \text{[33]}$$

and

$$s_0(m_{x1} - m_{x2}) + \frac{r}{R_d} I_x + \frac{r}{R_d} a(c - \bar{y})^2 + Eq_x I_x + Eq_y I_{xy} = 0 \quad \text{[34]}$$

where

m_{x1} = first moment of area above the neutral axis of bending taken about the X-axis

m_{x2} = first moment of area below the neutral axis of bending taken about the X-axis

Similarly Equation [25] becomes

$$s_0(m_{y1} - m_{y2}) + \frac{r}{R_d} I_{xy} + Eq_x I_{xy} + Eq_y I_y = 0 \quad \text{[35]}$$

where

m_{y1} = first moment of area above the neutral axis of bending taken about the Y-axis

m_{y2} = first moment of area below the neutral axis of bending taken about the Y-axis

It may furthermore be demonstrated that in Equations [34] and [35]

$$m_{x1} = -m_{x2} \text{ and } m_{y1} = -m_{y2}$$

In computing springback for the conditions of forming with a constant tensile load, specific values are substituted into Equations [34] and [35] to solve for the values of q_x and q_y. The values of q_x and q_y are then substituted into Equations [30] and [32] for R_p and R'.

Forming Method IV—Simultaneous Wrap and Stretch Forming With Increasing Tensile Force

When the extrusion is contoured and stretched with an increasing tensile force the whole section is subjected to tensile stress and the neutral axis, as defined, does not exist. Since the X- and Y-axes are centroidal axes and under the present condition the area above the neutral axis is the whole area a, and the area below the neutral axis is zero, it follows that

$$m_{x1} = m_{x2} = m_{y1} = m_{y2} = 0$$

From Equations [34] and [35]

$$q_y = 0 \text{ and } q_x = -\frac{r}{ER_d}$$

Substituting these values into Equations [30] and [32]

$$R_p = \frac{R_d}{1 + \dfrac{r}{E(1 + e_2)}} \qquad [36]$$

and

$$R' = \infty \text{ (no transverse curvature)} \qquad [37]$$

Discussion of Forming Methods

Comparison of Springback. The relative amounts of radius change resulting from the foregoing forming methods are computed for one specific condition and compared in Table 1. It may be seen that the radius change is negligible for Method I and is progressively greater for Methods IV, III with tensile pull, and Method III without tensile pull (pure bending). The results for Method II are difficult to predict but are expected to range between the results for Methods I and IV. Of the methods discussed, I and IV will produce little tendency to distort the part from the plane of bending even though the section is not symmetrical about the Y-axis.

TABLE 1 COMPARISON OF COMPUTED RESULTS FOR FORMING A $^3/_4$-IN. × 1-IN. BY 0.070-IN. EXTRUSION WITH $^3/_4$-IN. LEG IN PLANE OF BENDING

(Material: 24S-T aluminum alloy; and R_d = 20 in.)

Method of forming	Part radius in plane of bending R_p, in.	Resulting transverse radius of curvature R', in.
I	20	∞ (no distortion)
II (values not computed but range is estimated)	20–20.55	6000—∞
III (a) s_t = 53,000 psi	20.94	6000 (slight)
III (b) s_t = 0 (simple bending)	22.6	71 (appreciable)
IV	20.55	∞ (no distortion)

In any of the methods that involve stretching it is important that the jaws for gripping are so designed that they will develop the necessary gripping force without weakening or twisting the section to be formed. Another serious difficulty of contour-forming by stretching is due to friction between the part being formed and the die. This friction makes it difficult and sometimes impossible to stretch the part throughout the length in contact with the die when using Methods I and II. This factor may also interfere with the proper execution of Method IV. The use of a good lubricant and vibrating the die are beneficial for reducing friction between part and die.

Experience may show that friction between the part and die may make it impractical or impossible to attain the conditions assumed in Method IV, and that minimum amounts of springback in practice will correspond more nearly to those for Method III using a constant tensile force.

Accuracy of Results

There has been little opportunity to compare computed with measured values for change in radius due to springback. Where checks have been possible the results have agreed within the limits of accuracy of the measurements. Due to the simplifying assumptions that were made some slight disagreement between computed and measured values may be expected; however, experience has indicated that the error from this cause is considerably less than the variation that may be expected from nonhomogeneity and variation in physical properties of material used in forming.

(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until June 17, 1946)

Design Considerations for Welded Machinery Parts

By G. L. SNYDER,[1] COATESVILLE, PA.

Only some basic considerations for the designer in developing weldments are brought out in this paper, because of the wide scope of the subject. Separately, these basic considerations may seem elementary and self-evident, but when considered collectively in developing a weldment, their complexity becomes apparent. At least twelve different types of components can be utilized by a designer in his weldment and each of these components can be used in several ways. So, the designer must be familiar with limitations in processing weldments and with the scope and limitation of equipment and methods used in their production. These are the factors discussed here. However, this discussion is limited to dynamically loaded welded machinery parts.

HOT-rolled steel plate, a basic element, probably is the most universal component used in welded machinery parts. Undoubtedly, the freedom offered by the many sizing and shaping possibilities of hot-rolled steel plate has much to do with its widespread use as weldment components. Basically flexible raw material from the stand-point of dimensions, hot-rolled plate is obtainable in variable sizes to 195 in. wide or to 25 in. thick.

Flat Components

Shearing usually is the most economical method of sizing or shaping a plate to rectangular or circular dimensions. But when a plate is to be sheared to size, the designer should keep in mind the existence of shear droop which is the abrupt break in flatness that occurs around the sheared edge, because localized stresses imposed by the shearing pressure exceed the elastic limit of the material. Since this effect is confined to the area adjacent to the edge, it can be practically disregarded when the component is subjected to subsequent trimming or when it is in light gages, $^3/_8$ in. and under.

Flame-cutting undoubtedly is the most common method of shaping and sizing weldment components, particularly when the number of duplicate weldments required is small. Probably a big reason for this is the fact that many components of weldments necessarily are irregular in shape.

Since regular configuration generally is cheaper, the designer should think in such terms where possible. However, he need not be concerned if an edge is sheared or flame-cut so long as he designs the part so that the supplier is free to use either method or both on a particular component.

Fig. 1 shows an example of component produced entirely by flame-cutting. Fig. 2 is a plate part produced by a combination of shearing and flame-cutting.

Of course, in addition to shaping and sizing, the cutting torch

Fig. 1 Component Flame-Cut to Shape from Hot-Rolled Plate

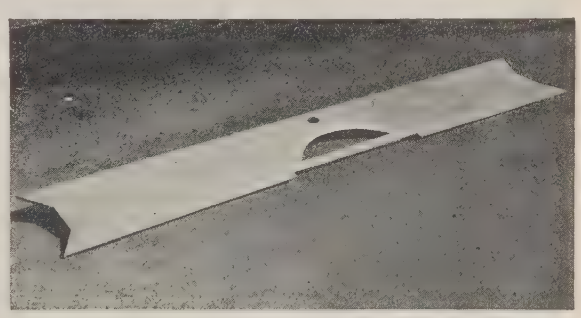


Fig. 2 Plate Component Shaped by Shearing and Flame-Cutting

also makes welding chamfers, or kerfs, on the edges of a component which provide the "grooves" for welds, other than plain fillets, when assembled with adjoining components.

Tolerance on components, disregarding the method used in producing them, merits careful consideration by the designer of weldments for an accumulation of tolerances can cause costly difficulty in fabrication, as is shown in Fig. 3. This illustration pictures a highly improbable coincidence of tolerance accumulation. Nevertheless it could occur within the limits of necessary commercial tolerances on flatness and straightness. Tolerances are important considerations on flatness and straightness, particularly if the fabricator's facilities for performing such operations cannot be operated as cheaply as those of the supplier of components.

Often it is advantageous to size components on machine tools, if only for the reason that much closer tolerances are obtainable. For a complicated assembly with much welding on it, prefabrication machining is indicated. Also, at times, machining of components will help achieve close tolerance on a complete weldment. The more generous the tolerances on components, the more prevalent the gaps in fitting. Gaps require the deposition of a

[1] Chief Engineer, Lukenweld, Inc., Division of Lukens Steel Company.

Contributed by the Production Engineering Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

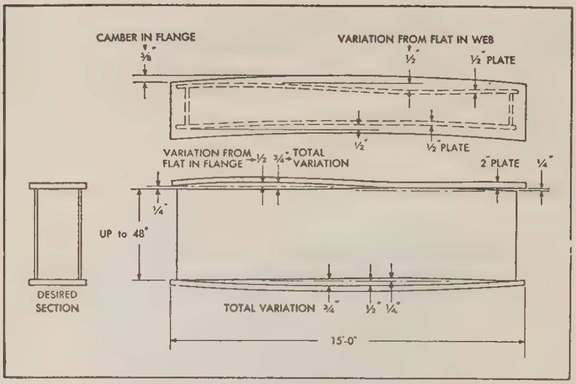


Fig. 3 Box Beam Showing Results of Camber and Flatness Tolerances

greater amount of weld metal, thereby increasing costs and destroying the metal-to-metal contact which resists tendencies to shrink or warp.

The main structural parts of the weldment shown in Fig. 4 are

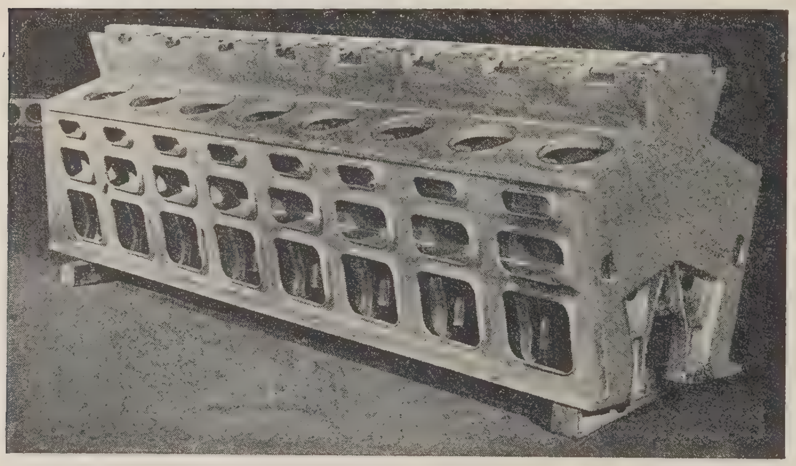


Fig. 4 16-Cylinder, V-Type, Welded Diesel-Engine Frame

premachined. Since this item is one of mass production, individual peculiarities in each weldment caused by the nonuniform accumulation of component tolerances could not be allowed.

Sometimes design considerations dictate prefabrication ma-

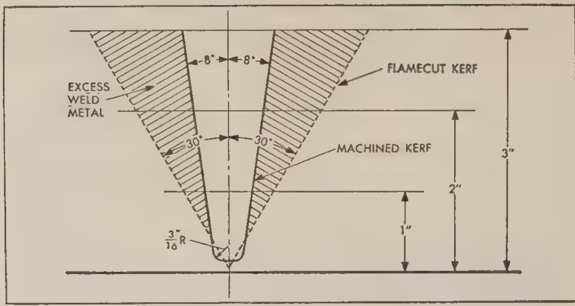


Fig. 6 Comparison Between "U" and "V" Welding Kerfs

chining as in the case of the joint between the heavy web and flange shown at the left in Fig. 5. This might be a detail of construction on the bed of a large hydraulic or mechanical press. The loading in the region of the compression flange of those machines is such that the joint detail, shown at the right in Fig. 5, would be necessary if the edge of the web were not machined. When machined, the metal-to-metal bearing, to withstand concentrated compression loads, is achieved. Fillet welds as indicated in this illustration are adequate then for withstanding the horizontal shear components in this region of the beam. The edging of such webs to a relatively close tolerance is a simple operation on a plate planer.

Another reason for prefabrication machining is the provision of economical welding kerfs in combination with good joint fit-up, particularly in welding thick plates. A kerf must be sufficiently wide to clear the tip of the welding electrode to permit the depositing of weld metal at the root of the weld; for as the plate thickness increases, it is apparent that the amount of metal wasted in the angularity of V-shape flame-cut kerfs in contrast to the U-shaped ones, as shown in Fig. 6, becomes an important factor.

Prefabrication machining is necessary, too, in instances of conflicting tolerances, in the fitting of circular components within each other. Studies of minimum tolerances reveal that gaps between pieces so fitted are inevitable but machine fits reduce such

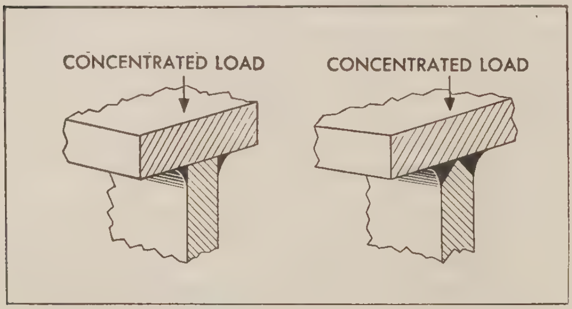


Fig. 5 Joint Between Heavy Flange and Web, Showing Machined Edge at Left and Full Weld at Right

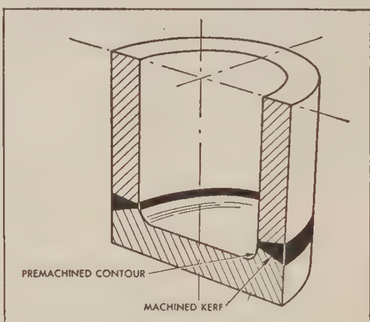


Fig. 7 Bottom of Hydraulic Cylinder, Typical of Those Used in Heavy Hydraulic Presses

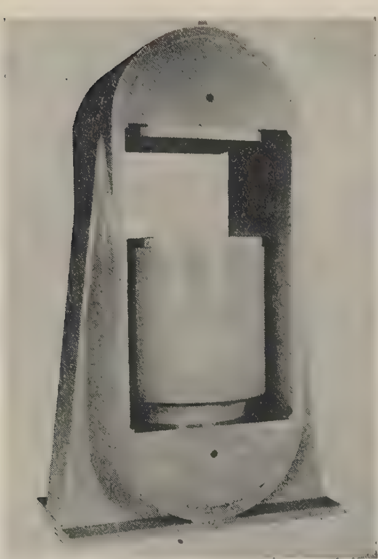


FIG. 8 CYLINDER ON HYDRAULIC PRESS

FIG. 9 BLANKED, AND BLANKED AND PUNCHED COMPONENTS MADE OF 1/4-IN-THICK STEEL PLATE

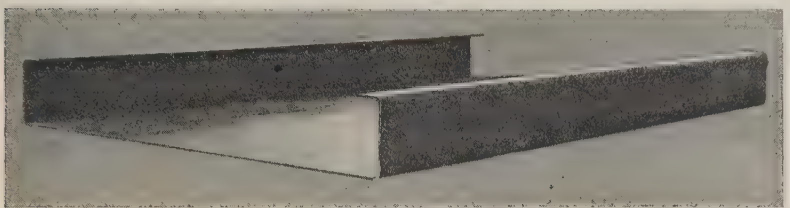


FIG. 10 HORIZONTAL ANGULAR BENDS PROVIDED BY PRESS-BENDING

FIG. 11 WELDMENT SHOWING UTILIZATION OF FLUED OPENING

gaps to a negligible point. Prefabrication machining is necessary at times, also, for providing proper contours in highly stressed weldments, or in those subject to fatigue. Fig. 7 shows such an instance in the bottom plate of a hydraulic cylinder premachined to provide proper curved contour at the corners. This sketch also illustrates an application of machined kerfs on thick plates. Fig. 8 shows the cylinder of a hydraulic press designed on such principles.

Another method used frequently for shaping plate components is "blanking" or "punching" on a power press. This operation which is simply shearing, using knives of special shapes, can be justified, usually, only when quantities required warrant the expense of dies. Fig. 9 shows typical blanked or blanked and punched pieces.

A simple example of blanking or punching is the shearing of a rectangular plate. By blanking the piece only one operation is required in comparison to four operations for each piece which would be necessary in shearing. Hence a comparison of blanking and shearing costs can be made merely by multiplying the cost of the added three shearing operations by the number of pieces required and comparing this with the cost of tooling. Naturally, the estimate will be approximate since the relative cost per hour of the machines used might affect the comparison. Another important benefit gained by blanking is the comparatively close tolerance that can be achieved.

"FORMED" COMPONENTS

Thus far we have considered only flat pieces in the preparation of components but we must give thought also to "formed" type components which are required frequently in weldments. Several methods are in general use for forming components for weldments. One of these is press-bending to make horizontal angular bends as are shown in Fig. 10.

Definite reasons for forming operations such as bending or fluing have been evolved. One is lower cost, for angular bends eliminate one or more welded joints. The cost of bending seldom equals that of the alternative assembly and welding. Careful examination of the design proportions of metal sections might show the economy of using the same metal thickness of web and flange to utilize the advantage of a bent section. A bent com-

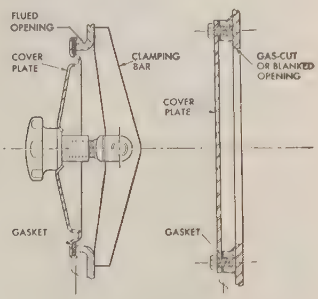


FIG. 12 THE FORMED SEAT COVER AT THE LEFT IS DESIGNED TO BE FASTENED BY AN INNER CLAMP

(Assembly, welding and consequent warpage, and the cost of drilled and tapped holes are eliminated as in the case of the cover at the right.)

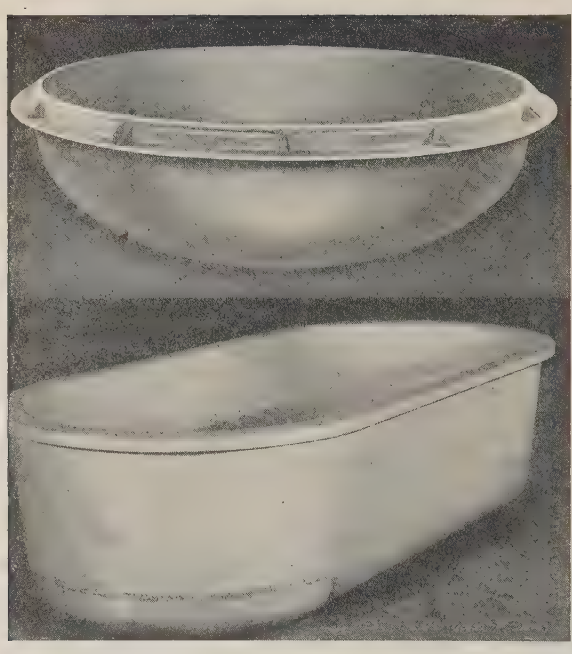


Fig. 13 Melting Pots Using Flanged Products as Components

ponent naturally is more rigid than a flat one. This can be important in the control of shrinkage and warpage.

Another method often of value in component production is the specialized forming operation known as fluing. Flued openings such as shown in Fig. 11, when machined, provide formed seats for covers. Generally, such a cover is designed to be fastened by an inner clamp as shown at the left in Fig. 12. Treatment permitted by the flued opening eliminates assembly, welding, and consequent warpage, and the cost of many drilled and tapped holes as shown at the right in Fig. 12. In addition, weight reduction is achieved.

Another function of flued openings in weldments is to provide stiffening lips which are executed normally by welding a band around the opening where required by design considerations.

Fig. 14 Corner Stamping for Water-Cooled Furnace Door

Fig. 15 Furnace Door Using Corner Stampings Shown in Fig. 14

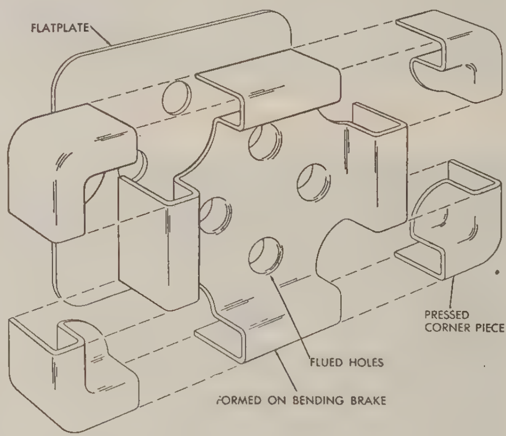


Fig. 16 Exploded View of Water-Cooled Furnace Door Showing Construction

Since dies are necessary, flued openings are unwarranted economically unless the quantity of openings justifies their use. When flued openings are being considered the designer should consult the supplier for it is possible that dies exist which can be adapted.

Products of the flanging or spinning machines often are utilized to advantage in weldments, as their applications in the melting pots in Fig. 13 show. The bottom corners of the pot in the lower illustration are composed of a flanged head split in half, with one half serving for each corner.

The use of flanged products as components may provide component rigidity, possibly simpler welding conditions, or a reduction in welding. Flanging also provides curved contours which may be desired for proper functioning or for appearance.

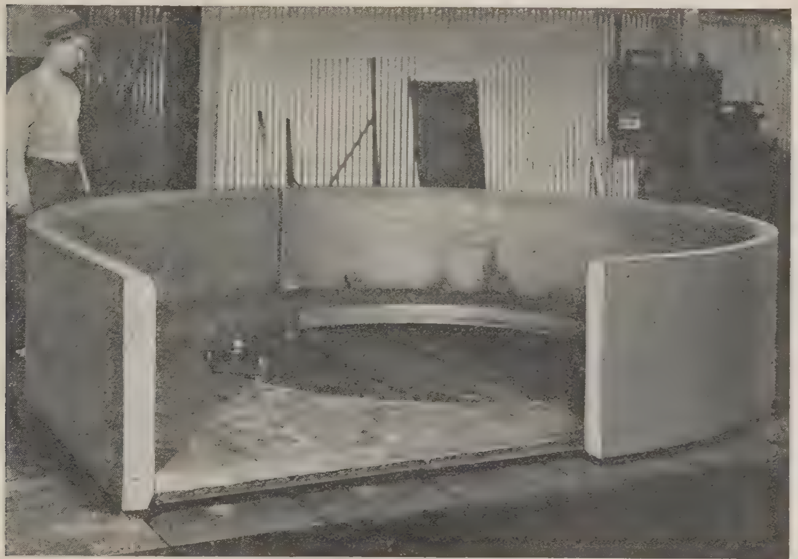


Fig. 17 Plates 4 In. Thick Formed on Bending Roll

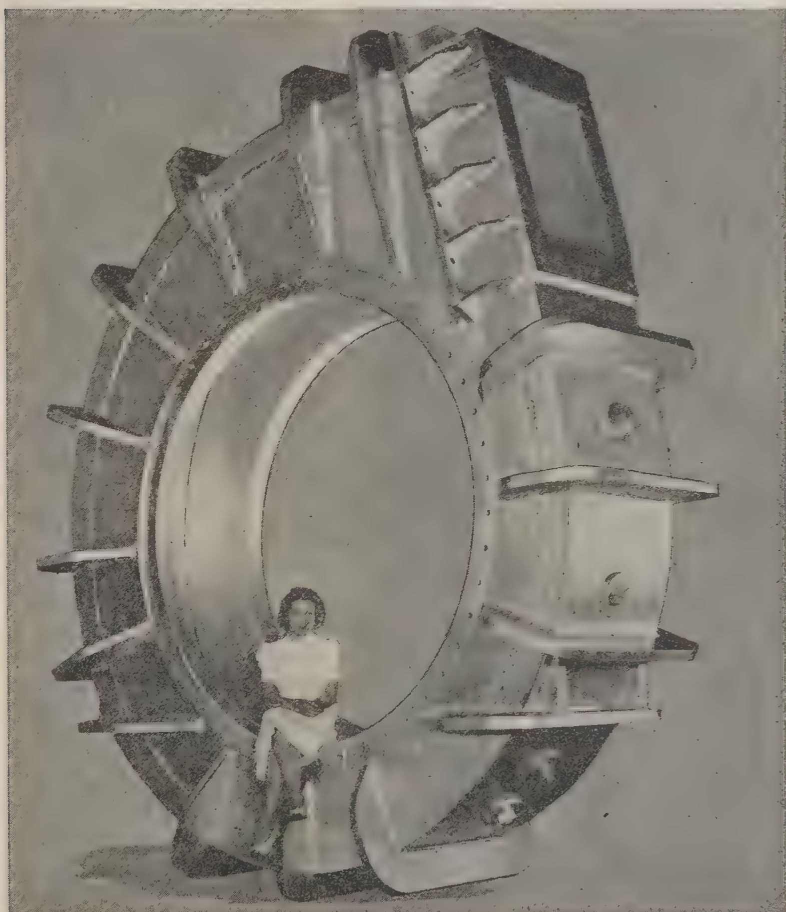


Fig. 18 Pump Casing Utilizing Components Shown in Fig. 17

Stamping and Pressing

When quantities justify, formed components produced in special shapes by forming dies on power presses may be used advantageously. Fig. 14 shows a small stamping and Fig. 15 shows the weldment in which four of these stampings form the corners of the piece. Forming, required on the remaining components, is done on a press brake. Fig. 16 shows the relation of the various components in this weldment.

A useful method of shaping cylindrical contours is provided by bending rolls. A component formed in such a manner is shown in Fig. 17. This component is seen readily in the finished weldment shown in Fig. 18.

Shapes Formed on Rolling Mill

At times, shapes formed on a rolling mill may be used justifiably in weldments. Their value as components results from two factors: A reduction in cost because of elimination of welding and initial rigidity which can tend to simplify fabrication problems of shrinkage and warpage. In considering the use of structural rolled shapes, the tolerances possible in such rolling-mill products should be studied carefully for they may affect adversely the design requirements.

Steel castings are used extensively as components in weldments where economy in producing complicated shape requirements or special contours at given points in a particular assembly are involved. When castings are to be used in weldments their physi-

Fig. 19 Typical Use of Drop Forgings and Flash-Welding to Provide Components of Irregular Shape and Difficult Proportions

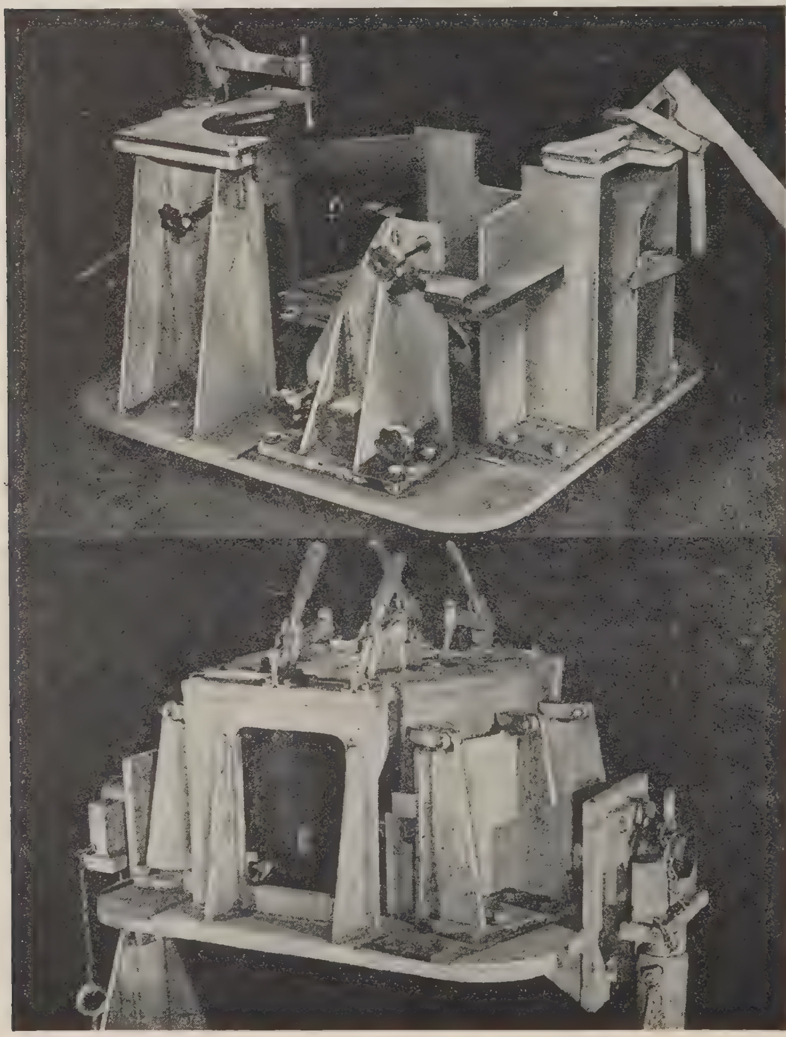


Fig. 20 Two Specially Designed Fixtures for Tooling Weldments in Quantity Production

cal and chemical properties should be specified carefully. Also, where size permits, it is desirable to have castings of electric-furnace steel which seems to possess greater cleanliness. This is important in obtaining good welds with minimum difficulty.

Drop forgings also may be used when their size and quantity justify the investment in dies. This product has good homogeneous properties, and when properly controlled, its tolerances are close. Fig. 19 shows such application where several drop forgings have been joined by flash-welding.

Fabrication of Weldments

Having considered the production and application of components, the subject of fabrication from the designer's viewpoint in developing a weldment is the next consideration. The first aspect covers the type and extent of available equipment in a weldery which will produce the pieces designed. This is important since the more flexible and extensive the equipment, the more freedom there is in design. In addition, when quantities are involved, advantage may be gained in designing the job to suit particular facilities of a weldery. Usually it is well to consult with the engineering staff of the weldery most likely to be involved regarding these matters, particularly if repetitive items are contemplated.

Production methods concerning the fabrication of weldments may be considered from two aspects. The first involves the extent of what might be termed "universal" equipment, such as positioning facilities, automatic welding units, inspection methods,

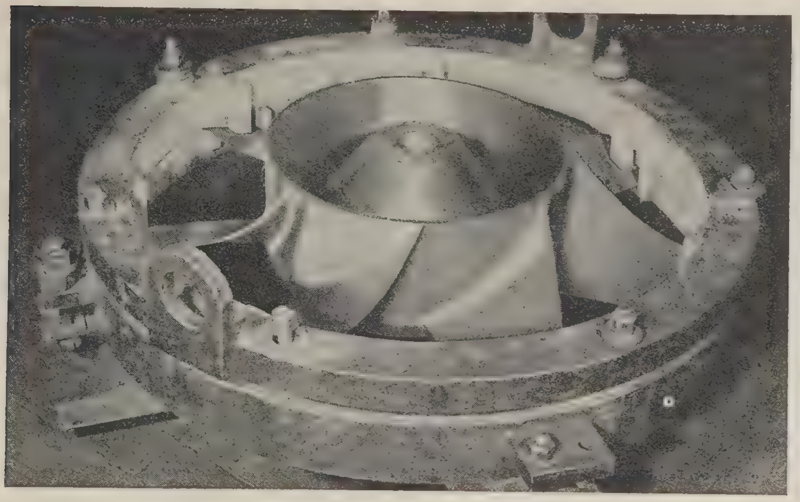


Fig. 21 High-Speed Fan, and Jig in Which It Is Welded

and stress-relieving facilities. Many types and sizes of positioning equipment are in use in welderies today. The other aspect involves special jigs or fixtures or other types of tooling that might be justified or imperative. Usually, if the product is to any degree repetitive in quantity, the possibilities in special tooling should be considered.

Also to be considered by the designer to promote economy and quality is the possibility of breaking up the weldment into subassemblies in sizes to suit production equipment. If the final size of the part as designed exceeds the limits of available equipment, possibly it can be redesigned so that a minimum of handling of the piece in its final size is necessary. Often the use of automatic welding equipment with its finite scope and features merits a thought in designing the weldment, especially in considering the advantage such equipment offers for cost reduction.

Fig. 22 Subassembled Component of a Weldment
(Welding is completed before further assembly is done.)

With special tooling, the designer should keep in mind that he is dealing still with rough component parts despite measures that might have been taken to minimize tolerances. Weldshop tooling naturally is more restricted than that usually available in machine shops. Tools, such as jigs or fixtures, should be designed with the necessity of flexibility in mind. Fig. 20 shows two special fixtures which are typical.

Special tooling also might be mandatory in order to hold components in proper relation to each other during the welding operation. Fig. 21 shows a special fixture with the weldment in place.

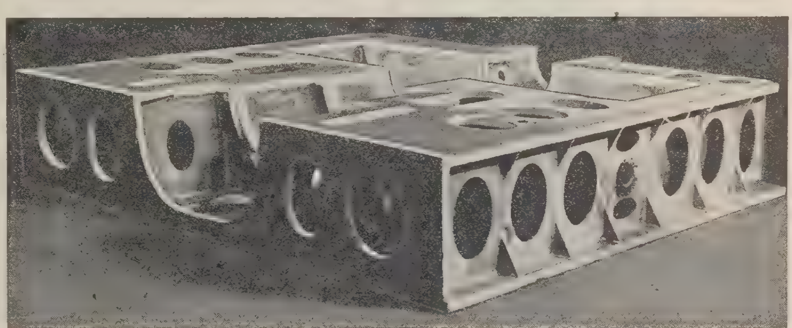


Fig. 23 Lower Section of Welded Gear-Reduction Housing

Although subassemblies frequently are important to the designer of weldments, design limitations often prohibit their use. Obviously, the more work done on small pieces, the easier and quicker will be the completion of the final assembly.

A completely welded subassembly is shown in Fig. 22. The final weldment is pictured in Fig. 23. Here, design controls the method of fabrication, for the lower flange member could be made in one piece. In that case at least a portion of the subassembly welding would have been required on the larger and more cumbersome piece.

Subassemblies of components should be made so that particular portions in certain instances can be sized before they become part of the final weldment. This practice helps insure that the final weldment is close to required dimensions. Where tolerances have accumulated, straightening or trimming might be involved. The effect of the welding on the completed subassembly from the standpoint of warpage or shrinkage has been eliminated as a factor on the finished weldment.

Sometimes in very complicated structures involving considerable welding, various subassemblies are stress-relieved be-

Fig. 24 Welded Subassembly With All Welding Readily Accessible

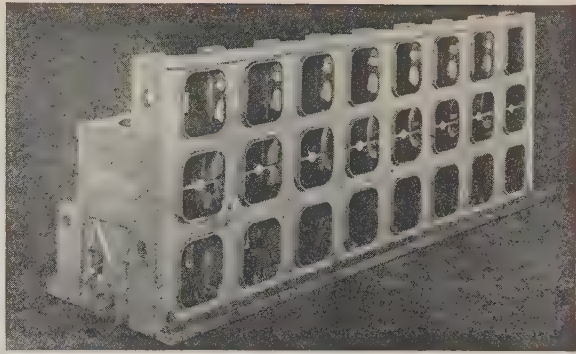


Fig. 25 Weldment of Which Subassembly Shown in Fig. 24 Is a Part

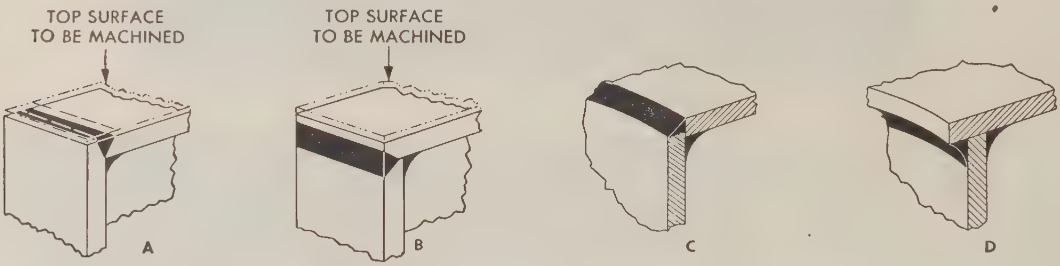


Fig. 26 Types of Welded Joints Which Deserve Design Consideration

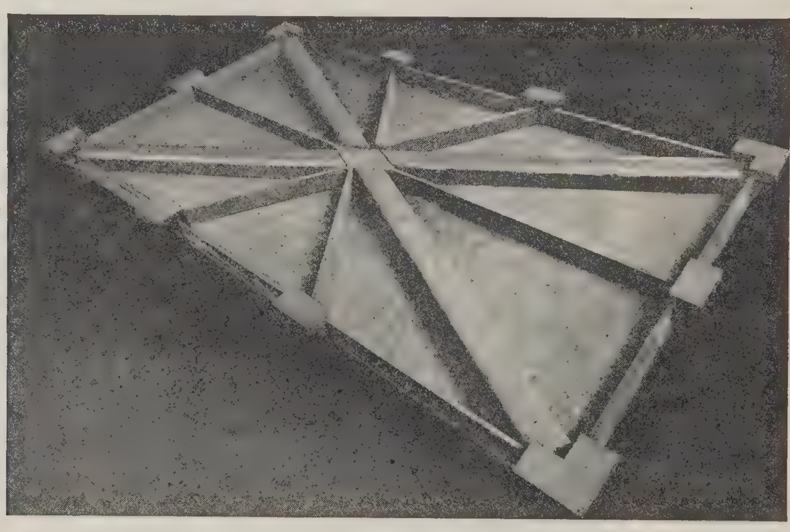


Fig. 27 Stiffener Arrangement on Underside of Machine Bed

fore being assembled into the whole structure to reduce the accumulation of residual stresses. Subassemblies also facilitate inspection of welds. At times, where x-ray inspection is specified, subassembly welding is necessary, for if the welding were not completed and x-rayed in the subassembly, the interference of adjacent components in the completed assembly might make it impossible to x-ray or repair such welds. At times, subassemblies are welded completely when they include compartments subjected to pressure or oil-retention tests.

Tests are made and necessary repairs completed on the subassembly piece. Clearly, such practice is more economical than to attempt such work on the final piece, if only from the standpoint of the relative bulk to be handled in testing and repairing.

An important reason for careful consideration of subassembly possibilities in design is the provision of maximum access for the greatest possible amount of the welding to be done, for the more accessible the welding, the less it will cost. Also, quality is more readily achieved if the welding operator can work under open or accessible conditions. Fig. 24 shows a subassembly on which all welding to be done is before the operator. Fig. 25 shows the

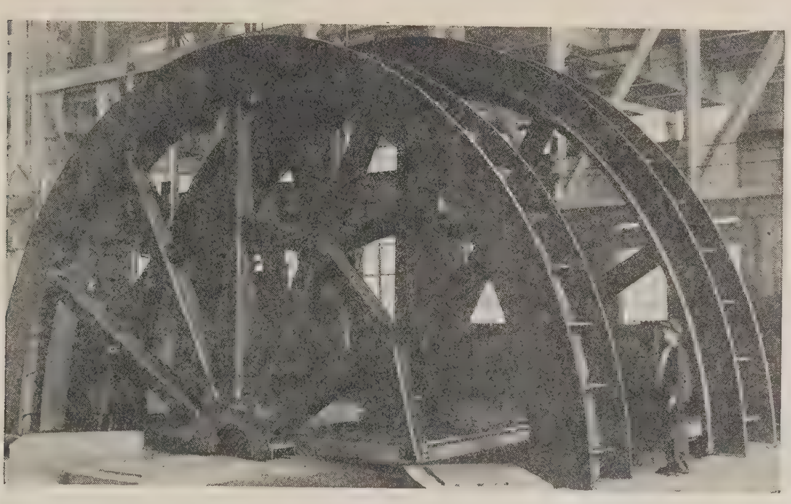


Fig. 28 Two Halves of Large Welded Machine Part Which Are Assembled and Welded Into One Piece After Shipment

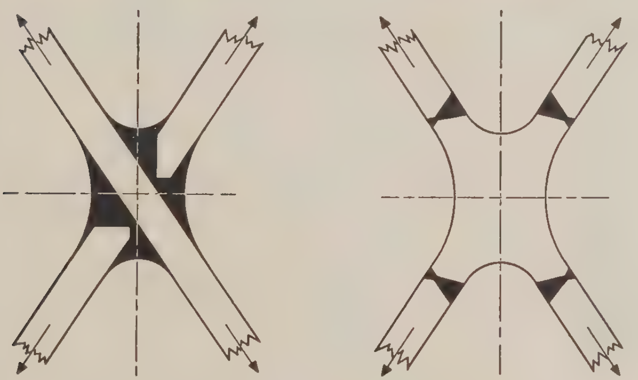


Fig. 30 Types of Intersection Without and With Transition Piece

completed weldment with the subassembly in place. Obviously, welding has been simplified since the joining welds between subassembly and main assemblies are completed by working through the access openings shown. If the welding had not been completed in the subassembly it would have had to be performed through such openings. This would have necessitated the use of a mirror by the welder with resulting slow and consequently expensive welding.

At times it is advantageous to design so that progressive subassembly is possible. Thus one portion of the weldment is completed before it is assembled and welded to other components as a step in the final assembly of the completed weldment.

When maximum access is provided by subassembly practice or other design control, inspection can be more conclusive. Fig. 23 illustrates design for accessibility with elliptically shaped openings, permitting access to the inner side of the joints to be welded. Here desirable structural qualities of a box member are not sacrificed for access but care has been taken in shaping the openings so that abrupt discontinuities in the contour of the members are avoided.

Welded joints of maximum quality and predictability from the standpoint of either external contour or internal soundness are almost impossible to execute with the manual arc if the joint is not reasonably accessible from both sides. However, at certain points

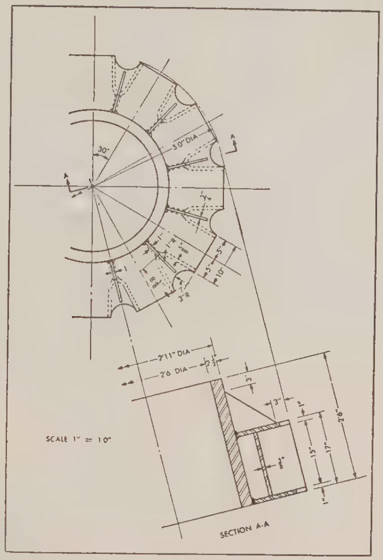


Fig. 29 Detail of Construction of Center Hub on Piece Shown in Fig. 28

where stresses are of secondary nature and fatigue-loading is not present in structures, the joints do not require work on both sides.

In addition to the type of welded joints used, their position in the weldment deserves careful design consideration for several important reasons besides positioning them for maximum access. Where machined surfaces occur in the design, care in placing joints can effect economy, as comparative illustrations given in Fig. 26 show. Clearly, if the joint is placed as in Fig. 26 (A), a portion of the deposited weld metal will be removed in machining operations. Depositing weld metal is expensive and removing it is wasteful. The joint placed in Fig. 26 (B) shows how the amount of necessary weld metal has been reduced. The joint shown need have only the cross-sectional area of that shown in Fig. 26 (A) after it is machined. Economy may be possible by positioning a joint as shown in Fig. 26 (C) which eliminates the kerf and its cost, or Fig. 26 (D) which also simplifies fitting, in contrast to the same joint detailed in Fig. 26 (A).

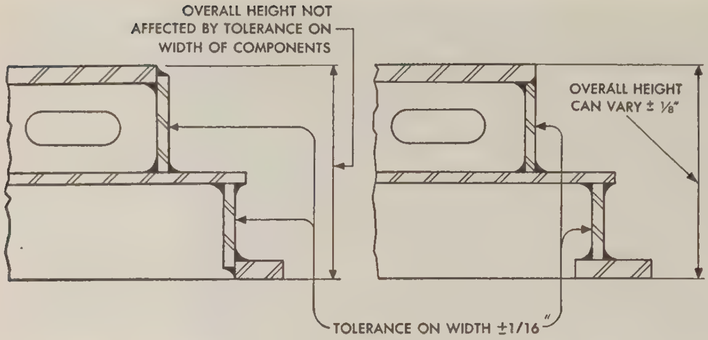


Fig. 31 Effects of Controlling Cumulative Tolerances

At times spoked or diagonal members are indicated by design considerations and their intersection usually presents a type of joint difficult to fit and costly to weld if proper external contours are to be maintained. An example is shown in Fig. 27. Here the diagonal pattern of the box stiffeners on the underside of this machinery bed is highly desirable from the standpoint of maximum rigidity. However, their central intersection presents a problem of the nature just described. By the utilization of a flame-cut central member, square joints at the intersection have been obtained; hence, fitting and welding are simplified.

Fig. 28 shows a simple treatment of an intersection of spokes. The central member in this is a subassembly so designed that the spokes do not converge on each other with obviously poor fitting and welding conditions resulting. An illustrative sketch showing the construction of this central hub is given in Fig. 29.

At intersections of members subject to high stress levels, fatigue, impact, or a combination of these, careful design treatment is imperative. Contrasting designs of such an intersection are shown in Fig. 30, where the left illustration shows the intersection as welded without benefit of a transition member. Clearly, in order to provide curved contours, an inordinate amount of weld metal would be necessary. In addition, it is practically impossible to execute such a welded joint so that full predictable strength will result. If necessary, x-ray inspection is practically impossible. Fitting conditions are poor and the excessive amount of weld metal adds to warpage and shrinkage problems. Desirable features of a similar joint as executed in the right illustration are self-evident.

The designer should remember the generality that in a weldment the fewer separate and different components required, the cheaper and better the design will be.

At times, particularly with secondary members such as ribs or stiffeners, slight changes will permit uniformity in size of different components. Such consideration might make quantity production by blanking economically possible. Another somewhat minor consideration is the difficulty caused the assembler or fitter in trying to identify parts that appear almost the same.

The designer should keep in mind, also, the possible utilization of hot-rolled-bar stock, for often such material can be used by making a slight change in dimension of a given cross section to conform to standards.

All components, regardless of the method of producing them, are subject to dimensional tolerances. The designer must keep this in mind for economy and good fit-up so that he can control the ill effects of cumulative tolerances. Fig. 31 gives a simple illustration of a typical weldment in which the designer has kept cumulative tolerances in mind. The drawing at the left shows a partial cross section, while the drawing at the right shows a similar cross section in which the point has been ignored.

Shrinkage and warpage problems which exist in the production of weldments will continue to be a factor so long as drastic heat gradients occur during welding. A degree of experience is needed to be able to predict such effects and to control and counteract them. It is impossible to discuss this factor here in detail. The designer should, however, sense the general aspects of such phenomena so that he does not develop designs that may be impossible to produce within necessary tolerances.

Warpage will occur to a great or little degree depending on the relative size of given welds and their distances from the neutral axis of the assembly. This is due simply to the relative ability of a member to resist shrinkage stress imposed at different points in its cross section with respect to the neutral axis of that cross section.

Welding results in shrinkage both longitudinally and at right angles to the weld metal. The extent varies for sizes as well as types of welds. When the number of different sizes and types of welds occurring in an average weldment, and their length and position with respect to each other are considered, it can be realized that control of warpage is to a large degree a matter of practical experience. Sequence of welding also should be carefully controlled as a counteracting measure. At times special fixtures are used to restrain warpage during welding.

Conditioning and Inspection Methods

Following completion of welding, it is an established practice to condition the weldment by removing spatter, grinding edges or surfaces where specified because of design requirements, and gritblasting when size permits.

Spatter is removed for appearance and to insure that it will not drop off progressively when the weldment is in service. Spatter can be detrimental mechanically if, for instance, it were left in a lubricating-oil compartment.

Weldments are gritblasted to remove mill scale from plate surfaces and to facilitate visual inspection of welds. Undercuts usually are more difficult to detect before the weld is gritblasted. Welds are inspected visually for proper size, surface cracks or other surface defects. X ray is used to inspect welds for internal defects. Various specifications such as those of the A.S.M.E. and the U. S. Navy provide inspection standards for the acceptance or rejection of welds by means of x-ray photographs.

Hydrostatic testing often is required by design specification. Oiltight compartments should be checked and tested before the part leaves the weldery. Finally, the weldment should be laid out for a final inspection at the weldery to verify that it is dimensionally correct within specified tolerances.

Any weldment to be machined, subsequently, to any appreciable degree should be stress-relieved if the machined surface or other parts of the weldment are to hold their relationship within service life. Any weldment subjected to severe stresses or to fatigue or impact, also should be stress-relieved. Especially is this advisable since locked-up stresses, the magnitude or direction of which cannot be predicted, can be of a high order following welding. If normal service loading imposes design stresses having the same direction at a given point as that of a residual or locked-up weld stress, structural distress or failure can result.

Many weldments are in use that have not been stress-relieved. Hence definite predictions cannot be made that difficulty will result for a given type of weldment in the unstress-relieved state. Stress-relief therefore may be regarded somewhat like insurance having a low premium rate because the per pound cost of stress-relieving is usually only a fraction of a cent.

Specifications for Weldments

A weldment should be specified clearly and concisely so that no misunderstanding results, and to insure from the equitable standpoint, that each potential vendor is quoting on the same conditions. It is important, for instance, that the weldery know if the weldment must be produced under a code requirement.

Over-all tolerance requirements on various dimensions of the weldment should be considered carefully. The designer might be inclined to make everything tight to "be on the safe side." But this can be a costly practice compared to that of evaluating requirements intelligently. Often, a relatively close tolerance, under given conditions of size, shape, and amount of welding, at some point in a weldment can be costly to achieve. Proper study might show that the tolerance can be loosened without affecting the service performance of the weldment. In this instance it is common practice to consult with the weldery to achieve, mutually, the best conditions.

This check list of specifications and drawing information will give the weldery proper and clear directions:

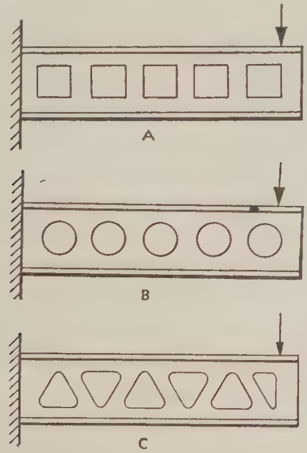


Fig. 32 Cantilever Beam Showing Three Designs in Web Openings

Steel specification?
Inspection agency, if any?
Weld sizes and to what standard?
General tolerances or any special tolerances?
Gritblast?
Grind? If so, where?
Stress relieve?
Paint? If so, what type and how many coats?
X-ray requirements? To what code?
Test—hydrostatic? If so, at what pressure?
Test—oiltightness?

Many types of weldments or components for weldments can be designed effectively into welded machinery parts. Categorically, machinery parts, as distinguished from other types of structural members, can be defined as parts required to resist or support moving forces; parts subjected to dynamic loading.

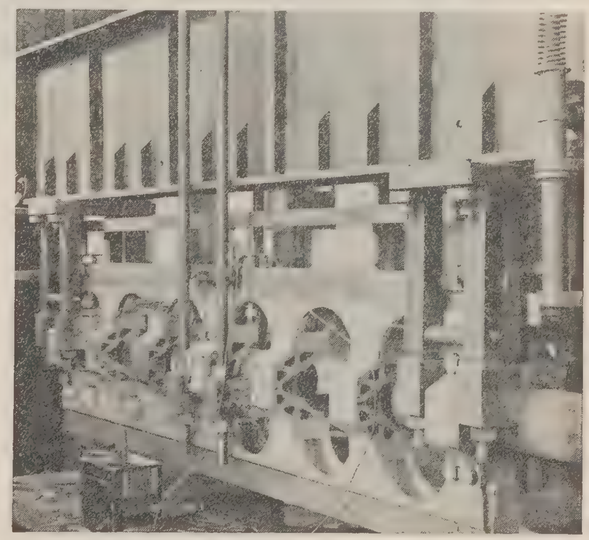


Fig. 34 Analysis on Model of Welded Engine Bed
(Stresses under lateral load can be read from the strain gages visible in the illustration.)

Dynamic loading or applied forces in motion can involve additional factors or criteria, i.e., fatigue and impact, to be considered in the design of the part and the proper selection of its material. These two factors, alone or in combination, often can affect materially, or control the shape, material, and manner of fabrication of a mechanical structural member.

If fatigue and impact are of major consideration, those properties of the material determining its resistance to such loading should be known. As we know, such properties generally are referred to as the endurance limit with respect to fatigue, and impact resistance in relation to impact. This latter property sometimes is related to temperature.

External contours of a part subjected

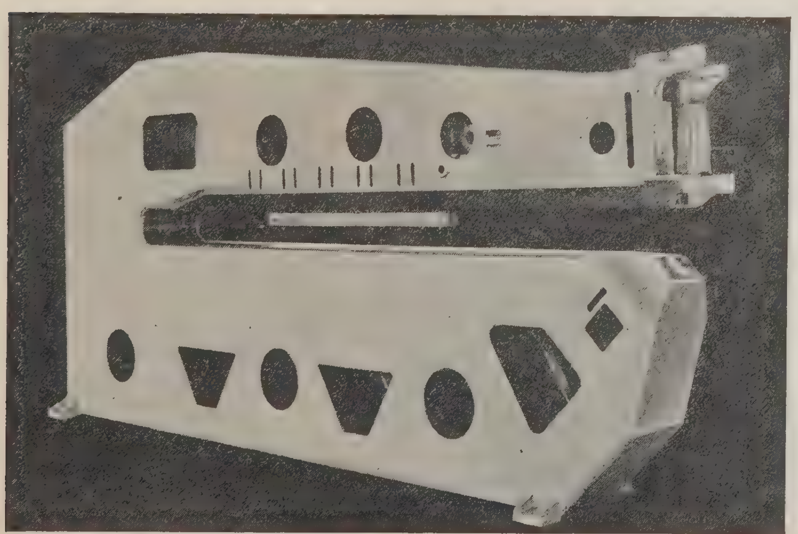


Fig. 33 Circle Shear Frame

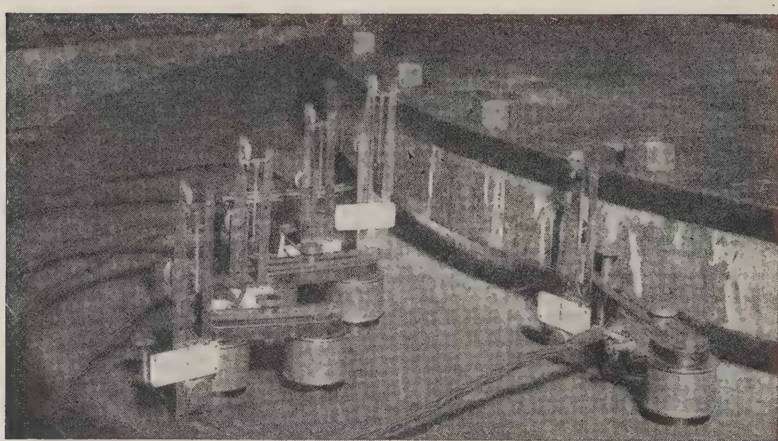


Fig. 35 Test Setup to Determine Actual Stresses Under Load on Piece Subjected to Internal Hydrostatic Pressure

to dynamic loading, particularly when fatigue is probable, are an important consideration especially when the member is subjected to primary loads. They are extremely important at the points of changes in contour in a structure of irregular shape.

Contours influence the effectiveness of a load-carrying member from a general standpoint. Proper contour denotes efficient disposition of material which is fundamental in control of weight. As an example, the component shown in Fig. 32 (A) is less rigid for a given over-all weight than that shown in Fig. 32 (B). Disposing metal as shown in Fig. 32 (C) is more effective from this standpoint. An example of such design thought in an existing welded structure is shown in Fig. 33.

Internal contours, those bounding openings in a structure, are worthy of careful consideration in the design of machine parts, especially since mechanical structures usually require openings of varying sizes and shapes for such reasons as fabricating or operating accessibility, or for mechanical clearance.

Often the effectiveness of proper, flowing contour, particularly its value in rigidity, is mathematically indeterminate. Where maximum rigidity or strength for least weight is important, model analysis is possible. Fig. 34 shows the setup for such an analysis of a welded machine part. Actual stresses in a product also can be determined by the use of instruments such as strain gages shown in Fig. 35.

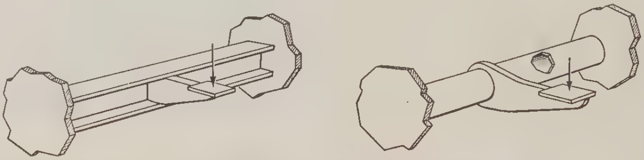


Fig. 36 Abstract Example of Two Methods of Resisting Torsional Force

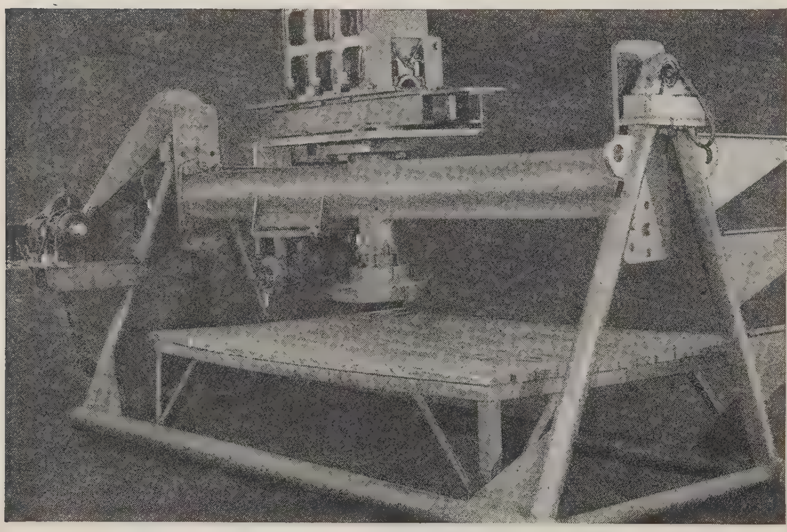


Fig. 37 Application of Welded Tubular Structure
(Main beam of this welding positioner is tubular in cross section. It is subjected to combined bending and torsional stresses.)

Weight Control

Frequently, minimum weight is a major factor in designing structural members of machines, and the weight of a mechanical part can be controlled by one or more considerations. Sufficient strength is the first fundamental. Several considerations can be involved in designing for the strength factor. One is a properly adjusted factor of safety. A minimum factor of safety is important in designing for least weight. If it is known definitely that no internal voids can or need exist in a given cross section subject to primary dynamic stress, one element making up the factor of safety has disappeared, or at least is under control.

This control can be exercised in production of weldments if design permits. It is axiomatic that in designing for least weight the shape should be as regular and flowing as possible, as the previous discussion of contours reveals.

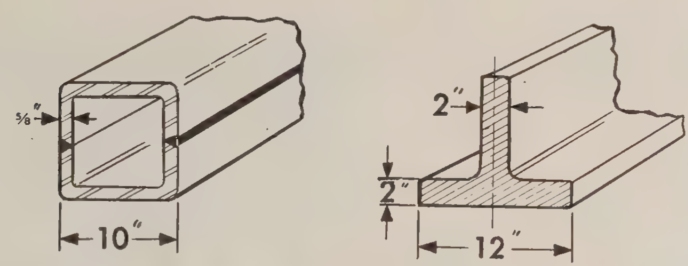


FIG. 38 DESIGN DETAIL OF SPOKES IN A WELDED MACHINE PART

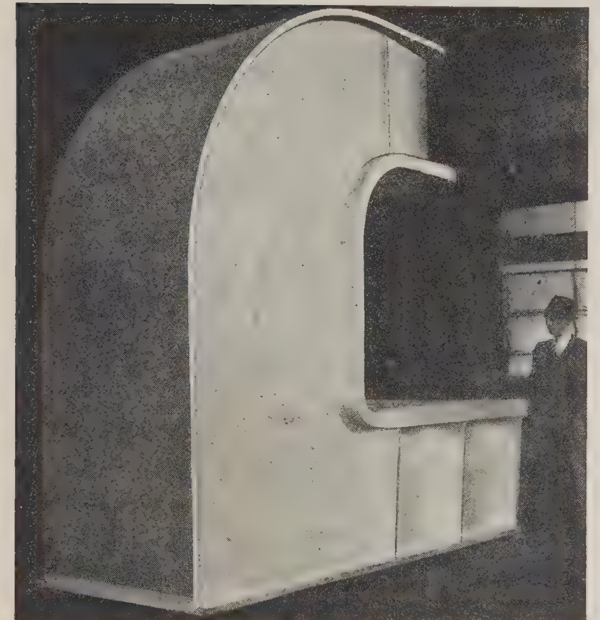


FIG. 39 WELDED FRAME FOR C-TYPE, OR OPEN-SIDE, HYDRAULIC PRESS

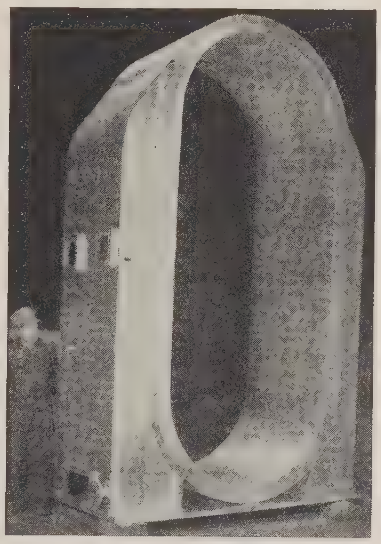


FIG. 40 WELDED-STEEL CONTINUOUS BAND HYDRAULIC-PRESS FRAME

Another consideration in the strength factor is the exactness of knowledge of the magnitude, direction, and location of the dynamic forces with respect to each other and their reactions involved in the machine. In designing machine parts, particularly for least weight, it is important that these be predicted as closely as possible.

After this has been done, the material resisting or supporting such forces or reaction can be disposed in a weldment to best advantage consistent with mechanical clearance requirements balanced against the necessity for least weight.

As an instance, Fig. 36 shows an abstract example of two methods of resisting a torsional force. There is no questioning the effective disposition of metal in a tubular design. Nor is there any questioning that the effective disposition of metal insures least weight. Fig. 37 shows an example of this concept in an existing machine part.

Another example of effective metal disposition is shown in the spokes of an enormous machine part, the cross section of which appears at the left in Fig. 38. The illustration at the right shows the spoke as it was designed initially because of casting limitations.

RIGIDITY A VITAL FACTOR

The dominant factor in the design of structural parts for machines sometimes is rigidity, or stiffness, rather than strength as in the instances just discussed. The primary function of the spokes in Fig. 38 is maintenance of alignment between machined parts in the structure to the greatest possible degree. Rigidity or minimum deformation dominates in the design of the C-type, or open-side press frame shown in Fig. 39. This weldment is composed exclusively of hot-rolled plate with metal thickness at any point dictated solely by design requirements.

A welded design that embraces directional control of rigidity is shown in Fig. 40. Purely vertical deformation or elongation is a secondary consideration in the function of this machine. A somewhat normal design for the structural parts of such a press is composed of top beam or platen, lower beam or bed, and side housings; all separate pieces integrated by two or four vertical tie rods secured by nuts at both ends.

In such a design, it can be seen readily that in addition to vertical, straight-line elongation, the upper and lower beams will tend to deflect as beams in a manner detrimental to the alignment of tools and work in the press. That such detrimental tendency is practically eliminated is shown in this design where the single structural member resists the working forces imposed on it in direct uniform tension.

The stabilization of thin and unsupported expanses of metal is a new element which requires watchfulness in the design of weldments for machinery, since the basic properties of the component materials are more predictable and process limitations on the thinness of metal sections are nonexistent. These thin and un-

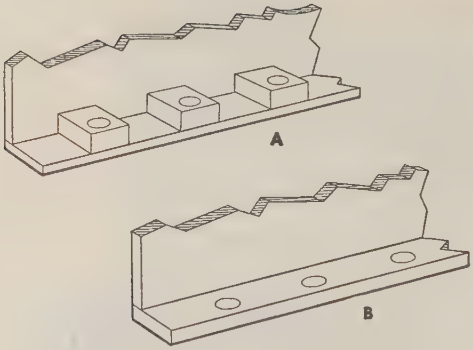


Fig. 41 Comparative Treatments of Bolting Flanges in a Welded Part

supported expanses of metal may be detrimental to the operation of the machine for several reasons. One is the possibility of "drumming" or vibration, another is fundamental instability, and the third is the possibility of dents or distortion in normal handling of the structure.

Many possible methods of stabilizing expanses of cross section exist. Usually the imagination of the designer is fertile in the solution of the problem, if it is recognized.

Size Limitation

Generally, the fewer number of pieces required to make up a machine structure, the less it will weigh and cost, and the stiffer it will be in contrast to a one-piece structure. The only limitations on the size of a weldment are those imposed by stress-relieving facilities and shipping clearance.

Another type of freedom that should be kept in mind by the designer of weldments is the possible use of higher-strength weldable alloys in highly stressed portions of the structure. The secondary components or elements of the piece can be of cheaper, plain carbon steels. Weldable alloys, of course, add nothing in rigidity or stiffness since the modulus of elasticity remains practically the same.

Factors of primary importance in the economic balance or competitive comparison of a welded-steel design must be evaluated clearly. The fundamental reason for assuming that a weldment is the proper medium must be unmistakably defined. This reason may include natural adaptability, and predictability, minimum weight, and initial cost. Having established this, it must be realized that the weldment undoubtedly will not re-

Fig. 42 In These Two Components, the Upper One Was Designed for Minimum Weight by Cutting Out Areas of Full Thickness Between Functional Holes and Filling Them With Thinner Plate. The Lower Component Is Composed of One Flame-Cut Plate

semble a casting in appearance. This does not imply that the welded machinery part will be displeasing to the eye. It means simply that anything produced by a radically different process logically will be different in appearance. If it is not, the design should be reviewed critically because the chances are that it is not economical as a weldment.

A simple instance is shown in Fig. 41 where at A, the common detail of a bolting flange is shown. In a casting many changes in contour do not affect the cost particularly. In a weldment such details necessitate a greater number of components and consequently a larger number of manhours to join them. At B an alternate detail is shown which fulfills the same function. These are examples of actual applications in use in thousands of weldments.

On the other hand, if weight reduction is of critical importance, the over-all economic balance might dictate the added complication in the weldment. A natural question here is, "why not cast it?" The answer has been found to be that if weight reduction is the primary criterion for given requirements of space and over-all size, a weldment can be designed which will weigh less than a casting and not affect the strength or rigidity.

Another comparison between two components of a weldment that perform the same function is shown in Fig. 42. The upper

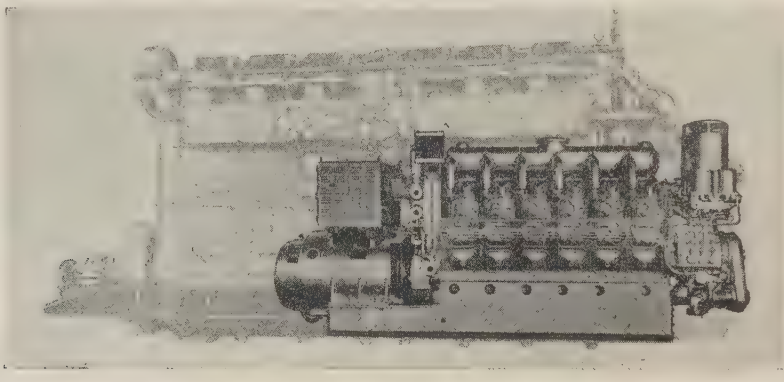


Fig. 43 Comparative Illustration Showing a Direct-Drive 4-Cycle Diesel Engine of 1926 With an Electric-Drive 2-Cycle Diesel Engine of Today

detail is more costly but lighter in weight than that shown in the lower half. If quantities of duplicate parts are involved, other methods of achieving minimum weight at lowest cost also are possible.

In making a decision on the value of weldments as structural members in a machine, the broad measures of their desirability should be known and considered. Logically, a weldment should not be used simply because a desire exists to design in this medium. By the same token a casting or forging should not be used if a weldment presents greater advantages.

In this discussion weight control or reduction may seem to have been emphasized. This is due simply to the progressive recognition of the fact, particularly in relatively large pieces weighing 200 lb or more, that weldments have been applied admirably where weight is important. Undoubtedly, the inherent features making this possible are predictability and the fact that the process practically imposes no limit on the thinness of sections. The weight factor can be important in machine design for many reasons. Fundamentally, it is the minimizing of inertia forces in moving parts of machinery. Another is the pyramiding of the effects of weight or mass in mechanisms subject to high velocity, as the modern railroad passenger car.

Conclusion

In conclusion, the result of the practical application of the many factors discussed probably can be best illustrated by a particular instance of weight and space efficiency in the use of weldments. A comparison, in Fig. 43, of a direct-drive four-cycle Diesel engine of 1926 with an electric-drive two-cycle Diesel engine of today shows that the modern smaller engine represents a reduction of 67.39 per cent in cubic feet occupied per brake horsepower, and also a reduction of 58.40 per cent in weight per brake horsepower. The development of welded-steel-type construction has materially helped in reducing weight. At the same time, the lighter engine of today is 50 per cent more efficient than the heavier engine of 20 years ago.

Stress-Rupture Characteristics of Various Steels in Steam at 1200 F[1]

By J. T. Agnew,[2] G. A. Hawkins,[3] and H. L. Solberg[4]

Small tensile specimens made from low-carbon, carbon-moly, 2¹/₄ Cr- 1 Mo, 5 Cr-Mo-Si, 9 Cr-Mo-Si, 12 Cr, 18 Cr-8 Ni, 25 Cr-20 Ni, and 5 Cr-Mo-Ti steels were placed in a steam reaction chamber at 1200 F and stressed in tension for periods of time ranging from 10 hr to 7700 hr. Data were taken on time to rupture, elongation, reduction in area, depth of scale layer, effect of type of flow, and type and angle of fracture. A photomicrographic study was made of the ruptured specimens. The straight-line relationship between stress and time to rupture on log-log co-ordinates postulated by White, Clark, and Wilson for tests in air also holds for tests in steam.

As the conditions of temperature and pressure to which alloy steels are exposed in modern steam power plants and other industrial apparatus become more severe, the need for experimental data concerning the behavior of these alloys under such conditions becomes more acute. This paper describes an investigation in which small steel specimens were placed in tension in an atmosphere of steam at 1200 F until rupture occurred. The stress-rupture data for various alloy steels in contact with high-temperature steam are needed for the design of various types of high-temperature steam-generating equipment.

Description of Apparatus

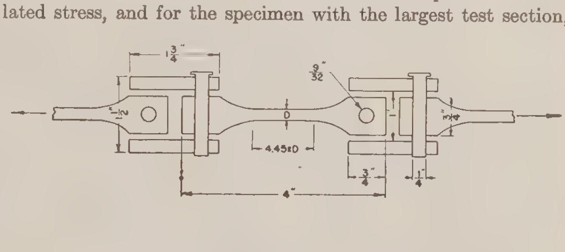
Small steel specimens were placed in tension inside four horizontal steel reaction chambers, each of which consisted of a 10-ft length of extra-heavy 2-in. steel pipe. These reaction chambers were uniformly heated externally to the test temperature by means of resistance heaters. The steam entering the reaction tubes was presuperheated to test temperature by means of gas-fired superheaters and, after leaving the reaction chambers, was conducted to water-cooled copper-tube condensers. In each reaction chamber, several specimens were connected end to end by universal joints and were loaded in tension by means of rods which passed through packing glands at either end of the reaction tubes. One end of each tension rod passed through a calibrated spring which, when compressed, caused the specimens to be placed in tension. Chromel-alumel thermocouples were used to measure the temperature at several positions along each reaction tube. The temperatures were evaluated by means of a Leeds & Northrup portable potentiometer.

The chemical analysis and heat-treatments of the various steels used in these tests are reported in Table 1. The numbers used to designate a specimen, for example, 913, have the following significance: The first number is the type of steel, as given in Table 1, the second number is the test number, and the third (or third and fourth) number is the specimen number. Therefore, 913, would be the third sample of 5 Cr-Mo-Ti steel used in test No. 1.

TABLE 1 CHEMICAL ANALYSIS AND HEAT-TREATMENT FOR STEELS USED IN THE TESTS

Type of steel	Type number	C	Mn	P	S	Si	Mo	Cr	Ni	Ti	Heat-treatment
		Chemical Analysis									
Low carbon	1	0.16	0.47	0.084	0.023	0.20					1650 F
Carbon-moly	2	0.20	0.53	0.01	0.016	0.24	0.50				Air cooled - roll. temp. strain tempered from 650C
2.25 Cr-1 moly	3	0.16	0.39	−0.03	0.03	0.33	0.90	2.24			An. 1550 F
5 Cr-Mo-Si	4	0.10	0.38	−0.03	0.03	1.55	0.51	4.83			An. 1550 F
9 Cr-Mo-Si	5	0.12	0.44	−0.03	0.03	0.67	0.95	9.5			Nor. 1750 F T. 1500 F 6 Hr
12 Cr	6	0.11	0.41	0.014	0.013	0.286		12.66	0.385		O.Q. 950 C Air cooled-625-635C
25–20	7	0.11	0.58	−0.03	0.03	0.75		23.6	20.65		Nor. 1700 F
18–8	8	0.06	0.50	−0.03	0.03	0.61		17.75	9.25		W.Q. 2000 F
5 Cr-Mo-Ti	9	0.10	0.44	−0.03	0.03	0.38	0.50	4.98		0.51	An. 1650 F

Test specimens similar to the one shown in Fig. 1 were machined from round stock of 1 in. diam. The dimensions of the straight central section of the test specimens were in the same proportions as the standard tensile specimen which is 2¹/₄ in. long and 0.505 in. diam. The straight central section of each specimen was machined to a tolerance of ±0.01 in. in length, and the diameter of the test section was held to ±0.001 in. For the specimen with the smallest test section, 0.138 in. diam and 0.61 in. long, this could result in an error of ±400 psi in the calculated stress, and for the specimen with the largest test section,



DIMENSION "D" DETERMINED BY STRESS DESIRED WITH ABSOLUTE LOAD OF 500 LBS

Fig. 1 Schematic Drawing of Test Specimen and Couplings

[1] Based on a doctoral thesis, Purdue University, Lafayette, Ind., 1944, by one of the authors.[2]

[2] Assistant Professor of Mechanical Engineering, Purdue University.

[3] Professor of Thermodynamics, Purdue University. Mem. A.S.M.E.

[4] Head, School of Mechanical and Aeronautical Engineering, Purdue University. Mem. A.S.M.E.

Contributed by the Power Division and presented at the Critical Pressure Steam Boiler Session of the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

0.366 in. diam and 1.63 in. long, an error of ±30 psi in the calculated stress would be possible. Holes $^9/_{32}$ in. diam were drilled in the large ends of the specimens, as shown in Fig. 1, for the purpose of allowing several specimens to be coupled together. Eight specimens were coupled together and placed in each tube by means of collars and pins shown in Fig. 1. The collars consisted of $1^3/_4$-in. lengths of $1^1/_2$-in-OD × 1-in-ID 18 Cr-8 Ni stainless-steel tubing. Stainless-steel pins $^1/_4$ in. diam were passed through the holes in the collars and specimens to hold the specimen train together. The pinholes in each collar and specimen were placed 90 deg apart around the circumference in order that each collar might act as a universal joint. The end collars of the specimen train were drilled and tapped with a standard $^1/_4$-in. pipe thread. Tension rods were screwed into the end collars and extended to the outer steel framework. The outer ends of the tension rods passed through holes in parallel I-beams which formed the end pieces of a rectangular frame around the reaction chambers. A collar, thrust bearing, and nut were placed on the end of the extension rod nearest the superheater to transfer the tension load to the I-beam frame. The tension rod at the other end of the specimen train passed through a calibrated coil spring which was compressed against the I-beam frame by means of a collar, thrust bearing, and adjusting nut on the tension rod. The springs were compressed to exert a constant force of 500 lb on the train of specimens.

By using specimens whose test sections were of various diameters, any desired value of stress could be obtained. The diameters and stresses used for the various specimens are indicated in Tables 2 and 3. The compression of the calibrated loading springs was checked continuously by means of gage bars, and the springs were held to the correct length with an accuracy of ± 0.005 in. so that the absolute force exerted by the springs was 500 lb ± 4 lb. For a specimen 0.2 in. diam this would mean a stress of 15,900 ±100 psi. A stop was provided to restrict the movement of the broken assembly to not more than $^1/_2$ in. Approximately 45 min were required to remove a ruptured specimen and replace it with a new one. The temperature at one end of the tube dropped to approximately 1150 F during this operation but returned to 1200 F within 1 hr after a new specimen was inserted.

Test Program

It was decided to conduct a short preliminary test on several specimens of the same steel to investigate the effect of type of steam flow and pressure upon the corrosion rate. Eight specimens of 5 Cr-Mo-Ti steel, designated as series 9 in the tests, were machined so that, with a load of 500 lb, four of the specimens would be under a stress of 6900 psi, and the other four under a stress of 6200 psi. One specimen corresponding to each stress was placed in each reaction chamber so that each chamber contained two specimens. The temperature, pressure, and steam-flow conditions were maintained as indicated in Table 2. The results of the test are also given in Table 2.

Specimens 916, 917, and 918, under a stress of 6200 psi, were not ruptured during the test. While waiting for specimens 911, 912, 913, 914, and 915 to break, additional specimens of three different types of steel were inserted into the reaction tubes to gather what additional stress-rupture data could be obtained. The results obtained from these additional specimens are included in Table 2. Since the results from this preliminary test were not conclusive, it was decided to maintain different flow and pressure conditions in the four reaction chambers during the remainder of the test program with several samples of each steel being exposed to the various conditions.

On the basis of the data of the relation of stress to time to rupture in steam, as determined from the preliminary test, the second or regular test was started with eight specimens in each reaction tube. The results of this test are given in Table 3. Flow rates were adjusted to maintain laminar flow with a Reynolds number of 1100 in the reaction chambers containing steam at 5 psig, and turbulent flow with a Reynolds number of 5500 in the reaction chambers in which steam was used at 75 and 100 psig, as indicated in Table 3. Each time a specimen ruptured it was replaced by a new specimen. A complete history of each specimen including temperature, type of steam flow, stress, time to rupture, type and angle of fracture, elongation in area, etc., is given in Table 3. The stress-rupture results are plotted in Fig. 2, with stress as the ordinate and time to rupture as the abscissa on log-log co-ordinates.

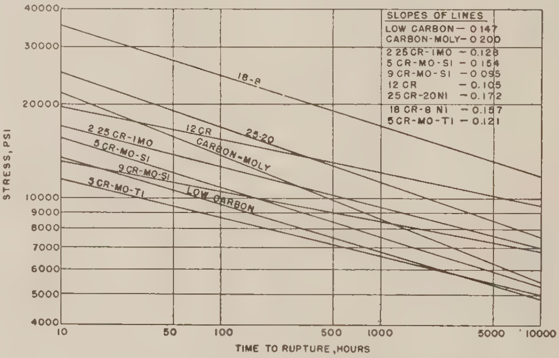


Fig. 2 Effect of Time on Corrosion of Nine Alloy Steels Exposed to Steam at 1200 F Under Stress

TABLE 2 RESULTS OF PRELIMINARY STRESS-RUPTURE TESTS

(Temperature: 1200 F ± 10 F)

Test No. 1
Test chamber *W* — Re = 1100, P = 130 psig
Test chamber *S* — Re = 5500, P = 130 psig
Test chamber *N* — Re = 5500, P = 75 psig
Test chamber *E* — Re = 1500, P = 5 psig

Spec. no.	Test chamber	Time for rupture, hr	Stress, psi	Elongation, ± 2 per cent	Reduction in area, ± 3 per cent	Depth of scale layer, in.	Type and angle of fracture D—Ductile B—Brittle
911	W	318	6900	88	77	0.0044	D — 45°
912	S	833	6900	79	63	0.0066	D — 45°
913	N	869	6900	104	71	0.0072	D — 45°
914	E	1010	6900	90	71	0.0105	D — 60°
915	W	545	6200	71	74	0.0049	D — 45°
916	S		Not broken				
917	N		Not broken				
918	E		Not broken				
711	W	159	14300	32	38	<0.0003	B — 90°
511	W	113	10800	89	81	0.0007	D — 80°
512	W	256	9600	68	82	0.0013	D — 85°
811	W	1305	16000	8	19	<0.0003	B — 90°

The elongation of the ruptured specimens is reported in Tables 2 and 3 as a percentage of the original length of the specimen and was determined by measuring with a pair of dividers the length of a straight section of each test specimen between two punch marks before insertion into the reaction chamber and repeating the measurement on the ruptured specimen with the two sections of the broken specimen placed as close together as possible in the positions they occupied prior to rupture.

The reduction-in-area data presented in Tables 2 and 3 are reported as the ratio of the difference between the original

TABLE 3 RESULTS OF STRESS-RUPTURE TEST NO. 2
(Temperature: 1200 F ± 10 F)

TEST NO. 2

SPECIMEN NUMBER	TIME FOR RUPTURE, HOURS	STRESS, PSI	ELONGATION, ±2%	REDUCTION IN AREA ± 3%	DEPTH OF SCALE LAYER, INCHES	STEAM PRESSURE, PSIG	TYPE OF FLOW EXPOSED TO L-LAMINAR T-TURBULENT	TEMPERATURE	VISUAL DETERMINATION OF TYPE AND ANGLE OF FRACTURE D-DUCTILE I-INTERMEDIATE B-BRITTLE	INITIAL LENGTH OF TEST SECTION, INCHES	INITIAL DIAMETER OF TEST SECTION, INCHES
121	3588	5770	19	29	SCALE SHATTERED	75	T		B-45°	1.48	0333
122	4671	4740	12	50		100	T		B-90°	163	0366
126	10	12500	26	61	00047	5	L		D-45°	100	0225
127M*	57	9500	37	66	00075	5	L		D-45°	115	0259
128M	75	8200	24	59	0.0101	5	L		D-45°	1.24	0279
129M	1261	7000	20	42	0.0367	100	T		B-80°	134	0301
1210	27	16000	30	54	0.0045	5	L		I-90°	088	0.199
1211	857	6800	21	41	0 0288	5	L		B-45°	1.36	0306
221	63	15200	32	46	0 0080	5	L		B-90°	0.91	0.204
222	151	12600	36	42	0.0124	5	L		B-60°	1.00	0.224
223	696	10700	20	29	0.0182	100	T		B-85°	1.08	0.243
224	253	9550	36	46	0 0218	5	L		B-70°	1.15	0.259
225	760	8400	20	47	0.0322	5	L		B-45°	1.23	0.276
226	3047	7800	13	35	0.0596	75	T		B-80°	1.27	0.285
227	1301	7400	23	41	0.0398	5	L		B-60°	1.30	0.292
228	7587	6900	16	47		75	T		B-80°	1.34	0.302
229		5900								1.47	0.329
2210	144	16000	20	40	0.0101	75	T		I-90°	0.88	0.199
2212M	312	9100	29	37	0.0208	5	L		B-90°	1.18	0.265
321	226	11300	62	89	0.0164	5	L		D-45°	1.05	0.236
322	1114	10400	45	80	SCALE SHATTERED	100	T	ALL TEMPERATURES 1200F ± 10F	D-80°	1.10	0.247
323	2884	9200	21	70		5	T		D-45°	1.17	0.263
324		8600								1.21	0.271
325	1908	8400	41	89	SCALE SHATTERED	5	L		D-90°	123	0.276
326	1519	8400	61	88	SCALE SHATTERED	100	T		D-90°	1.23	0.276
327		7750								1.28	0.287
328	46	14400	49	89	0.0075	5	L		D-90°	0.94	0.210
329	50	13000	71	91	0.0079	5	L		D-60°	0.98	0.220
421	154	9550	117	91	0.0016	75	T		D-85°	1.15	0.259
422	209	8500	113	84	0.0058	5	L		D-60°	1.22	0.273
423	1237	7750	67	78	0.0069	100	T		D-45°	1.28	0.287
424	1709	7300	68	71	0.0088	75	T		D-45°	1.32	0.295
425	2522	6700	56	66		100	T		D-45°	1.37	0308
426		6450								1.40	0.314
427		6400								1.41	0.316
428	4007	6100	60	56		100	T		I-45°	1.44	0.323
429	3613	5450	52	55		5	L		I-90°	1.52	0.341
4210	142	11000	158	89	0.0029	100	T		D-70°	1.07	0.240
4211	1166	7600	61	75		75	T		D-60°	1.29	0.289
4212M	1340	6900	82	65	0.0075	75	T		D-45°	1.35	0.303
521	56	12000	114	86	0.0006	5	L		D-90°	1.02	0.230
522	83	11000	85	84	0.0007	5	L		D-90°	1.07	0.240
523	287	9100	107	86	0.0013	5	L		D-80°	1.18	0.264
524	258	8600	155	84	0.0016	5	L		D-85°	1.21	0.272
525	1734	8300	95	81	0.0019	75	T		D-85°	123	0277
526	1970	8300	84	78	0.0023	75	T		D-70°	1.23	0.277
527		7650								1.29	0.289
528	616	9100	87	83	0.0018	100	T		D-90°	118	0265
529	946	8800	78	83	0.0016	100	T		D-90°	1.20	0.270
5210M	498	8100	94	82	0.0013	5	L		D-85°	125	0280
621	164	15100	30	27	0.0007	5	L		B-90° CUP	091	0.205
622	624	13700	38	48	0.0008	100	T		B-90°	096	0.215
623	942	12300	43	54	0.0009	100	T		B-85°	1.01	0.226
624	260	11800	51	68	0.0009	5	L		I-80°	1.03	0231
625	538	11000	53	71	0.0012	5	L		I-80°	107	0.240
626	2284	10700	68	81		75	T		D-90°	1.09	0.244
627	2792	10400	71	78		75	T		D-90°	1.10	0.247
628	2517	10000	59	79	0.0017	75	T		D-85°	112	0.252
629M	15	17750	58	92	0.0002	5	L		D-90°	085	0.190
6210	339	16100	30	46	0.0011	100	T		I-70°	088	0.198
6211	899	13000	20	53		100	T		I-80°	0.98	0.220
6212M	569	11700	57	88	0.0012	75	T		D-85°	1.03	0232
721	937	12500	58	36	> 0.0001	75	T		B-90°	1.00	0.224
722	1552	9950	75	49	> 0.0001	100	T		B-90°	1.13	0.253
723	3810	8400	70	42		5	L		B-90°	1.23	0.276
724	107	17000	49	31	> 0.0001	5	L		B-90° CUP	0.86	0.193
725	177	15300	40	41	> 0.0001	5	L		B-80°	0 90	0203
726	977	11000	64	44	> 0.0001	75	T		B-90°	107	0240
821	263	19000	36	33	> 0.0003	5	L		B-85°	081	0.183
822	7720	14000	15	11		75	T		B-85°	095	0213
823M	151	22600	28	31	> 0.0003	5	L		B-90°	0.75	0168
824	15	33400	23	25	> 0.0003	5	L		B-90° CUP	061	0138
825	45	27600	51	38	0.0008	5	L		B-90°	068	0152
826	1145	17000	43	31	0.0013	75	T		B-80°	086	0193
921	182	9400	83	71	0.0038	100	T		D-45°	126	0260
922	291	7000	81	66	0.0088	5	L		D-45°	134	0.301
923	2606	6300	77	61	0.0102	75	T		I-45°	142	0.318
924	3109	5950	52	49		75	T		B-45°	1.46	0.328
925	55	9700	61	70	0.0049	5	L		D-45°	1.14	0.256
926	288	8000	55	59	0.0084	5	L		I-?	1.25	0.281
927	1100	7000	45	49		100	T		I-45°	1.34	0.301

*THE LETTER "M" REFERS TO A SPECIMEN WHOSE TEST SECTION WAS MACHINED FROM A PREVIOUSLY TESTED UNBROKEN SPECIMEN (SEE DISCUSSION OF RESULTS)

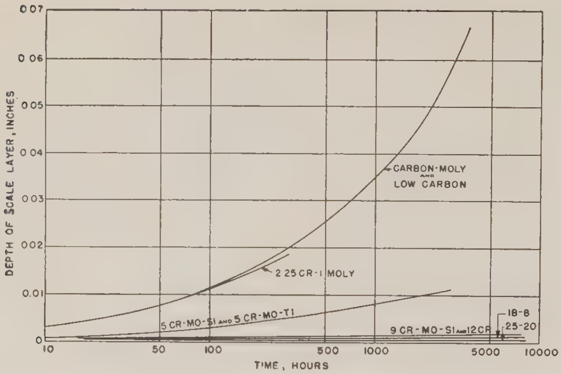


Fig. 3 Stress-Rupture Characteristics of Nine Alloy Steels in Steam at 1200 F

and final areas to the original area in per cent and are based on a measurement of the original diameter of the specimen by a micrometer and a measurement of the final diameter, not including the scale layer, by a pair of dividers.

Due to the size and shape of the specimens it was not possible to determine the loss of metal due to corrosion by electrolytic stripping as was used in previous corrosion studies (1, 2, 3, 4, 5)[5] at Purdue University. Some of the specimens were exposed to the steam for a short time interval, thereby producing a thin layer of oxide which could not be removed by the electrolytic stripping method. For the small specimens used in these stress tests a satisfactory method of determining the penetration of the scale layer was developed. A small section of the scale-covered ruptured specimen approximately 1/4 in. long was cut off by means of a cut-off wheel. This section was molded in bakelite and polished. By means of a precision microscope it was possible to measure the thickness of the scale layer quite accurately. The results of the corrosion measurements are given in Tables 2 and 3, and in Fig. 3.

Discussion of Results

The slopes of the stress versus time-to-rupture lines for the various alloys, as plotted in Fig. 3, are as follows: Low carbon, 0.147; carbon-moly, 0.200; 2 1/4 Cr-1 Mo, 0.128; 5 Cr-Mo-Si, 0.154; 9 Cr-Mo-Si, 0.095; 12 Cr, 0.105; 25 Cr-20 Ni, 0.172; 18 Cr-8 Ni, 0.157; and 5 Cr-Mo-Ti, 0.121. The significance of these various slopes is that a steel with a large slope would not be expected to have a high rupture stress for long time intervals such as 100,000 hr. Table 4 presents the stress required to rupture each of the steels for time intervals of 10, 1000, and 10,000 hr.

Of the nine steels tested, carbon moly is sixth in order at 10,000 hr, whereas it was third at 10 hr. 9 Cr-Mo-Si was eighth at 10 hr, is fifth at 10,000 hr and would be third at 100,000 hr if the straight-line relationship continued to hold.

In general, the stress required for rupture in a given time in an atmosphere of steam at 1200 F is higher than the stress required for rupture in the same time in air at 1200 F, the air values being taken from the data of the Timken investigators (6, 7, 8). The nine alloys which were included in this investigation are listed in Table 5, in the order of decreasing magnitude of the differences between the stress in steam and the stress in air required for rupture in 10,000 hr. The magnitude of the ratio of stress in steam to stress in air is indicated following each alloy.

[5] Numbers in parentheses refer to the Bibliography at the end of the paper.

TABLE 4 STRESS REQUIRED TO CAUSE RUPTURE OF VARIOUS ALLOY STEELS AT 1200 F

	In 10 hr		In 1000 hr		In 10,000 hr[a]	
(1)	18 Cr-8Ni	35000	18 Cr-8 Ni	16800	18 Cr-8 Ni	11600
(2)	25 Cr-20Ni	25000	12 Cr	12000	12 Cr	9400
(3)	Carbon-moly	21500	25 Cr-20 Ni	11200	25 Cr-20 Ni	7550
(4)	12 Cr	19500	2 1/4 Cr-1 Mo	9400	2 1/4 Cr-1 Mo	7000
(5)	2 1/4 Cr-1Mo	17000	Carbon-moly	8600	9 Cr-Mo-Si	6800
(6)	5 Cr-Mo-Si	15500	9 Cr-Mo-Si	8500	Carbon-moly	5450
(7)	Low carbon	13500	5 Cr-Mo-Si	7600	5 Cr-Mo-Si	5300
(8)	9 Cr-Mo-Si	13200	Low carbon	6900	5 Cr-Mo-Ti	5000
(9)	5 Cr-Mo-Ti	11500	5 Cr-Mo-Ti	6700	Low carbon	4900

[a] Extrapolated.

TABLE 5 DIFFERENCES BETWEEN STRESS IN STEAM AND STRESS IN AIR REQUIRED FOR RUPTURE IN 10,000 HR OF VARIOUS ALLOYS

	Type of steel	Ratio of stresses in steam to air
1	Low carbon	3.6
2	12 Cr	3.2
3	Carbon moly	1.8
4	25 Cr-20 Ni	1.4
5	5 Cr-Mo-Si	1.2
6	18 Cr-8 Ni	1.2
7	2.25 Cr-1Mo	1.2
8	9 Cr-Mo-Si	1.1
9	5 Cr-Mo-Ti	0.98

The results of the attempt to determine the effect of turbulent or streamline flow upon the stress-rupture results were inconclusive and no definite conclusions could be drawn concerning this effect.

The results of the measurement of the thickness of the scale layers are given in Tables 2 and 3, and in Fig. 3. As in previous tests at Purdue University (1, 2, 3, 4, 5), the chromium content of the alloy is the critical factor in the determination of the depth of corrosion. The alloys can be divided into four general groups relative to the type and extent of corrosion. The two alloys, low carbon and carbon-moly, show serious corrosion, having a thick, porous but tightly adhering layer of scale. Alloys 2 1/4 Cr-1Moly, 5 Cr-Mo-Si, and 5 Cr-Mo-Ti show appreciable corrosion and have a brittle flaky layer of scale. Alloys 9 Cr-Mo-Si and 12 Cr exhibit a slight amount of corrosion, while the corrosion of alloys 18 Cr-8 Ni and 25 Cr-20 Ni is very slight. It is to be noted that the corrosion results of the stress-rupture tests cannot be directly compared to the corrosion results obtained previously at Purdue University (1, 2, 3, 4, 5), since in the previous tests the results were reported as penetration in inches as computed from measured loss of metal, whereas in the stress-rupture tests the results are reported as depth of scale layer on the ruptured specimen. An examination of the data reveals no correlation between type of steam flow and depth of scale layer.

A comparison of the types and characteristics of the fractures of the various alloys is of considerable interest in that it gives some indication of the type of fracture that can be expected in service. A discussion of the fracture of each alloy will be given. A metallographic study was made of the original microstructure and the microstructure of two fractured specimens of the longest and shortest fracture time and the following comments are a result of these studies:

(*a*) Low-Carbon Steel (Alloy No. 1). Of the eight specimens tested, three had ductile fractures, four had brittle fractures, and one had an intermediate fracture. However, the values of ductility as measured by elongation were consistently low (from 12 to 37 per cent) with the ductile fractures giving the greater elongation. By an intermediate fracture is meant a fracture with just a slight amount of necking at the break as compared to no perceptible necking for a brittle fracture and considerable necking for a ductile fracture. This method of classification holds for all nine alloys. The three low-carbon specimens with the ductile fractures, 126, 127M, and 128M, and the one with the intermediate fracture, 1210, had the largest elongation and reduction of area at rupture. The short-time ductile fracture of specimen

126 was characterized by stretching of the grains while the long-time brittle intergranular fracture of specimen 121 was characterized by the remaining equi-axed grains pulling apart at the grain boundaries. In both the long- and short-time specimens a dark intergranular material appeared which was probably an oxidation product together with voids resulting from parting of the grains. Spheroidization of the carbides present in the original sample occurred in both the long- and short-time specimens of this alloy with considerable agglomeration of the spheroids present in the long-time specimen.

No clearly defined graphite was detected in the structure of the specimens.

(*b*) Carbon-Moly (Alloy No. 2). All of the carbon-moly specimens had the brittle type of fracture except specimen 2210 which showed a slight amount of necking and was classified as having an intermediate fracture. All of the specimens had low values of ductility as measured by elongation (13 to 36 per cent). As in the case of the low-carbon alloy, the only appreciable change in the microstructure was general spheroidization of the carbide structure. Very little agglomeration was present in the microstructure of the short-time specimen, 221, the original normalized structure still being somewhat in evidence. Considerable agglomeration of the carbides was present in the long-time specimen, 226. Parting of the grains had occurred in both specimens 221 and 226, but to a greater extent in the short-time specimen 221. This parting resulted in intergranular fracture for both the long- and short-time intervals and also resulted in the presence of dark areas between the grains, probably consisting both of oxidation product and voids.

No clearly defined graphite was observed in the microstructure of the carbon-moly specimens.

(*c*) $2^1/_4$ Cr-1 Mo (Alloy No. 3). In contrast to the carbon-moly steel, all of the specimens of $2^1/_4$ Cr-1 Mo steel had the ductile type of fracture as evidenced by the higher values of ductility as measured by elongation (21 to 71 per cent). Without exception, all of the specimens necked considerably at the break, resulting in high values for reduction in area. Another peculiarity of this alloy was the very brittle flaky type of scale which curled away from the parent metal as the corrosion progressed, thus continuously leaving fresh metal exposed to the steam. Due to the excessive necking at the break, the metal there should have been very highly strained. The microstructure of the short-time specimen 328 showed considerable elongation of the grains. The microstructure of the fractured specimens showed coalescence of the carbides which were present in the original steel before testing. The degree of coalescence was considerably higher in the long-time specimen 325 than in the short-time specimen 328. The specimens showed more than normal inclusions which, however, did not appear to be graphite when examined unetched. Since there was evidence of some inclusions in the original microstructure of this steel, it is not believed that the inclusions present in the microstructure of the fractured specimens is due to any appreciable extent to oxidation and stress. This steel is to be contrasted to the low-carbon and carbon-moly steels which did show parting of the grains and a resultant dark constituent at the grain boundaries.

(*d*) 5 Cr-Mo-Si (Alloy No. 4). This alloy was characterized by a very ductile fracture resulting in high values for elongation and reduction in area, the values of elongation ranging from 52 to 158 per cent. The surface of the scale was lined with deep cracks but the scale itself adhered very tightly to the parent metal. The comments made concerning the microstructure of $2^1/_4$ Cr-1 Mo steel (Alloy No. 3) in paragraph (*c*) apply equally well to the microstructure of Alloy No. 4, that is, the amount of inclusions did not increase over those present in the unstressed steel, no clearly defined graphite was detected, and coalescence of the carbides originally at the grain boundaries occurred.

(*e*) 9 Cr-Mo-Si (Alloy No. 5). This alloy was also characterized by high values of elongation (from 68 to 155 per cent) and reduction in area which indicated a ductile fracture. The fracture specimens were covered with a smooth tightly adhering layer of scale which was lined with a fine network of cracks. Photomicrographs of the microstructure of this alloy indicated that the only significant change was an increasing degree of agglomeration of the carbide spheroids with time under the test conditions. In the long-time specimen, 526, a preferred orientation and stretching of the spheroids and other elements of the microstructure could be detected.

(*f*) 12 Cr (Alloy No. 6). The fractures for this alloy varied considerably. Of the twelve specimens tested there were three with brittle fractures, five with ductile fractures and four with intermediate fractures. The ductility as measured by elongation was fairly low, ranging from 20 to 71 per cent. The scale formed on fractured specimens of this alloy resembled very closely that formed on specimens of Alloy No. 5 (9 Cr-Mo-Si). There was practically no change in the microstructure of specimens of this alloy except for enough agglomeration of the carbide spheroids to make them appear to have a uniform and random distribution, whereas in the original structure the outline of the austenitic grains was indicated by the chain of spheroids.

(*g*) 25 Cr-20 Ni (Alloy No. 7) and 18 Cr-8 Ni (Alloy No. 8). Both of these alloys were characterized by very brittle fractures for both the long- and short-time intervals, resulting in low values for reduction in area and elongation. The values for elongation for the fractured specimens of 18 Cr-8 Ni ranged from 8 to 51 per cent while those for 25 Cr-20 Ni were a little higher, ranging from 32 to 75 per cent. Specimens of these alloys were covered with a thin film of scale which visually appeared to be thicker on the stressed test section than on the relatively unstressed part of the specimen. The only noticeable change in the microstructure of the 25 Cr-20 Ni specimens was the growth and coalescence of the carbides and of the peculiar black phase which has been reported by other investigators (9). The exact identity of this black phase is not yet known. The short-time specimen, 824, of 18 Cr-8 Ni which was very highly stressed (33,400 psi), resulting in a very short time to rupture (15 hr), suffered a decided change in grain size; the grain size for the ruptured specimen being much smaller than that of the original structure. Considerable carbide precipitation at the grain boundaries and inside the grains appears to have taken place. Also, the black phase which is known to appear in 25 Cr - 20 Ni steel as just described appears to be present in the microstructure of the long-time specimen, 826, of the 18 Cr - 8 Ni alloy. It is doubtful that this black phase is graphite since chromium is known to be a very effective inhibitor of graphitization and the carbon content of this steel is only 0.06 per cent.

(*h*) 5 Cr-Mo-Ti (Alloy No. 9). This alloy tended to be somewhat erratic in its behavior. Referring to Tables 2 and 3 it can be seen that of the twelve specimens tested, eight had ductile fractures, three had intermediate fractures, and one had a brittle fracture. However, it should be noticed that the one brittle fracture was the longest time specimen, 924, whose time to rupture was 3109 hr and whose value of elongation was 52 per cent. The values for elongation (45 to 104 per cent) and reduction in area tended to be somewhat lower than those for Alloy No. 4 (5 Cr-Mo-Si), an alloy differing only in the content of Si and Ti. The scale formation was identical to that of Alloy No. 4, the specimens having a fairly thick layer of scale heavily lined with deep cracks. The only appreciable change in microstructure was spheroidization of the carbides which were originally present at the grain boundaries.

Referring again to the data in Tables 2 and 3 concerning the measurements of reduction in area and elongation, it would appear that the relative ductility of the nine alloys as measured by these two methods is not the same. For example, Alloy No. 3 ($2^1/_4$ Cr-1 Mo) has relatively low ductility as measured by per cent elongation, but as measured by per cent reduction in area, as a group it has the highest ductility. The authors are of the opinion that per cent elongation is the safest indication of ductility because it is a measure of ductility over a longer period of time. Again taking Alloy No. 3 as an example, the high values of reduction in area are due simply to the fact that the specimens necked down considerably before actual rupture, which probably took place during the last few minutes of the life of the specimen.

The carbon steels tested, i.e., low-carbon (Alloy No. 1) and carbon-moly (Alloy No. 2), corroded excessively so that the reduction-in-area values could be attributed both to mechanical shrinkage due to the stress and loss of metal due to corrosion.

It can be seen in examining Tables 2 and 3 that for all of the alloys tested the values of elongation tend to be less when the brittle-type fracture occurs. A good example is Alloy No. 6 (12 Cr) in which the five specimens with the highest values of elongation all had ductile fractures while the specimens with intermediate and brittle fractures all had lower values of elongation. For the other alloy which had all types of fracture, Alloy No. 9 (5 Cr-Mo-Ti), the only specimen with a brittle fracture had the second lowest value of ductility as measured by elongation.

For all of the nine alloys taken as a whole no satisfactory correlation between time to rupture and either elongation or reduction in area could be obtained.

An examination of the microstructure of all the alloys revealed no clearly defined graphite in any case. An extensive study was made of the three alloys most likely to be susceptible to graphitization, low-carbon, carbon-moly and $2^1/_4$ Cr-1 Mo, by the research department of The Detroit Edison Company (13). They reported that while each of the steels responded "abnormal" to the McQuaid-Ehn test, no clearly defined graphite was found in any of the samples examined.

Conclusion

1 The straight-line relationship between stress and time to rupture on log-log co-ordinates, postulated by White, Clark, and Wilson, for tests in air (9) also holds for tests in steam.

2 The break in the stress-rupture line encountered in air tests by other investigators (9, 10) was not found in the present tests in steam.

3 The tests indicate that for the small specimens of the nine alloys tested the stress required for rupture in steam in 10,000 hr or longer is higher than the stress required in air at 1200 F.

4 The type and characteristics of the scale formed on the stress-rupture specimens affects the position of the stress-rupture line on log-log co-ordinates.

5 The slope of the stress-rupture line is important since an alloy whose stress-rupture line on log-log co-ordinates has a low value of slope might not appear to be desirable at short rupture times, but for long rupture times might be better than other alloys which were superior at short rupture times.

6 Ductility values, as measured by per cent elongation, are more reliable values for ductility over a long period of time than are ductility values as measured by reduction in area. Also reduction-in-area values are more affected by the amount of corrosion.

7 For the two austenitic alloys tested, the fractures were the brittle type, even for short time intervals.

8 In general, the ductility, as measured by elongation for all of the alloys tested tends to decrease as the time to rupture increases.

9 Longer times and lower stresses are favorable for the production of brittle fractures.

10 The tests confirm the conclusion based on air tests (11), that elongation for a given steel is always less when the brittle fracture occurs.

11 As in unstressed tests, the chromium content of the alloy is the critical factor in determining the amount of corrosion taking place. The corrosion decreases as the chromium content increases.

Acknowledgments

The authors wish to express their appreciation to the following for their co-operation in the preparation and the analysis of the microstructures of the fractured specimens: Dr. C. L. Clark of the Timken Roller Bearing Company, Canton, Ohio; Dr. H. K. Ihrig of the Globe Steel Tubes Company, Milwaukee, Wis.; Mr. C. H. Fellows and Mr. I. A. Rohrig of The Detroit Edison Company, Detroit, Mich.; Mr. J. P. Magos, Mr. N. A. Ziegler, Mr. J. J. Kanter, and Mr. W. L. Meinhart of Crane Company, Chicago, Ill.; and Mr. R. J. Raudebaugh, Assistant Professor of Metallurgical Engineering, Rochester University, Rochester, N. Y.

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Discussion

Arthur McCutchan.[6] The steam-corrosion work at Purdue has been followed with great interest. The stress rupture characteristics of various steels in steam should be of particular value to designers of high-temperature equipment. The choice of 1200 F as the test temperature doubtless was made on the basis of accelerating the scaling effect. Since the field of usefulness

[6] Engineer, Engineering Division, The Detroit Edison Company, Detroit, Mich. Jun. A.S.M.E.

of the low-carbon and carbon-moly steels will, in general, be below 1000 F, and the $2^1/_4$ per cent chrome-moly steel below 1100 F it is of importance to recognize that in some cases the higher temperature of test may give a false indication of the relative value of these materials for high-temperature service.

Through the co-operation of Professor Solberg, a number of the ruptured specimens of low-carbon, carbon-moly and $2^1/_4$ per cent chromium, 1 per cent molybdenum steels were examined for graphite by I. A. Rohrig of The Detroit Edison research department. Despite the fact that all were found to be "abnormal" in their response to the McQuaid-Ehn test, no graphite was reported. Since, in abnormal low-carbon and carbon-moly steels, graphite is readily produced by heating for 1000 hours or less at 1000 to 1100 F, failure to produce graphite in heating periods of 4000 to 7000 hours at 1200 F might be considered unusual. It has been hazarded that the AC_1 temperature of these materials might be lowered sufficiently in the presence of stress so that 1200 F is above the temperature at which graphite tends to precipitate.

Because of the possible precipitation of graphite, particularly at welded joints, the long-time load-carrying ability at, say, 1000 F of the low-carbon and carbon-moly steels is much less than that of the $2^1/_4$ per cent chromium, 1 per cent molybdenum steel. It would seem therefore that the stress rupture data which are given in this paper should be considered as directly applicable only at 1200 F.

On the basis of a rupture time of about 1200 to 1500 hours it is of interest to note the relatively high stress supported by the low-carbon steel in 1200 F steam. The stresses supported by the several materials were as follows:

STRESS CAUSING RUPTURE IN 1200 TO 1500 HOURS IN 1200 F STEAM

	Specimen	Stress, psi
Low carbon steel	129 M	7000
Carbon molybdenum	227	7400
$2^1/_2$ per cent chromium, 1 per cent molybdenum	326	8400

The authors' conclusions regarding the tendency to intergranular fractures are particularly noteworthy. The point that longer times and lower stresses are favorable for the production of brittle intergranular fractures of many creep-resistant steels is not always appreciated. The ductile type of fracture of the $2^1/_4$ per cent chromium — 1 per cent molybdenum steel would seem to be a count in its favor.

Authors' Closure

The authors wish to thank Mr. McCutchan for his interesting discussion. It is true that the temperature of 1200 F is severe for low-carbon, carbon-moly, and $2^1/_4$ Cr-1 moly steels, but it is felt that the data will be applicable in the future when steam temperatures in power plants and other industrial apparatus become higher.

Also it should be pointed out that the effects of specimen size and type and characteristics of the scale formed in a steam atmosphere are two variables which are yet to be thoroughly investigated.

Effect of Molding Pressure and Resin on Results of Short-Time Tests and Fatigue Tests of Compreg

By W. N. FINDLEY,[1] W. J. WORLEY,[2] and C. D. KACALIEFF[3]

Tests were performed on six samples of compressed-impregnated plywood, including static tension tests, static compression tests, static torsion tests, and rotating-beam type of fatigue tests. The results provide data on the effect of different molding pressures and different resins on the "static" strength properties, stiffness, and fatigue strength of "compreg."

THE tests reported in this paper were undertaken in order to provide information on the effect of different molding pressures and different resins on the "static" strength properties, stiffness, and fatigue strength of compressed-impregnated plywood (compreg).

The following tests were performed on six different samples (described later) at a temperature of 77 F and a relative humidity of 50 per cent: Static tension tests, static compression tests, static torsion tests, and rotating-beam-type fatigue tests. All specimens were cut with their long axis parallel to the grain of the wood used in making the compreg. Values of ultimate strength, yield strength, and modulus of elasticity were obtained from all of the static tests. The fatigue tests were carried out as far as 10,000,000 cycles.

Several investigators have reported the static properties (1–9)[4] of compreg and some other investigators have studied fatigue characteristics (10, 11) of this type of material.

Material Used for Tests

Six compreg panels were prepared especially for this series of tests by the United States Forest Products Laboratory, Madison, Wis. The panels were made from rotary-cut sap yellow-birch wood of $^1/_{16}$-in. plies. All veneers were from the same shipment and met the grain requirements of Army Air Forces Specification 15065. Prior to laminating, the $^1/_{16}$-in. birch veneers were cylinder-treated for 6 hr in the case of the water-soluble resin and 16 hr in the case of the alcohol-soluble resin at 75 psi and dried for 18 hr at 130 F. All plies were glued on one face with Bakelite XC7381 resin, using 6.7 g of dry glue per sq ft of glue line; all panels were pressed at the indicated pressure (see Table 1) under conditions such that the center of each panel was held at 305 to 310 F for 15 min. All panels were cooled to 150 F in the press before releasing the pressure. The data in Table 1 were prepared by the Forest Products Laboratory. As shown in the table, three different resins, *A*, *B*, and *C* were used, and two panels were made with each resin. One panel for each resin was molded at 1500 psi; the other panel for each resin was molded at 600 psi.

TABLE 1 FOREST PRODUCTS LABORATORY DATA

Panel number	204	205	206	207	208	209
Type of resin[a]	*A*	*A*	*B*	*B*	*C*	*C*
Resin content,[b] per cent	28.5	28.0	31.5	32.0	28.0	28.5
Number of plies	17	15	17	15	17	15
Bonding pressure, psi	1500	600	1500	600	1500	600
Specific gravity	1.36	1.22	1.36	1.24	1.37	1.24
Shear parallel to laminations,[c] psi	4840	3640	4450	4230	4340	3620
Izod,[d] ft-lb per in. notch	10.0	8.3	7.8	6.1	9.4	6.6
Water absorption,[e] per cent	3.0	7.0	1.7	5.0	2.0	6.7
Swelling,[f] per cent	14.2	13.9	9.7	9.9	11.5	12.2
Residual swelling,[g] per cent	2.90	2.97	1.87	2.22	2.28	2.00

[a] Resin *A* Bakelite BV 10678 (alcohol-soluble) by Bakelite Corporation, Bloomfield, N. J.
Resin *B* Bakelite BV 15100 (water-soluble) by Bakelite Corporation, Bloomfield, N. J.
Resin *C* Compregnite (water-soluble) by Casein Company of America, Bainbridge, N. Y.

[b] Resin content on basis of dry weight of untreated wood.

[c] Shear tests made by Forest Products Laboratory block-shear method in plane of laminations.

[d] Izod values calculated from notched-toughness values, using empirical conversion factor obtained from Izod and notched-toughness tests on matched specimens.

[e] Measurements made according to Army Air Forces specifications No. 15065 on 3 in. × 1 in. × $^3/_8$-in. specimens (1 in. in fiber direction), immersed in water for 24 hr.

[f] Specimens were cut the full thickness of the panel and $^1/_8$ in. long in the direction of the grain. The specimens were immersed in water for 48 hr at room temperature and were measured in the direction of the thickness of the original panel before and after immersion.

[g] Thickness of specimens used for swelling tests after subsequent air-drying and oven-drying, expressed as a percentage of the thickness of a matched specimen subjected to the same drying cycle but not to the water immersion. The "thickness" measured was in the direction of the thickness of the original sheet.

The test panels were made from large sheets of veneer from which six matched panels were laminated with the grain of all plies parallel. The six panels were matched with two crosswise of the grain and three lengthwise of the grain.

Additional data shown in Table 1 include the percentage resin content, the code number for the resin, number of plies used in the panel, the specific gravity, the shear strength parallel to the laminations, the impact strength (expressed as an equivalent Izod test), percentage water absorption, percentage swelling, and percentage recovery after wetting.

Preparation of Specimens

A drawing of the "static" specimens used in this investigation is shown in Fig. 1, and the fatigue specimen is shown in Fig. 2. All specimens were cut from the panel with their long axis parallel to the grain of the wood. All specimens were machined on a lathe and were finished by polishing with grade 2/0 emery paper. Some of the tension specimens were machined with a straight cylindrical portion and used for stress-strain data; other of these specimens were machined with a reduced section or neck at the center of the gage length and were used to determine the ultimate strength of the compreg in tension.

The compression specimens were turned on a lathe to the two lengths shown in Fig. 1(*c*). The 2-in. specimen was used as shown in Fig. 6, to obtain stress-strain relationships. The short specimen was used to obtain ultimate-strength values only. The torsion specimens were also made by turning on a lathe.

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[1] Assistant Professor of Theoretical and Applied Mechanics, College of Engineering, University of Illinois, Urbana, Ill. Jun. A.S.M.E.

[2] Instructor in Theoretical and Applied Mechanics, College of Engineering, University of Illinois. Jun. A.S.M.E.

[3] Research Assistant in Theoretical and Applied Mechanics, College of Engineering, University of Illinois.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

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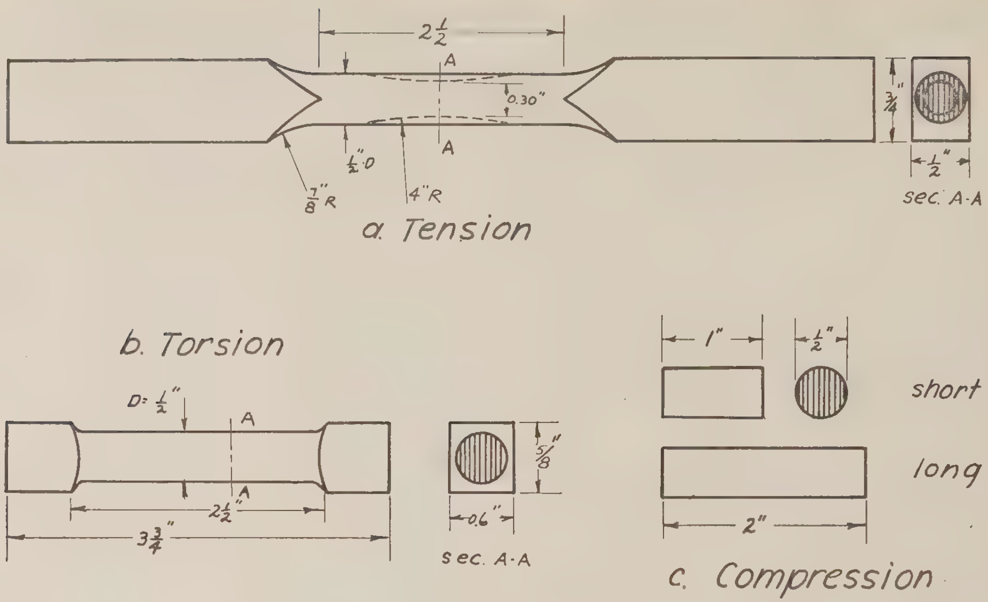

FIG. 1 STATIC SPECIMENS USED IN TEST

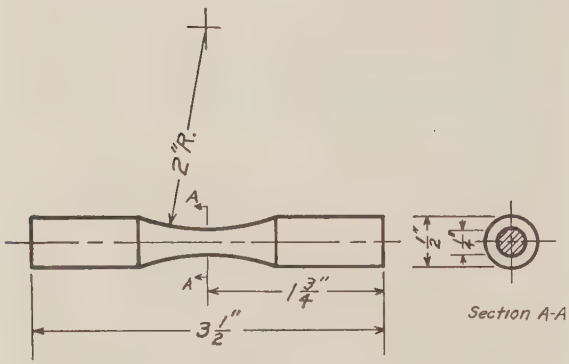

FIG. 2 ROTATING-CANTILEVER-BEAM FATIGUE SPECIMEN

The machining of all specimens was done in a laboratory maintained at constant temperature and constant relative humidity. After the specimens were completed, they were allowed to remain in this room, which was maintained at a constant temperature of 77 ± 1 deg F and constant relative humidity of 50 ± 2 per cent, for at least 2 weeks before the tests were started.

TENSION TESTS

Short-time tension tests were performed on specimens as shown in Fig. 1(*a*). These specimens were tested in tension on an Olsen 10,000-lb four-screw machine, of the beam-weighing type, equipped with a separate variable-speed drive. The specimens were held in Templin wedge grips *A*, Fig. 3, mounted in such a way as to provide an axial load on the specimen. Strain of the specimen was measured by means of a Moore-Hayes 2-in-gage-length extensometer *B*, Fig. 3. This instrument provided a multiplication such that one division on the dial indicated a strain of 0.0001 in. per in. in the specimen. In order that these specimens need not support the weight of the extensometer and to prevent damage to the instrument if the specimen should fracture while the extensometer was attached, the extensometer

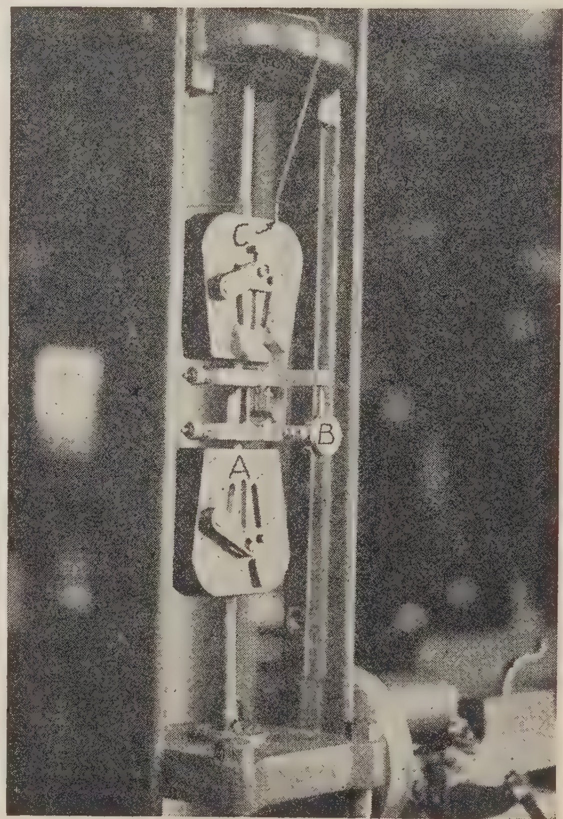

FIG. 3 TENSION APPARATUS

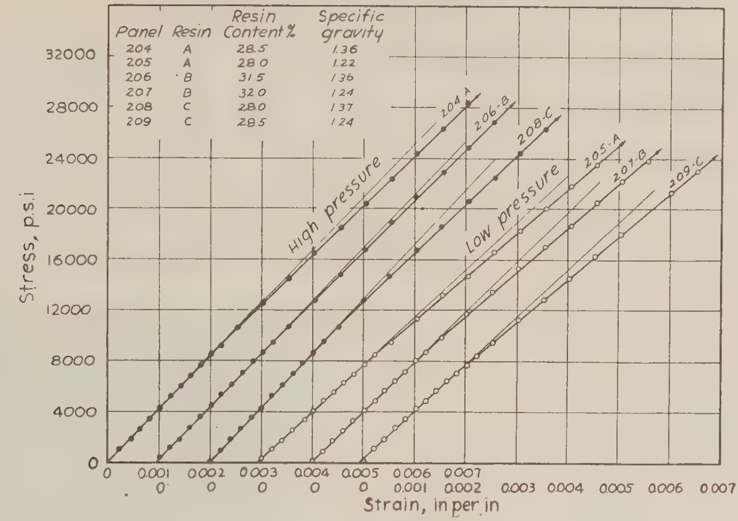

FIG. 4 TENSION TESTS OF COMPREG

was supported by means of the light coil spring C, shown in Fig. 3.

Two sets of tension tests were run; one set in which the stress-strain characteristics were determined, and the other set in which only the ultimate strength was obtained. In the former group the gage section of the specimen was straight and readings of load, deformation, and time were simultaneously obtained "on the run" throughout the test. In the latter case only the load at fracture was recorded. For this purpose the specimens were necked down by turning on a lathe to a 4 in. radius. This was accomplished by rotating the compound of the lathe. These specimens were reduced from $^1/_2$ in. diam at the end of the test section to $^3/_{10}$ in. diam at the center. This reduction was found necessary in order to cause the specimen to fail at the center of the test section rather than in the shoulder of the specimen. Straight specimens failed at the shoulder as a result of the stress concentration.

A preliminary test was made to determine the testing speed required to produce a rate of tensile strain of about 0.0016 in. per in. per min. All succeeding tests were run at or near this rate of strain. This rate was selected in order to permit correlation between the results of these tests and tests performed by the authors on other material (12). This rate of strain corresponds roughly to the rate of strain produced by testing machines operated at a head speed of 0.05 in. per min. However, it should be noted that different machines, and even different materials tested in the same machine at the same rate of crosshead motion will not in general produce the same rate of strain in the specimen. This is due to different relative stiffnesses of the machine, specimen, and auxiliary gripping apparatus.

The rate of strain was made uniform in these tests rather than the rate of increase of stress as used by some investigators, because the same rate of strain can be used without difficulty in testing materials which differ widely in modulus of elasticity, whereas if such materials are tested at the same rate of stressing, the required testing-machine speeds become excessive for materials of low modulus, and the rate of strain becomes too high to follow with ordinary equipment.

Both the tension and the compression tests were performed outside of the air-conditioned laboratory. The temperature of the testing room was about 75 F, and the relative humidity was variable. In order to minimize the effect of differences in temperature and relative humidity, the specimens were placed in an insulated box while being transferred from the conditioned storeroom to the laboratory, and the tests were conducted within 5 to 10 min after the specimen was removed from the insulated box. During the test, readings of load, deformation, and time were recorded up to a point within a few per cent of the load at which failure was expected. The extensometer was removed before failure.

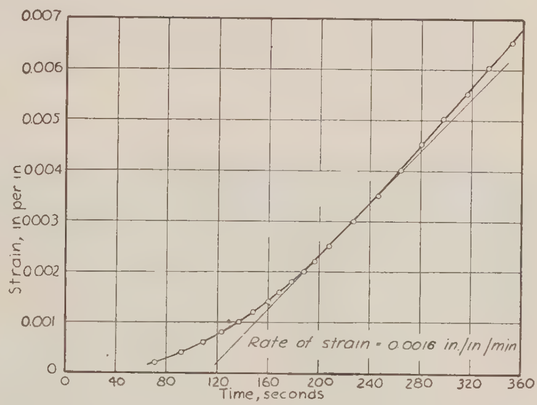

FIG. 5 STRAIN-TIME CURVE OF COMPREG IN TENSION

From these data the stress and strain were computed. Then diagrams of stress versus strain, Fig. 4, and strain versus time, Fig. 5, were plotted. The modulus of elasticity was determined in each case from the slope of the initial part of the stress-strain curve. The yield strength[5] at 0.05 and 0.2 per cent offset could not be determined owing to the fact that the straight specimens failed at the shoulder before the stress in the gage section became high enough to produce an offset of 0.05 per cent.

The rate of strain was determined from a time-strain curve (see Fig. 5), by measuring the slope of this curve in the region just below the value of strain above which the stress was no longer

[5] See "Standard Definitions of Terms Relating to Methods of Testing," A.S.T.M. Standards, part 3 (E6-36), 1942, p. 849.

TABLE 2 TENSION TESTS

Stress-strain tests (constant cross section specimens)						
Panel number	204	205	206	207	208	209
Type of resin	*A*	*A*	*B*	*B*	*C*	*C*
Molding pressure, psi	1500	600	1500	600	1500	600
No. of specimens tested	2	2	2	2	2	2
Modulus of elasticity, psi $\times 10^{-6}$	4.10–4.13	3.70–3.74	4.20–4.20	3.90–3.94	4.12–4.34	3.86–3.90
Average	4.12	3.72	4.20	3.92	4.23	3.88
Approximate rate of strain, in. per in. per min	0.0017	0.0013	0.0017	0.0015	0.0015	0.0014
Ultimate-strength tests (reduced cross section at center of specimen)						
No. of specimens tested	3	3	3	2	3	2
Ultimate strength, psi	56500–51700	44500–42800	55600–53600	44400–42600	53000–50000	49000–46800
Average	53700	43600	54600	43500	51300	47900

proportional to strain. The tangent line used is shown in Fig. 5.

Results of the tension tests are shown in Table 2. The average modulus of elasticity in tension was found to vary from 3,720,000 to 4,230,000 psi, depending on the resin and molding pressure used; and the average ultimate strength was found to vary from 43,500 to 54,600 psi, depending on the resin and molding pressure. The data in Table 2 show that both the ultimate strength and modulus of elasticity are higher for compreg molded under 1500 psi pressure than for compreg molded at 600 psi. This is true for each of the three resins investigated. The increase in ultimate strength, resulting from increasing the molding pressure from 600 to 1500 psi, was 23.2 per cent when resin *A* was used, 25.5 per cent when resin *B* was used, and 7.1 per cent when resin *C* was used. The corresponding increase in tensile modulus of elasticity was 10.7 per cent with resin *A*, 7.2 per cent with resin *B*, and 90 per cent with resin *C*.

Short-Time Compression Tests

Compression specimens were tested in the same machine as the tension specimens and under the same temperature and humidity conditions. In order to minimize the effect of possible eccentric loading the specimens were tested by using a compression tool *A*, shown in Fig. 6. A compressometer *B*, of 1 to 1 ratio having a 0.0001-in. "Last Word" dial and 1-in. gage length was used to determine the strain. As in the case of the tension tests, the instrument was supported on a light coil spring.

Two different shapes of specimen, Fig. 1, were required, one to determine stress-strain relations and the other the compressive strengths. The 2-in. specimen $\left(\frac{l}{r} = 16\right)$[6] was used with the 1 in-gage-length compressometer to determine the compressive strength of the material.

[6] $\frac{l}{r}$ is the ratio of the length of the specimen to the radius of gyration of the cross section of the specimen.

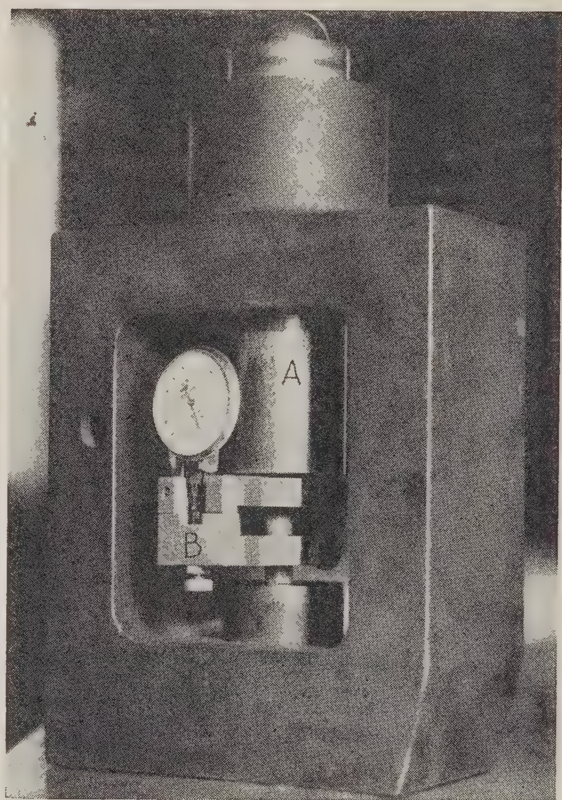

Fig. 6 Compression Apparatus

TABLE 3 COMPRESSION TESTS

Stress-strain tests (2-in. specimens)						
Panel number	204	205	206	207	208	209
Type of resin	*A*	*A*	*B*	*B*	*C*	*C*
Molding pressure, psi	1500	600	1500	600	1500	600
Number of specimens tested	2	2	3	3	3	2
Modulus of elasticity, psi $\times 10^{-6}$	4.00–4.42	3.64–3.94	4.16–4.18	3.92–4.20	4.30–4.38	3.90–3.98
Average	4.21	3.81	4.17	4.07	4.33	3.94
Yield strength at 0.2 per cent offset psi	20900 20400	18500–19300	21400–22500	20400–21600	21500–21800	19500–20200
Average	20700	19000	21900	21000	21600	19900
Yield strength at 0.05 per cent offset psi	18900–19500	16400–18100	19800–20300	19000–20200	20000–20100	18200–19000
Average	19200	17500	20000	19400	20100	18600
Approximate rate of strain, in. per in. per min	0.0014	0.0015	0.0015	0.0014	0.0015	0.0015
Compressive-strength tests (1-in. specimens)						
Number of specimens tested	3	4	4	4	4	4
Ultimate strength, psi	24100–24200	22400–22800	26600–27000	25000–25300	23800–25600	22600–23000
Average	24200	22700	26800	25200	24500	22800

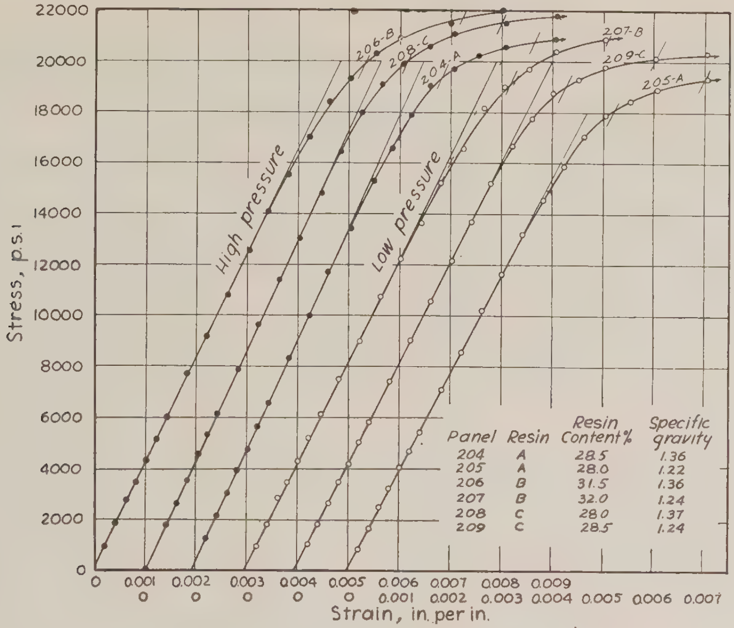

Fig. 7 Compression Tests of Compreg

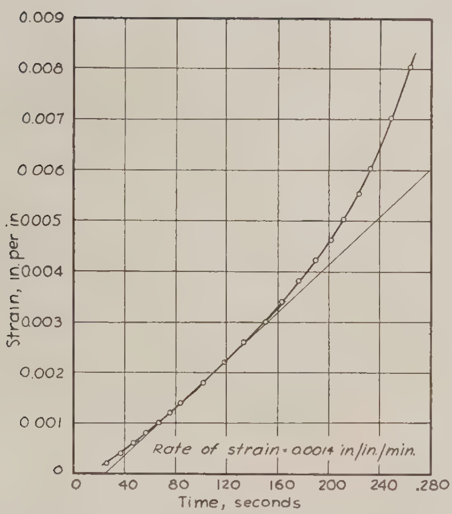

Fig. 8 Strain-Time Curve of Compreg in Compression

During the compression tests of the 2-in. specimens, readings of load, deformation, and time were recorded. From these data the stress and strain were computed, and stress-strain and strain-time curves were plotted. Fig. 7 shows the stress-strain curves for the compression tests, and Fig. 8 shows a sample strain-time curve from a compression test. The modulus of elasticity, yield strength at 0.05 and 0.2 per cent offset, and the rate of strain were determined from these curves. The rate of strain in the 1-in. specimen was determined by comparing the loading rate with the loading rate in a 2-in. specimen. The loading rate was obtained by plotting a load-versus-time curve.

Results of the compression tests are shown in Table 3. The average modulus of elasticity in compression was found to vary from 3,810,000 to 4,330,000 psi, depending on the resin and

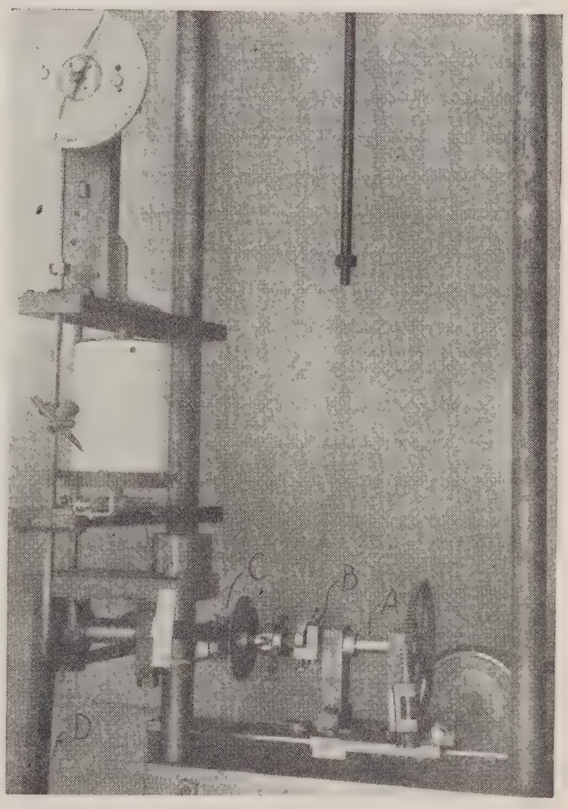

Fig. 9 Torsion Apparatus

molding pressure used, and the average ultimate strength in compression was found to vary from 22,700 to 26,800 psi, depending on the resin and molding pressure. The data in Table 3 show that the modulus of elasticity, ultimate strength, and yield strengths in compression are about 5 per cent higher for compreg molded under 1500 psi pressure than for compreg molded at 600 psi. This is approximately true for each of the resins investigated.

The increase in modulus of elasticity in compression resulting from increasing the molding pressure from 600 to 1500 psi was 10.5 per cent when resin A was used, 2.5 per cent when resin B was used, and 9.9 per cent with resin C. The corresponding increase in ultimate strength was 6.6 per cent with resin A, 6.3 per cent with resin B, and 7.5 per cent with resin C. The increase in yield strength at 0.2 per cent offset was 8.9 per cent with resin A, 4.3 per cent with resin B, and 8.5 per cent with resin C. Corresponding values for the yield strength at 0.05 per cent offset are 9.7, 3.1, and 8.1 per cent, respectively.

Short-Time Torsion Tests. The special torsion-testing machine used for these tests is shown in Fig. 9. The machine was constructed as an attachment for a low-capacity tension-testing machine. The pendulum-weighing system of the tension-testing machine was used as the torque-measuring device for the torsion machine. This was accomplished by attaching to the tension machine a twisting head, A, Fig. 9, driven by a double worm drive. A special chuck B was attached to the shaft of this twisting head and another chuck C to the axis of the pendulum D. These chucks were designed to apply a torque to the specimen with slight danger of inducing bending. This action was further assisted by mounting the specimen on centers and applying the torque as a couple by means of adjustable screws.

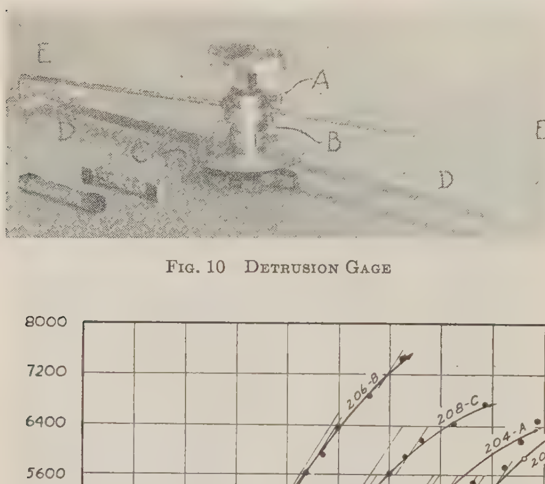

Fig. 10 Detrusion Gage

The gage used for measuring the shearing strain is shown in Fig. 10. It was designed to accommodate materials whose ultimate shearing strain was relatively small and also materials which might twist 2 or 3 revolutions in a length of 2 in. The instrument consisted of two rings A, Fig. 10, which were slipped over the specimen and fastened to it by three adjusting screws in each ring. A gage length of 2 in. was obtained by use of a removable spacer B. To one of the rings was fastened a circular scale C, for measuring large angles of twist. Two 10-in. arms D fastened to the same ring carried scales on the ends, which were used to measure small shearing strain. Adjustable pointers E, were attached to the other ring in such a way as to indicate the readings on their respective scales.

Laminated materials, such as the compreg described herein, are anisotropic, that is, their properties are not the same in all directions. In the tension and compression specimens the maximum normal stress was parallel to the grain direction throughout the test specimen. However, in a torsion specimen with axis parallel to the lamination, both the tension and shearing stresses had different directions (relative to the direction of the wood grain and the plane of laminations) at different points around the circumference of the specimen. Thus the equations of stress and strain for torsion of an isotropic member cannot be expected to indicate accurately the stress or strain in this anisotropic material. Nevertheless, the following equations developed for isotropic materials were used here to give nominal values of shearing stress and strain as a basis for comparison: The equation for shearing stress in a circular member of an isotropic material subjected to torsion is $\tau = \frac{Tc}{J}$, and the corresponding equation for shearing strain is $\gamma = \frac{c\theta}{l}$.

In the case of a laminated material, the torsion test may serve as an indication of the relative shearing strength of the bond between laminations, that is, in a wood-laminated material failure is likely to take place by shearing parallel with the grain.

Short-time torsion tests were performed on specimens, Fig. 1(b), cut from the sheet with their longitudinal axes parallel to the grain.

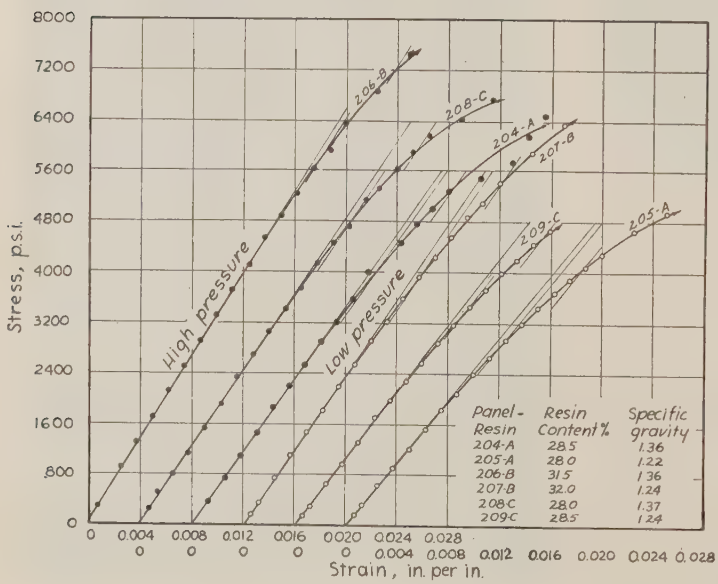

Fig. 11 Torsion Tests of Compreg

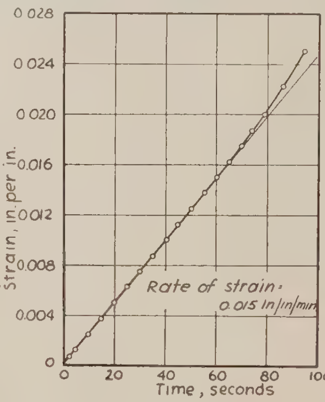

Fig. 12 Strain-Time Curve of Compreg in Torsion

TABLE 4 TORSION TESTS

Panel number	204	205	206	207	208	209
Type of resin	*A*	*A*	*B*	*B*	*C*	*C*
Molding pressure, psi	1500	600	1500	600	1500	600
No. of specimens tested	2	3	3	3	2	2
Shearing modulus of elasticity, psi $\times 10^{-6}$	0.286–0.292	0.246–0.249	0.326–0.347	0.295–0.302	0.303–0.308	0.238–0.263
Average	0.289	0.248	0.337	0.298	0.306	0.251
Shearing yield strength at 0.2 per cent offset, psi	4740–5460	4040–4420	7230–8000	5610–5820	5840–5920	4540–5150
Average	5100	4180	7720	5700	5880	4850
Shearing yield strength at 0.05 per cent offset, psi	3180–3980	2760–3340	5840–6700	3960–4400	4460–4580	3420–4290
Average	3580	2960	6340	4120	4520	3860
Modulus of rupture in torsion, psi	6720	4940–5110	9630–9960	6740–7120	8400	4670–5680
Average	6720[a]	5060	9800[b]	6980	8400[a]	5180
Approximate rate of strain, in. per in. per min	0.015	0.016	0.014	0.015	0.015	0.016

[a] One specimen.
[b] Two specimens.

During the torsion test simultaneous readings of torque, angle of twist, and time were recorded. The nominal shearing stress at the surface of the cylindrical specimen and the nominal shearing strain were computed from the relationships just given.

Curves of shearing stress versus shearing strain, Fig. 11, and time versus shearing strain, Fig. 12, were then plotted. The shearing modulus of elasticity G and rate of shearing strain were determined from these curves as in the case of the tension tests. It may be observed that the strain-time curve for torsion is practically a straight line, as compared with the curved strain-time diagrams for tension and compression. This is due largely to the greater stiffness of the machine compared to that of the specimen in the case of the torsion tests.

The results of the torsion tests are given in Table 4. The average shearing modulus of elasticity G, as determined from a torsion test, was found to vary from 248,000 to 337,000 psi, depending on the resin and molding pressure used, and the average modulus of rupture varied from 5060 to 9800 psi, depending on the resin and molding pressure. The data in Table 4 show that the shearing modulus of elasticity, modulus of rupture, and yield strengths were higher for compreg molded under 1500 psi pressure than for compreg molded at 600 psi. This was true for each of the three resins. The increase in modulus of rupture was particularly marked. The increase in shearing modulus of elasticity resulting from increasing the molding pressure from 600 to 1500 psi was 16.5 per cent with resin A, 13.1 per cent with resin B, and 21.9 per cent with resin C. The corresponding increases in modulus of rupture were 32.8 per cent for resin A, 40.4 per cent for resin B, and 62.1 per cent for resin C. The yield strengths at 0.2 per cent offset were increased 22.0, 35.4, and 21.2 per cent for resins A, B, and C, respectively; and the yield strengths at 0.05 per cent offset were increased 21.0, 53.9, and 17.1 per cent for resins A, B, and C, respectively.

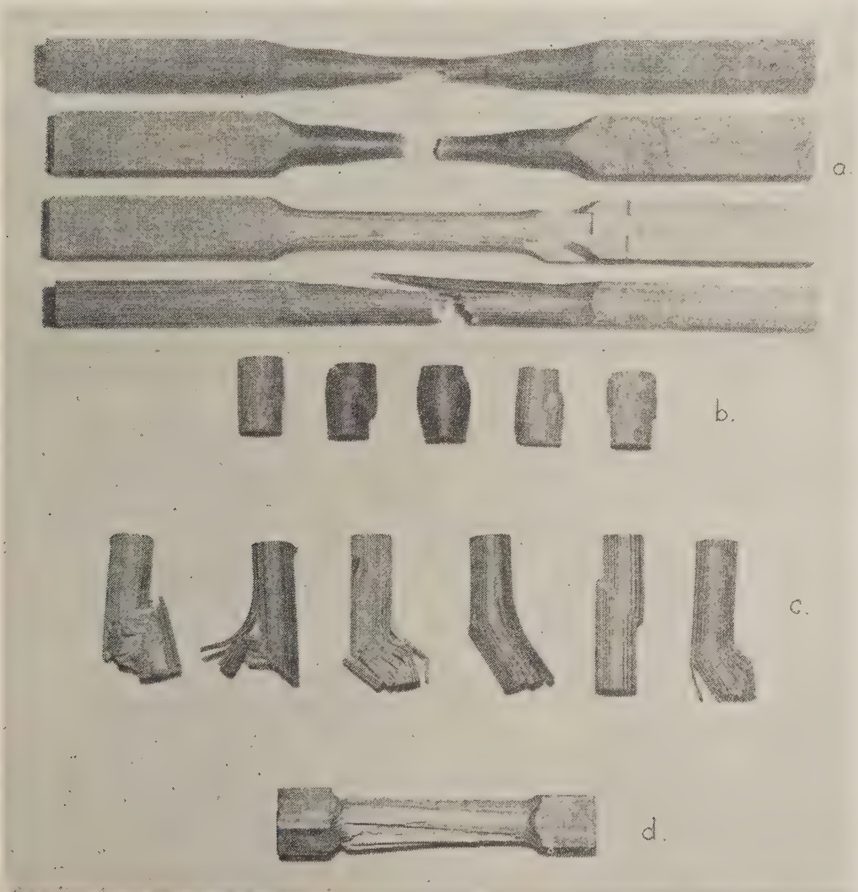

Fig. 13 Fractured Specimens
(*a*, Tension; *b*, short compression; *c*, long compression; *d*, torsion.)

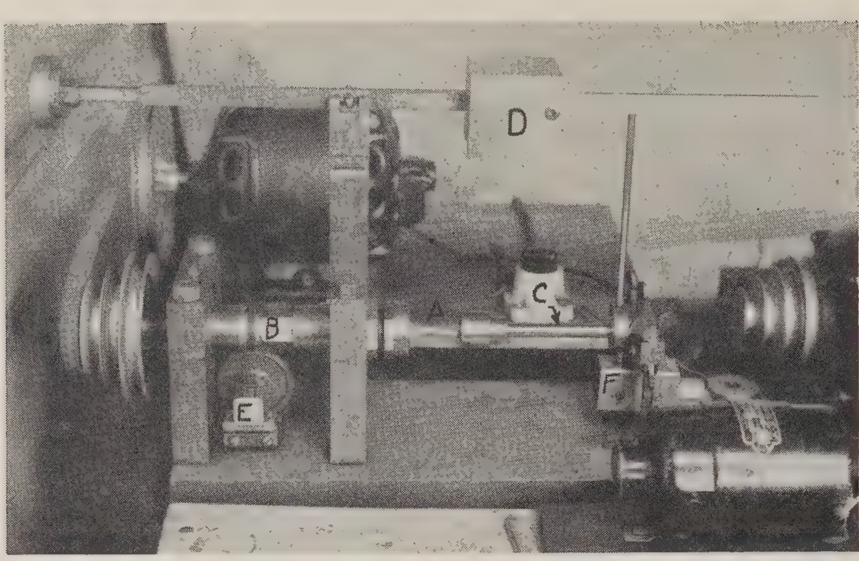

Fig. 14 Rotating-Cantilever-Beam Fatigue Machine

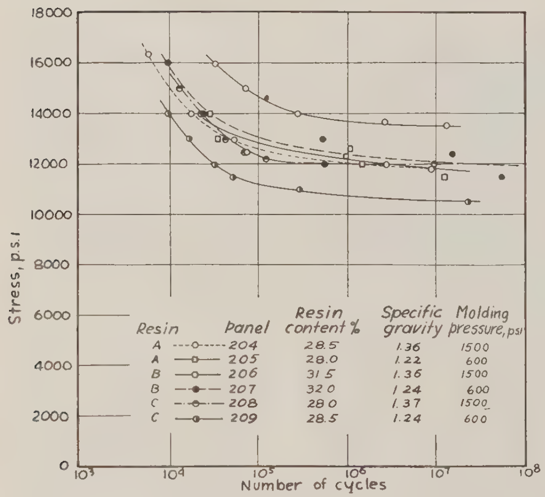

Fig. 15 Rotating-Beam Fatigue Tests of Compreg

Mode of Failure

Fractured tension, compression, and torsion specimens are shown in Fig. 13. Tension specimens are shown in Fig. 13(*a*). One of the straight tension specimens shows the type of failure which frequently occurred at the shoulder. Short compression specimens are shown in Fig. 13(*b*). These failures started as shear failures at 45 deg to the axis of loading.

The long compression specimens, Fig. 13(*c*), failed initially in much the same manner as the short specimens. It was observed that the shear failures which are visible in the illustration did not occur until after the maximum load had been passed. A slight bulging was observed, however, as the maximum load was approached.

A failed torsion specimen is shown in Fig. 13(*d*). This specimen has been severely twisted to show the type of fracture. The initial fracture in each torsion specimen was a slight longitudinal crack resulting from the shearing stress parallel to the axis of the specimen. The crack was parallel to the grain but usually did not follow the glue line between plies.

Fatigue Tests

Fatigue tests were performed on a rotating-cantilever-beam type fatigue machine shown in Fig. 14. This machine consisted of a motor-driven spindle *B* to which the specimen *A* was attached coaxially by means of a split collet. A shaft extension *C* was fastened to the other end of the specimen by means of a collet machined integral with the shaft. The entire assembly (spindle, specimen, and extension shaft) was rotated at a speed of 6200 rpm by a motor driving through a V-belt.

The specimen was bent downward by a load applied through a small ball bearing to the end of the extension shaft. This load

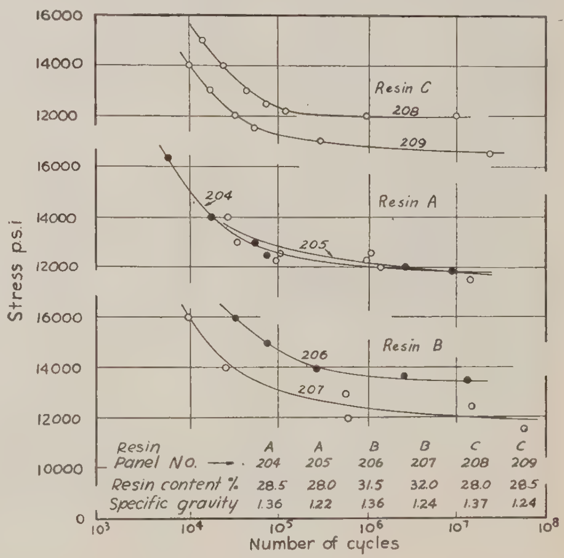

Fig. 16 Rotating-Beam Fatigue Tests of Compreg

TABLE 5 SUMMARY

Number	Item	Resin *A*		Resin *B*		Resin *C*	
1	Panel number	204	205	206	207	208	209
2	Molding pressure, psi	1500	600	1500	600	1500	600
3	Specific gravity	1.36	1.22	1.36	1.24	1.37	1.24
4	Tension modulus, psi $\times 10^{-6}$	4.12	3.72	4.20	3.92	4.23	3.88
5	Tension strength, psi	53700	43600	54600	43500	51300	47900
6	Compression modulus, psi $\times 10^{-6}$	4.21	3.81	4.17	4.07	4.33	3.94
7	Compression strength, psi	24200	22700	26800	25200	24500	22800
8	Compression yield strength at 0.2 per cent offset psi	20700	19000	21900	21000	21600	19900
9	Torsion (shearing) modulus, psi $\times 10^{-6}$	0.289	0.248	0.337	0.298	0.306	0.251
10	Modulus of rupture (torsion), psi	6720	5060	9800	6980	8400	5180
11	Shearing (torsion) yield strength at 0.2 per cent offset, psi	5100	4180	7720	5700	5880	4850
12	Shear strength parallel to laminations, psi	4840	3640	4450	4230	4340	3620
13	Impact, ft-lb per in. of notch	10.0	8.3	7.8	6.1	9.4	6.6
14	Fatigue strength at 10^7 cycles, psi	11800	11800	13400	12000	12000	10600
15	Water absorption, per cent	3.0	7.0	1.7	5.0	2.0	6.7
16	Swelling, per cent	14.2	13.9	9.7	9.9	11.5	12.2
17	Recovery, per cent	2.90	2.97	1.87	2.22	2.28	2.00

was produced by a beam-and-poise mechanism *D*. The stress σ at the minimum section of the specimen was computed from the equation $\sigma = \frac{Mc}{I}$ in which M was obtained from the load applied by the poise with suitable correction for the moment produced by the extension shaft.

In order to determine the number of cycles to cause failure a counter *E* was attached to record the number of cycles and a microswitch *F* was used to stop the machine when a crack had started in the specimen.

The data obtained from the fatigue tests of the six compreg panels is shown in Fig. 15, plotted as $\sigma-N$ diagrams. The tests were carried out to at least 10,000,000 cycles. It was observed that the fatigue-test results were nearly the same for four out of the six panels. The panel made with resin *B* and molded at the high pressure had the best fatigue characteristics, while the panel made with resin *C* and molded at low pressure had the poorest fatigue characteristics.

Fig. 16 shows the fatigue curves grouped in pairs according to the resin used. This diagram shows that the fatigue curves for resin *A* are almost identical. The different molding pressures made no difference. In the case of resins *B* and *C*, however, the ordinate (stress) of the fatigue curve was about 12 per cent higher when the panels were molded at 1500 psi than when molded at 600 psi.

The fatigue strength at 10,000,000 cycles is tabulated as item number 14 in Table 5. The term "fatigue strength"[7] refers to the maximum amplitude of an alternating stress cycle which will not cause fracture of the material for a given number of cycles of alternating stress. In other words, the fatigue strength is given by the ordinate to the $\sigma-N$ curve at the given number of cycles. In this report, the number of cycles used was 10,000,000.

The fatigue strength at 10,000,000 cycles was 11,800 psi with resin *A* for both molding pressures; with resin *B*, the fatigue strength was 12,000 psi, for the low molding pressure and 13,400 psi for the high pressure, an increase of 11.7 per cent; with resin *C* the fatigue strength was 10,600 psi for the low pressure and 12,000 psi for the high pressure, an increase of 13.2 per cent.

Comparison of Results

Average values of all properties determined for the six panels are summarized in Table 5. A study of this table discloses that panel 206, molded at 1500 psi with resin *B*, showed the best properties in 9 out of 14 items tabulated (numbered 3 through 16). The 17th item, recovery, was difficult to evaluate in this manner and will not be included in the comparison. This panel (number 206) was next to the highest in tensile modulus, compression modulus, and shear strength parallel to the laminations, but was fourth from the highest in impact strength, and of course had a higher specific gravity than the materials molded at low pressure. Panel 204 had the highest impact strength and shear parallel to the laminations; and panel 208 had the highest tension and compression moduli.

When the properties of the three panels molded at low pressure were compared, it was found that panel 207 molded with resin *B* had the best properties in 11 out of 14 items. It had a specific gravity slightly higher than panel 205, a tensile strength inferior to that of panel 209, and had the lowest impact strength of the group.

No account was taken of the specific gravity in comparing the properties of the six panels so that the low-pressure panel of resin *B* may be as good or better (depending on the property and application) than the high-pressure panel with resin *B*, for applications where weight is a prime factor.

[7] See A.S.T.M. designation D671-42T, A.S.T.M. Standards part III, 1942, p. 1251.

Acknowledgment

As previously noted, this project was carried out as a part of the work of the Engineering Experiment Station of the University of Illinois, Dean M. L. Enger, Director, in the Department of Theoretical and Applied Mechanics of which F. B. Seely is head. The authors are indebted to F. B. Seely for assistance in the preparation of this manuscript.

The authors wish to express their appreciation to the Forest Products Laboratory for preparing the samples and supplying the information quoted.

A part of tests reported herein were conducted as a senior thesis by C. D. Kacalieff.

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Discussion

A. G. H. Dietz.[8] The authors have carried out an excellent series of tests and have obtained information which extends our knowledge of the behavior of compreg.

In an attempt to correlate their results with those reported by the writer and his collaborator in a previous joint paper,[9] the results summarized in Table 5 of the paper have been reduced to a common base by dividing the values obtained in tests 4 to 14, inclusive, by the specific gravities of the individual panels. Furthermore, taking panels 206 and 207 as points of reference, because they have the highest resin contents and also, generally speaking, have the highest values, ratios of values for panels 206 and 207 to those of 204 and 208, and 205 and 209, respectively, have been drawn. The results are summarized in Table 6 of this discussion.

It is, of course, not safe to generalize from these results, but certain factors are worthy of note because they may be indicative of trends:

1 When reduced to a common base (specific gravity reduced to 1.00) most of the values become more nearly equal, but considerable differences still exist. In panel-pairs 204–205 and 206–207, the higher density material has higher tensile strength, but the reverse is true of 208–209. In every pair, however, the lower density material has the higher compressive strength. Furthermore, in every pair the lower density material has higher moduli of elasticity in both tension and compression, although the differences are generally slight.

2 Shear characteristics and impact strength, unlike the general trend in tension and compression, appear to be benefited by higher densities. With one exception, every test, e.g., torsion (shearing) modulus, torsion modulus of rupture, shearing yield strength, shear strength parallel to laminations, impact, the higher density material is superior to the lower. The one exception is in 206–207, shear strength parallel to laminations.

3 In fatigue, the results are apparently not so clear. In pairs 206–207 and 208–209, the higher density material is slightly superior, in 204–205 the reverse is decidedly indicated.

It is interesting to compare panels 206 and 207, which have the highest resin contents, with the other panels made at the same pressures. In the high-density panels, the resin ratios for 206/204 and 206/208 are 1.10 and 1.13, respectively. For the low-density panels, the ratios for 207/205 and 207/209 are 1.14 and 1.12, respectively. With these in mind, the following factors may be worthy of note:

1 Although the ratios of most of the tensile and compressive values of panels 206 and 207, as compared with the other panels, are greater than unity, none of them is as great as the ratios of the resin contents. Increasing the resin content does not appear to increase these values proportionately. As might be expected from the characteristics of wood and resin, compressive strength is most nearly proportional, tensile strength less so.

2 With the exception of shear strength parallel to laminations, all the ratios of shear values, e.g., torsion (shearing) modulus, torsion modulus of rupture, shearing yield strength, are proportionately higher than the ratios of resin content. Increasing resin content appears to aid these properties. Shear parallel to laminations is proportionately lower in the high-density panels but the reverse in low density.

3 Impact values follow previous experience; the higher the resin content, the lower the impact strength.

4 Fatigue values are inconclusive, high-density panels give results favorable to higher resin content, low-density panels the reverse.

In the paper previously referred to,[9] fatigue results were compared with static moduli of rupture (bending). In the present paper no moduli of rupture are given, but if it is assumed, as a rough approximation, that moduli of rupture lie halfway between the tensile and compressive strengths, the fatigue values are approximately 30 to 35 per cent of the moduli of rupture. This is in approximate agreement with the results previously reported.[9]

[8] Associate Professor of Structural Engineering and Materials, Department of Building Engineering and Construction, Massachusetts Institute of Technology, Cambridge, Mass.

[9] Reference 11 of authors' Bibliography.

Henry Grinsfelder.[10] The matching of panels is not entirely clear to the writer. Will the authors explain this matter more fully?

Do the values obtained with resin B represent definite proof that this is the best of the three resins tested, or is the improvement obtained by virtue of the higher resin content?

While the properties tested and the values obtained show the merit of compreg as it is prepared initially, is there any knowledge of how much of its strength is retained after outdoor exposure for one year?

[10] The Resinous Products & Chemical Company, Philadelphia, Pa.

TABLE 6 AUTHORS' TABLE 5 REDUCED TO COMMON BASE[a]

No.	Item							Panel ratios 206/204	206/208	207/205	207/209
	Panel number	204	205	206	207	208	209				
1	Resin content, per cent	28.5	28.0	31.5	32.0	28.0	28.5	1.10	1.13	1.14	1.12
2	Molding pressure, psi	1500	600	1500	600	1500	600				
3	Specific gravity	1.00	1.00	1.00	1.00	1.00	1.00	1.00	0.99	1.02	1.00
4	Tension modulus, psi 10^{-6}	3.04	3.05	3.09	3.16	3.09	3.13	1.02	0.99	1.05	1.01
5	Tension strength, psi	39500	35800	40100	35100	37500	38600	1.02	1.06	1.00	0.91
6	Compression modulus, psi 10^{-6}	3.10	3.12	3.06	3.28	3.16	3.18	0.99	0.97	1.07	1.03
7	Compression strength, psi	17800	18600	19700	20300	17900	18400	1.11	1.09	1.11	1.10
8	Compression yield strength at 0.2 per cent offset, psi	15200	15600	16100	16900	15800	16000	1.06	1.01	1.10	1.05
9	Torsion (shearing) modulus, psi 10^{-6}	0.212	0.203	0.248	0.240	0.224	0.202	1.17	1.20	1.10	1.19
10	Modulus of rupture (torsion), psi	4940	4150	7200	5630	6130	4180	1.45	1.17	1.38	1.35
11	Shearing (torsion) yield strength at 0.2 per cent offset, psi	3750	3430	5680	4600	4300	3900	1.51	1.31	1.36	1.17
12	Shear strength parallel to laminations, psi	3650	2980	3270	3410	3170	2920	0.92	1.03	1.16	1.17
13	Impact, ft-lb per in. of notch	7.4	6.8	5.7	4.9	6.9	5.3	0.78	0.83	0.74	0.92
14	Fatigue strength at 10^7 cycles, psi	8700	9700	9900	9700	8800	8600	1.14	1.12	1.02	1.03

[a] Specific gravity reduced to unity.

A. J. Stamm.[11] While the tests on compreg are obviously not extensive enough to afford a close quantitative evaluation of the effect of molding pressure and resin content on static and dynamic strength, the fact that the veneer was carefully matched in making up the samples affords a means of securing fairly accurate comparative values for the three resins and two pressures used. The data, however, should not be construed as necessarily representing average values for compreg made with these particular resins, such as would result from more extensive sampling.

The Forest Products Laboratory has, in the past, advocated the use of a water-soluble phenolic resin in preference to an alcohol resin for making compreg, largely because of the greater dimensional stability and lack of recovery. The alcohol-soluble resin used in the panels described in the paper, however, gave much less water absorption, swelling, and recovery than the alcohol-soluble resins in use in making compreg at the beginning of the war. The explanation is that this alcohol-soluble resin has been modified to make it more like the water-soluble resins with a low degree of condensation and high fiber-penetrating power, and still retain its advantage in giving higher impact strength. The advantages claimed in various Forest Products Laboratory reports for water-soluble resins, although still existing, are no longer as important.

It is felt that all three of the resins used in making the panels for which test data are given in this report are suitable for making high-quality compreg.

[11] Chief, Division of Derived Products, U. S. Forest Products Laboratory, U. S. Department of Agriculture, Madison, Wis.

Authors' Closure

The authors wish to express their thanks to the discussers for their valuable contributions to this paper. Mr. Dietz has made some very interesting comparisons, but there is one point on which the authors are in disagreement with him. He has implied that dividing the various mechanical properties by the specific gravity reduces them to a common base for all panels.

This seems misleading. The results as reported by the authors were on a common base with regard to actual strength, stiffness, etc., and as indicated in the paper the relative merits of the panels might be different for different applications when weight is considered.

Mr. Dietz has implied that the strength of structural members of equal weight made of materials of different specific gravity can be compared by dividing the appropriate mechanical property by the specific gravity and comparing the resulting quantity. This is true for only certain specific cases, such as: tension members of the same length, short compression blocks of the same length, beams of the same length and depth, and long columns of the same length and depth of section which fail by buckling in the plane of the depth dimension.

However, this is not true for many other cases. The ultimate load-carrying capacity of beams of equal weight having the same length and the same width but made of materials having different specific gravity may be compared by dividing the flexural modulus of rupture for each material by the *square* of its specific gravity and comparing the resulting quantities.

Similarly the ultimate torque-carrying capacity of circular shafts of equal weight and length may be compared by dividing the torsional modulus of rupture of each of the materials by its (specific gravity)$^{3/2}$ and comparing the resulting quantities.

Also the ultimate load-carrying capacity of long columns of equal weight and of the same length and *width* of section which fail by buckling in the plane of the depth dimension may be compared by dividing the effective elastic modulus for bending of each of the materials by the *cube* of its specific gravity.

The stiffness of structural members can be compared in a similar manner. The stiffness of bars of equal length subjected to axial loads of either tension or compression (where no buckling is involved) can be compared by dividing the appropriate elastic modulus of each material by its specific gravity; and the stiffness of circular shafts of equal length in torsion can be compared by dividing the shearing modulus of elasticity of each material by the *square* of its specific gravity.

These relationships can be derived from the equations for load or stiffness and the given restricting conditions such as the same weight for both structural members and the same length for both structural members.

Inasmuch as the panels for these tests were prepared by A. J. Stamm, who has been directly connected with the development of compreg, the authors have referred the questions raised by Mr. Grinsfelder to Mr. Stamm. His reply is as follows:

"The panels were matched as follows: Eighteen sheets of $^{1}/_{16}$ by 28 by 54 in. rotary cut yellow-birch sapwood veneer conforming to AAF Specification 15065 were stacked and cut into 8 piles of plies approximately 14 in. wide and $13^{1}/_{2}$ in. long. The stacks of plies were numbered 204 to 211; 204, 206, 208 and 210 being matched consecutively in the fiber direction, and 204 and 205, 206 and 207, 208 and 209, and 210 and 211 being matched in the tangential direction of the tree. Stacks 210 and 211 were retained by the Forest Products Laboratory for future reference. Stacks 204 and 205 were treated with resin A, 206 and 207 with resin B, and 208 and 209 with resin C. The even-numbered stacks were assembled at 1500 psi and the odd numbers at 600 psi. As matching in the fiber direction of the wood is better than side matching, the matching for the resin type variable was better than for the pressure variable.

"The data presented in this paper are insufficient to draw conclusions regarding the small variations in properties. The only differences which we believe are real on the basis of other data obtained by the Forest Products Laboratory are the water absorption, swelling, and recovery, and the compressive strength and impact strength (Izod). Water-soluble resins of the type illustrated by resin B, we believe, penetrate the cell-wall structure more completely than the alcohol-soluble resins as represented by resin A.

"This greater penetration results in somewhat better moisture stability and makes the cell wall more rigid, resulting in an increase in compressive strength but a decrease in impact strength. Tensile strength, modulus of rupture in bending, and shear strength appear to be affected little by the type of resin used. Little significance can be placed on differences in these values.

"All three of these compregs have excellent properties. From the user's standpoint there is little to choose between them. Resin A, however, is exceptional among alcohol-soluble resins in giving high dimensional stability. More advanced alcohol-soluble resins will give inferior dimensional stability properties."

Glue-Line Stresses in Laminated Wood

By A. G. H. Dietz,[1] Henry Grinsfelder,[2] and Eric Reissner[3]

With intensified use of timber beams, consisting of laminations of relatively thin boards and planks glued together, the problem of glue lines coming apart under changing conditions of moisture content in the timber has become of considerable importance. From an examination of glued timbers undergoing changes in moisture content, it appears that delamination, when it occurs, most frequently does so when the moisture content is being reduced; during such a dehydrating phase not only does shrinking take place but also checking of the wood. Results of experimentation and field trials indicate that thin laminations are better than thick ones for gluing, and wood species with low moduli are better than species with high moduli. Adhesives must be employed which have high shearing strength and high tensile strength perpendicular to the glue line if laminated beams are to be dependable in service.

Introduction

Within recent years, timber beams, consisting of laminations of relatively thin boards and plank glued together to form large curved or straight members, have found rapidly increasing use. It has been customary to employ cold-water-mixed room-temperature-setting adhesives in the fabrication of such timbers. Experience indicates that under conditions of changing moisture content in the timber, glue lines sometimes come apart.

It has been known for some time that thick wood members are more difficult to glue than thin. Plywood has been successfully bonded over a period of years with resin adhesives, but serious difficulty has often been encountered when the thickness of one or more plies has exceeded about $^1/_4$ in. It is also known that plywood made of $^1/_8$-in. veneer is more apt to delaminate during outdoor exposure than is plywood made of $^1/_{16}$-in. veneer. In edge-glued lumber difficulties have been caused by sunken joints. Troubles have arisen in gluing plywood surfaces to lumber cores. More recently, considerable interest has been aroused by the problem of laminating oak for exposure to such severe conditions as are found around ships' keels.

The difficulty has not been one of obtaining a good initial bond but rather one of preventing delamination after relatively short periods of exposure to service conditions. In the laboratory, joints which initially fail in the wood at values above 2800 psi, when tested in compression-shear blocks, are frequently delaminated by immersing in water for 7 days and drying to 8 per cent moisture content. The glues used are those usually considered to be waterproof or water-resistant.

Many theories, i.e., acids of the wood, tannins in the wood, high hydrogen-ion content of the wood in salt water, each or all weakening the adhesive bond, have been advanced to explain this phenomenon. However, no real proof has been produced supporting these theories, and from an analysis of the available facts, it would appear that certain fundamentals concerning wood and gluing need re-examination.

When wood increases in moisture content it expands. If wood is glued and then permitted to increase or to decrease in moisture content, either the wood expands and takes the adhesive with it, the adhesive restrains the wood and prevents expansion, or delamination occurs and the wood leaves the adhesive. However, when wood shrinks due to loss of moisture, wood failure in the form of splits, or "checking," may occur in place of glue failure or delamination. In laminated wood, unlike plywood, noticeable expansion and contraction take place as the moisture content of the laminations changes. Good glue joints, therefore, depend upon the ability of an adhesive to follow or to restrain the wood during the changes in dimensions caused by changes in moisture content.

Glue-Line Stresses Caused by Moisture Changes

In order to obtain an indication of the nature of the stresses set up in the glue lines of laminated timbers as the moisture content changes, two cases have been investigated mathematically. In both, the beam consists of two parallel-grain laminations of equal thickness, Fig. 1(a). They represent the

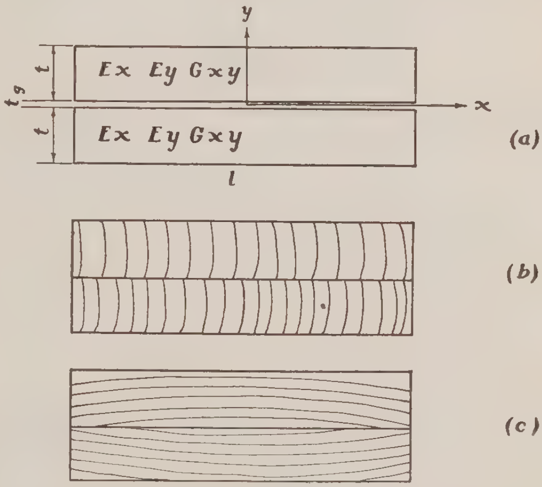

Fig. 1 Types of Laminated Beams

(a, Cross section through two-layer laminated beam of material having Young's moduli E_x and E_y in x- and y-directions; shear modulus G_{xy}, and dimensions shown. b, Two-layer beam of edge-grain wood. c, Two-layer beam of flat-grain wood.)

probable extreme cases; one in which high shear stresses are set up in the glue line without any normal stress perpendicular; the other in which high normal stresses perpendicular to the glue line occur but no shear stresses. For most beams consisting of several laminations, the usual condition would be intermediate between these extremes.

The mathematical analysis, omitted here for the sake of brevity, leads to conclusions which follow:

[1] Associate Professor of Building Engineering and Materials, Massachusetts Institute of Technology, Cambridge, Mass.

[2] Laboratory Head, Resinous Products and Chemical Company, Philadelphia, Pa.

[3] Associate Professor of Mathematics, Massachusetts Institute of Technology. Jun. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

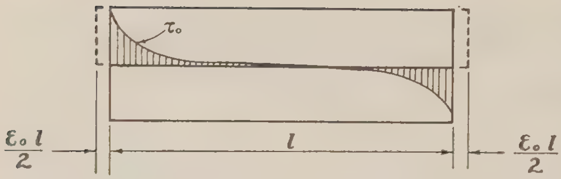

Fig. 2 Case 1—Cross Section Through Two-Layer Beam With One Lamination Tending to Expand or Contract Uniformly Because of Uniform Change in Moisture Content

(Other lamination unchanged in moisture content. Shear stresses have sharp maxima at edges of beam.)

Case 1. Two laminations of equal thickness, one of which undergoes uniform moisture change tending to increase or decrease its width whereas the other remains unchanged, as illustrated in Fig. 2.

No normal stresses perpendicular to the glue line are set up. "Rolling" shear stresses do occur as one lamination tends to expand or contract past the other. These stresses are largely concentrated at the ends of the glue line, decaying toward zero over a distance approximately equal to the thickness t of the lamination. Their maximum intensity is, approximately

$$\tau_{0\max} = 0.7\epsilon_0\sqrt{E_x G_{xy}} \qquad [1]$$

$$\epsilon_0 = \alpha(M_2 - M_1) \text{ (see Fig. 2)}$$

where α = hygroscopic coefficient of expansion
M_1 = initial moisture content
M_2 = final moisture content
E_x = Young's modulus of laminations in x-direction (parallel to glue line)
E_y = Young's modulus of laminations in y-direction (perpendicular to glue line)
G_{xy} = Shear modulus in xy-plane

Case 2. Two laminations of equal thickness, in which the moisture content changes in a parabolic manner, as shown in Fig. 3. Moisture change is symmetrical about the glue line, and no shear stresses are set up in the glue line. Normal stresses σ_0 occur perpendicular to the glue line, with maximum intensities at the ends, as shown in Fig. 3. For isotropic materials their maximum intensity is given by

$$\sigma_{0\max} = 0.45\epsilon_0 E_x \qquad [2a]$$

A modification of this formula for anisotropic materials, such as wood, is

$$\sigma_{0\max} = 0.45\epsilon_0\sqrt{E_x\sqrt{2E_y G_{xy}}} \qquad [2b]$$

where the symbols have the same significance as in Equation [1], except that ϵ_0 is now the difference in hygroscopic strain between the outermost fibers and the fibers at the glue line.

Equations [1] and [2a] have been obtained by minimum energy methods. A stress distribution was assumed of known shape in the direction across the thickness of the laminations while the variation of the stresses in the direction of the glue line was determined entirely by the energy method. A further approximation was made by saying that the state of stress was plane. It is believed that the values of the maximum stresses will be changed slightly only if plane strain is assumed instead of plane stress but this should still be confirmed by calculation. With regard to Equation [2b] it should be said that this is a plausible generalization of Equation [2a] for the isotropic material. Equation [2b] has, however, not been obtained by solving directly the problem for the anisotropic material. It is hoped that the theoretical work leading to the foregoing results may be continued so as to cover the points mentioned, as well as to obtain results which go beyond those given here.

Numerical Examples. In order to obtain numerical values of stress for the foregoing expressions, it is necessary to find values of shear modulus and Young's moduli across the grain, and of expansion across the grain caused by changes in moisture content.

In Table 1 are given some measured and some estimated values of shear modulus and Young's moduli for various species of wood. Shear modulus is in the radial-tangential plane (cross section of the wood); Young's moduli are in the radial and tangential directions of the cross section.[4] In Table 1 are also given estimated average values of the hygroscopic coefficient of expansion α_r and α_t in the radial and tangential directions. These are based largely upon values of shrinkage from fiber-saturation point to oven-dry condition.[5] The hygroscopic coefficients α are therefore the expansion per unit width of material for each per cent increase in moisture content.

Case 1. Example. Suppose, for example, that at the time of gluing the lumber is at 12 per cent moisture content (M_1) and that subsequently one lamination of the glued-up beam rises to the fiber-saturation point (M_2) or 28 per cent. The total expansion per unit width then occurs over a range of 16 per cent, and

$$\epsilon_0 = 16\alpha_r$$

for edge-grain material, (Fig. 1*b*).

$$\epsilon_0 = 16\alpha_t$$

for flat-grain material, (Fig. 1*c*).

For the various species of Table 1, the values of maximum shear-stress intensity, as calculated by Equation [1], are then approximately as given in Table 2 under the heading Case 1.

These values must be considered approximate, because the various constants, especially the moduli, are known only approximately and show wide variation. Nevertheless, it becomes clear that heavy shear-stress concentrations may be set up at the outside edges of glue lines when only the outer lamination under-

[4] "Report on Materials of Construction Used in Aircraft," by C. F. Jenkin, Aeronautical Research Committee (British), 1920; (also, from data of Forest Products Laboratory.)

[5] Wood Handbook, U. S. Department of Agriculture, Forest Products Laboratory. (Slightly revised June, 1940.)

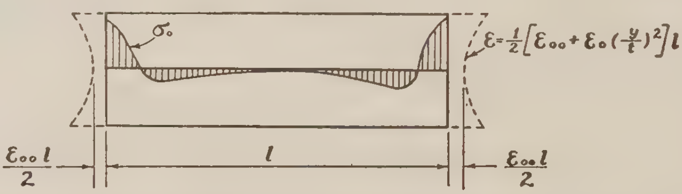

Fig. 3 Case 2—Cross Section Through Two-Layer Beam With Parabolic Change in Moisture Content

(Normal stresses perpendicular to glue line have sharp maxima at edges of beam.)

TABLE 1 HYGROSCOPIC AND ELASTIC CHARACTERISTICS OF SELECTED SPECIES

Species	Hygroscopic coefficients: Radial α_r	Tangential α_t	Elastic constants: Young's moduli: Radial E_r, psi	Tangential E_t, psi	Shear modulus G_{rt}, psi
Ash	0.00179	0.00272	238000	140000	36000
Douglas fir	0.00178	0.00278	155800	113200	7100
Mahogany	0.00170[a]	0.00270[a]	140000	70000	22000
Oak	0.00189	0.00321	175000[a]	91000[a]	35000[a]
Spruce	0.00154	0.00274	130000	70000	4600
Walnut	0.00185	0.00253	172000	92000	34000

[a] Estimated.

goes marked change in moisture content while the others remain unchanged. This "rolling" shear stress would be sufficient to rupture either the wood or the glue line for a short distance inward, and failure would be progessively inward until the ratio of thickness of lamination to width of unruptured glue line became large enough to cause a diminution of shearing-stress intensity. It is worthy of note that stresses set up by edge-grained laminations are generally less than those set up by flat-grained laminations.

This is, of course, an extreme case, and one not apt to occur very often in practice. The usual conditions are more nearly approximated by Case 2, which will be examined next.

Case 2, Example. In this example parabolic distribution is assumed for the change in moisture content across the total depth of a two-layer beam, Fig. 3. It was shown that under these conditions there would be no shear in the glue line, but that normal stresses perpendicular to the glue line would be induced as given by Equation [2*a*] for isotropic materials and by Equation [2*b*] for anisotropic materials such as wood.

Assuming, for the sake of comparison with case 1, a 16 per cent change in moisture conditions; that is

$$\epsilon_0 = 16\alpha_t$$

or

$$\epsilon_0 = 16\alpha_r$$

the normal stresses perpendicular to the glue line for the various species of Table 1, are as given in Table 2 under the heading Case 2. As was true of case 1, flat-grain laminations induce higher stresses than do edge-grain laminations, but the difference is considerably greater than was true of Case 1.

It is to be noted that the normal stresses given in Table 2 are maximum compression when the moisture content is increasing, and maximum tension when the moisture content is decreasing.

Here again we have a case in which extreme conditions are postulated. The beam is only two layers thick and both layers are undergoing extreme moisture changes, tending to cause both of them to bow in opposite directions. It must be kept in mind, also, that all the constants are subject to wide variation and that the data supporting all of them are meager. It is entirely probable that some of them change markedly with moisture content. If so, the computed stresses will be altered.

To summarize, depending upon the distribution of moisture during a change in the moisture content of a laminated beam and depending upon whether the change is an increase or a decrease, the extreme conditions are as follows:

1 No normal stress in the glue line, but shear occurring with a sharp maximum at the ends of the glue line. "Rolling" shear stresses as high as 2100 psi or more are indicated under severe conditions.

2 No shearing stresses in the glue line, but normal stresses occurring with a sharp maximum, either compression or tension, at the ends of the glue line. Theoretical stresses as high as 3240 psi are indicated.

In a laminated beam consisting of a number of pieces, the outer laminations would undergo the most marked moisture changes, and the inner laminations would change moisture content along their exposed edges. Under these conditions both shear and direct stress perpendicular to the glue lines would occur, each perhaps one half as large as indicated in Cases 1 and 2. In any event, stresses would probably be concentrated at the ends of the glue lines, tending to cause delamination there.

TABLE 2 SHEAR AND NORMAL STRESS INTENSITIES IN GLUE LINES

Species	Case 1 τ_{0max} —Shear stresses, psi— Edge-grain laminations	Flat-grain laminations	Case 2 σ_{0max} —Normal stresses, psi— Edge-grain laminations	Flat-grain laminations
Ash	1850	2160	1740	3020
Douglas fir	660	880	940	1580
Mahogany	1060	1150	910	1720
Oak	1660	2020	1360	3240
Spruce	420	550	550	1130
Walnut	1590	1590	1330	2130

Conclusions

From an examination of glued timbers undergoing changes in moisture content, it has appeared that delamination, when it occurs, most frequently does so when the moisture content is being reduced. In other words, the dehydrating phase of a moisture-change cycle is the more severe. Not only does glue-line delamination occur more frequently during shrinkage but so also does checking of the wood.

Extensive experimentation and field trials lead to the conclusion that thin laminations are better than thick for gluing, and that wood species with low moduli are better than species with high moduli. Increased strength of the wood when dry abetted by loss of plasticizing water in the glue line apparently are cumulative and a severe test for a glue.

Consideration of the factors causing glue-line delamination should enable timber laminators to produce better, more durable stock. Lower bonding temperatures or shorter high-temperature schedules may be used if the glue-line stresses are reduced, so that in many instances the extra cost of thinner laminations or a thicker beam of a weaker species may be offset by increased production and durability performance. In any event, adhesives must be employed which have both high shearing strength and high tensile strength perpendicular to the glue line both when dry and when wet if laminated beams are to be dependable in service.

Discussion

F. J. Hanrahan.[6] The authors are to be commended for their exploratory mathematical studies on intensity and distribution of glue-line stresses caused by moisture-content changes in wood; also for suggesting a method of reducing these stresses by interlaying thin veneer in each glue line.

There are some features of the paper on which additional data, explanation or discussion would be desirable.

In case 1, it is stated, "These stresses are largely concentrated at the ends of the glue line, decaying toward zero over a distance

[6] Structural Engineer, National Lumber Manufacturers Association, Washington, D. C. Mem. A.S.M.E.

approximately equal to the thickness t, of the lamination." Is this statement based upon internal stress-strain studies or is it an assumption as to distribution of intensity of stress? Does this statement mean that the shear stresses set up at the glue line are independent of the face width (beyond a width equal to twice the thickness of lamination) of the lumber laminations glued together?

The formula for maximum intensity of stress (authors' Equation [1]), appears to contain no thickness-of-lamination factor. Does this mean that the intensity of the shear stresses set up at the glue line due to moisture change is independent of the thickness of the laminations glued together?

Without further explanation it is difficult to reconcile the foregoing quoted statement with the following statement under Case 1: ". . . failure would be progressively inward until the ratio of thickness of lamination to width of unruptured glue line became large enough to cause a diminution of shearing-stress intensity."

Also it would be of interest if the authors could suggest an explanation of why the use of a thin interlayer of low-density low-moduli veneer between laminations of dense strong woods reduces or prevents delamination. Could it be that the thin relatively weak wood flows plastically, tending to permit some relative lateral movement of the denser layers? Or is the movement believed to be purely elastic?

In such a complicated investigation one should not expect complete answers from early investigations but additional light on the subject, if available, would be helpful.

Verne Ketchum[7] and O. H. Schrader.[8] Fearing the possibility that the opening paragraph of this paper may leave the impression that delamination of glue lines in laminated wood construction is a common occurrence in service, it is suggested that explanation be included that fabricators do not class such types of glue as waterproof or intended for exterior use, except where adequately protected from the elements.

It would also be desirable to explain the reasons for increased difficulty in gluing with thick laminations. At least three factors enter into this relationship: (1) The problem of accurate surfacing; (2) the effect of dimensional changes that take place after surfacing, both of which are emphasized with an increased thickness; and (3) the difficulty of obtaining intimate contact between laminations as pressure is applied.

Both white oak and Douglas fir have been successfully laminated in quantity for ship parts at the two pilot plants, established by the WPB during the war, and numerous laminated boat parts have successful records of years of service under the severest kind of service conditions. In fact, not a single failure by delamination has occurred in these boat parts, and delamination need not be a problem if adhesives now on the market are used under the conditions for which they are fitted.

The authors leave the impression that the cold-water-mixed room-temperature-setting adhesives, such as casein or urea-resin types, should resist delamination after exposure to exterior conditions. Late tests by the Forest Products Laboratory show that under such service conditions only phenol, melamine, or resorcinol-type glues, employed with careful observance of the required curing conditions, will withstand extreme service conditions. It would be well to indicate that "waterproof" and "water-resistant" are not synonymous terms, as the plywood manufacturers have spent a great deal of money in recent years in attempting to make this plain to the buying public, and to engineers and designers who may employ plywood under a wide range of service conditions.

The formulas and calculations seem valid in so far as the constants employed are valid. However, there are also several other factors which might have a bearing upon the mechanics of the problem:

1 It is known that wood possesses certain "plastic-flow" or "yield" characteristics, although at present no accurate determinations of these factors have been reported in literature. This property of wood is now the subject of extended study at the Forest Products Laboratory. The plastic-flow characteristics of wood are materially affected by changes in moisture content and may be responsible for the observed fact that delamination or wood failure along a glue line is far more evident during the drying or shrinking phase than in the swelling phase of laminated timbers.

2 The assumption is made that a glue line in laminated wood is a continuous sheet or film and is capable of transmitting stresses as a continuous member. Actual observations of glue lines, made after delamination tests or shear tests, disclose that the continuous-sheet theory is not entirely substantiated, but it is found that the glue line is frequently broken and separated into strips which often coincide with the distribution of springwood and summerwood on the face of the lamination.

In summarizing, the authors' conclusions 1 and 2 should both be modified by the consideration of the plastic-flow characteristic which will tend to reduce the computed stresses.

Again, under conclusions, the phrase, "consideration of the factors causing glue-line delamination," is so used that it infers that timber fabricators are not concerned with these factors. This is not the case, as witness the many sound efforts made to improve commercial gluing techniques, the interest in specifications that is being manifested, and the research on this subject that has been carried out in recent years. The fabricator is ready and willing to adopt any practice that will improve the quality without undue increase in cost. The use of low-strength interlayers of thin veneer would seem to increase cost and decrease shear and bending strength.

It would be desirable to distinguish between "delamination," and "wood failure at the glue line," as the former may result from improper laminating practices or improper use of adhesives, while the latter would indicate that it results from severe exposure and may be corrected by providing surface protection of paint, boxing or other type water-resisting coverings.

It is important to bear in mind that laminating is a new industry which is laboring under severe handicaps in the development of machinery and methods adaptable to volume production. It is desirable to make every effort to encourage good practices, provided they are not prohibitive from the cost standpoint, as it is certain that the industry will strongly oppose methods which increase the cost differentials that are now encountered in a competitive market.

T. D. Perry.[9] This paper reveals a problem in the use of laminated timber that is fundamental, i.e., the internal stresses that may occur within the timber and independent of any superimposed load. However, it is quite unlikely that two adjacent layers will at any time have a net differential of as much as 16 per cent moisture content, since exterior moisture diffuses inward at a diminishing ratio, and even the imperfect barrier of a glue line will not retard such diffusion to any substantial extent. Do the authors agree with this statement?

Another point to consider is the width of the glue joint across the grain of the lumber. It is conceivable that unit stresses would not be troublesome over a 1-in. width, but they might be very serious over a 12-in. width. It is probably a fair assumption that the percentage of shrinking or swelling over a 16 per cent

[7] Chief Engineer, Timber Structures, Inc., Portland, Ore.

[8] Chief Wood Technologist, Timber Structures, Inc.

[9] Development Engineer, Resinous Products & Chemical Co., Inc., Philadelphia, Pa. Mem. A.S.M.E.

moisture-content range might be as much as 2 per cent, which would not be appreciable for pieces 2 or 3 in. wide, but on a 12-in. width it would amount to $^1/_4$ in. It is doubtful whether any adhesive would maintain a strong joint between two pieces of 4/4-in. lumber, one of which was endeavoring to increase its width by $^1/_4$ in., and was restrained only by the strength of the glue line. If the widths were greater or the percentage higher, the magnitude of this stress would be much more serious. Have the authors any comment to make on this?

It has been a rough and ready rule among plywood manufacturers that test pieces, made with a suitable resin adhesive, of an all-veneer construction with $^1/_{16}$-in. veneer layers, can be put through prolonged wet-dry cycles without delamination. When $^1/_8$-in. layers are used, the case becomes marginal and delamination may occur. If, however, $^3/_{16}$-in. or $^1/_4$-in. layers are glued together, delamination is reasonably certain to take place. This has been explained by the theory that the total expansive power of a $^1/_8$-in. layer is about double that of a $^1/_{16}$-in. layer, and in other ranges varies approximately as the thickness. My observation, without scientific analysis, seems to confirm this theory. Therefore, would the thickness of the adjacent layers invalidate or modify any of the assumptions or conclusions made in the paper?

G. M. Rapp.[10] The paper is stimulative and timely; because, within the writer's knowledge, it is the first time that methods of the mathematical physicist and of the structural engineer have been applied to the long-observed phenomena of delaminating surfaces; and furthermore, because the application of laminating and modern gluing techniques, which have been so notably advanced during the war, gives promise of even greater acceleration and much wider use in the immediate future. With the "rediscovery" of wood as one of our foremost structural materials, it is necessary that we advance our knowledge of its behavior under varying conditions of moisture and temperature, quantitatively, rather than to treat these, as heretofore, as anomalies to be allowed for but not accounted for. The authors are to be congratulated for pointing out a specific problem relating to a very important limitation of wood, i.e., its dimensional instability under moisture change in glued assemblies.

The writer has long been interested in the intrinsic causes underlying the delamination of adhered surfaces. Although the authors have confined themselves to wood, the question really is a far more general one to which mathematical analyses can bring much in the way of a clearer understanding of the adverse actions involved and the remedies possible. Not only wood but most organic materials are susceptible to dimensional change with moisture content, with the inevitable result that stresses are produced in heterogeneous assemblies and even in the homogeneous materials themselves when they are subjected to moist environments. It is the writer's opinion that somewhat the same action as postulated by the authors in a simplified example of a laminate, obtains also in paint coatings on wood, ceramic glazes on clay tile, plaster on walls, in fact, in any combination where materials, which are unlike in their moisture-expansion properties, are adhered to each other. He agrees wholeheartedly with the conclusion that the cause of many observed delaminations is induced stresses. For example, the "popping" of an expanding plaster on a wall, the lifting or peeling of paint, the crazing of the glaze on tiles and of portland-cement-mortar overlays. An extreme illustration, and one intriguing to the analytically minded, is the crackling of glass produced by certain glues; a phenomenon put to practical use by the glass technologists.

Furthermore, like the authors, the writer has attempted to evaluate some of these problems as the stress analyst would investigate a bridge or other structure. After all, with a quantitative knowledge of the moisture change and other elastic properties of the materials involved, the only difficulties in obtaining a solution are those of defining correctly the equilibria conditions, establishing reasonable assumptions for simplification, and solving the mathematical equations. Analogous problems exist and have been solved in such cases, for example, as determining the stress performance of bimetallic elements (from unequal thermal-expansion coefficients) and in the calculating of the shear stresses induced in fillet-welded joints in steel members or, again, in theorizing as to the distribution of bond stresses along a concrete reinforcement bar. In all these cases, as in the authors' case, the general pattern of the stress distribution along the joint follows the exponential (e^x) law, which follows naturally from the fact that the changes in stress along the bond line are proportional in some way to the stress itself. Thus it would be expected, as the authors show, that maximum stresses would occur at the ends of the glue line and dissipate themselves rapidly as the neutral point is approached.

It is impossible to comment authoritatively upon the authors' formulas since their mathematical derivations have been omitted from the paper. It is regrettable that this was done inasmuch as the reasonableness and true significance of the conclusions depend so completely upon the mathematical processes. However, the final equations given appear to be rational though apparently simplified by the insertion of certain limits which, although they serve the purpose of a solution for the specific problem, do not permit of generalization. Obviously, the authors have made assumptions in their analysis which do not appear in the paper. Presumably one of these was that there is no slip in the glue line itself. Such an assumption would complicate the mathematics but might be a valuable addition to the theory in view of the greater attention being given to thermoplastic adhesives and to the use of "cushioning" interlays.

The writer objects to the terminology "hygroscopic coefficient of expansion." He would suggest the expression "coefficient of moisture expansion," which applies regardless of whether absorption or adsorption processes are involved, and whether the environment is air with a high water-vapor content or merely liquid water.

In his own attempts to solve similar problems mathematically, the writer has usually ended up with expressions which include the thickness of the laminate. The necessary summation of the distortions from the tensile, compressive, and shear stresses induced in both cases illustrated is dependent upon this thickness factor. The authors have contented themselves with stating that observation has shown adverse results with thicker veneers.

Also, one's attention is drawn to the statement under Case 1, "no normal stresses perpendicular to the glue line are set up." In considering infinitesmal elements directly beneath and above the glue line and the combination of the shear and direct stresses acting upon these elements, the conclusion is inevitable that there must exist a resultant or principal stress which is not parallel to the glue line. These principal stresses, which also exist in beams, have therefore components normal to the glue line, which predicate the presence of tensile or compressive forces, dependent upon the relative magnitude and direction of the shear and direct stresses. Tensile stresses, if they truly exist, may be significant in accounting for delamination. In fact some theorists go so far as to hypothesize tension as a prerequisite for every material failure.

The authors properly point out the greater susceptibility of flat-grain as compared with edge-grain veneers to delamination and checking. This is accounted for by the relatively greater dimensional change tangentially in most woods. The writer

[10] Assistant Director, John B. Pierce Foundation, New Haven, Conn.

has just seen a somewhat dramatic corroboration of this in certain tests in his laboratory involving the moisture changes in edge-grain and flat-grain laminated oak. It is a fact of practical significance worthy of note by fabricators.

The authors are to be commended for the realistic approach which they take in Case 2, wherein they assume a parabolic distribution of moisture content throughout the depth of their sample beam. Because all wood structures absorb and desorb moisture from the outside inward, this has special significance. It might well be pointed out, however, that the induced stresses will be dependent upon the rate at which such moisture gradients are generated within the wood or, in other words, on the time element. This factor, which is well known to everyone experienced in the kiln-drying of wood, offers a possible means of control against delamination in laminating practice, i.e., controlled curing.

In closing, the writer wishes to appeal to the authors to submit their mathematical analyses either in an appendix to this paper or in a future presentation. It is the only foundation upon which the true significance of their results can be appraised.

C. B. Norris[11] and A. C. Knauss.[12] It is difficult to comment on this paper without the inclusion of the mathematical analysis which leads to the results obtained. The mathematical analysis is exceedingly difficult and has not previously been accomplished. Probably some simplifying approximations have been made and the results therefore may be misleading.

It has been found that wood flows plastically at stresses well below the published proportional-limit stresses. This plastic flow will materially modify results obtained by use of the mathematical theory of elasticity. It does not seem that sufficient is known about the plastic behavior of wood to make the solution of the problem possible.

We believe that some of the assumptions and conclusions drawn are not generally applicable to nor borne out in experience with laminated-timber construction, although they may be applicable in certain specific instances.

C. D. Dosker.[13] The authors' suggestions are in line with the writer's observations of the behavior of laminated timbers when exposed to variable moisture conditions. It is true that delamination tends to take place along the exterior edges of the glue lines. It is true, also, that the first evidence of delamination frequently occurs with the first substantial change in moisture content, but does not continue to increase in depth with subsequent moisture changes. The authors' analysis suggests an interesting and seemingly logical explanation of this phenomenon.

One question which occurs to the writer concerns the distribution of stress assumed in Case 2, as illustrated in Fig. 3. No "rolling shear" stress is assumed and this assumption is undoubtedly correct if it is assumed further that there has been no change of moisture content in the fibers immediately adjacent to the glue line. If a moisture change develops adjacent to the glue line, however, and if the glue exerts a restraining influence, it would seem that some shear stress would be present on both edges of the glue line. In the absence of a moisture change adjacent to the glue line, all stress would be normal to the glue line as indicated by the authors.

It is not apparent, however, why this stress should be zero at the center part of the glue line, as is indicated in Fig. 3. On the contrary, it would seem that in the case illustrated, the maximum tension stress should occur at the center point. This, however, would not in any way tend to invalidate the conclusions derived by the authors.

[11] Forest Products Laboratory, U. S. Department of Agriculture, Madison, Wis. Mem. A.S.M.E.

[12] Forest Products Laboratory, U. S. Department of Agriculture, Madison, Wis.

[13] President, Gamble Brothers, Louisville, Ky.

Authors' Closure

The comments made in the foregoing discussions serve the very useful purpose of emphasizing certain aspects of the investigation and of indicating again its limits.

Only two cases, capable of a reasonably close mathematical solution, were selected. These two cases, furthermore, represent two extreme conditions, one in which only shear stresses are induced at the glue line, and one in which only stresses normal to the glue line are induced. Both of these lead to high stress values. In an actual instance, the true condition would probably be intermediate between these extremes.

The discussion re-emphasizes that a 16 per cent change in moisture content is an extreme case. This case was used in computing the values given in the tables, and would represent material changing from fiber-saturation point (green wood) to air-dry. Such a condition would seldom occur.

Basic to all this discussion, of course, is the uncertainty respecting the constants employed. These are known only approximately, are subject to wide variation, and presumably vary considerably with changes in moisture content. Furthermore, as re-emphasized several times in the discussion, wood is probably subject to plastic flow and therefore the constants are subject to considerable modification.

These factors not withstanding, the principal point brought out by the analysis is that both shear and tension perpendicular to the glue line may be induced by changes in moisture content, the stresses may be high, and the maximum intensity is likely to occur at the ends of the glue lines. It follows, of course, that the same stresses exist in the wood directly adjacent to the glue, and the high-quality glues presently employed in the laminating industry have shown themselves capable of withstanding any stresses which wood itself will withstand. Fabricators of laminated timber, who understand and appreciate these various factors, select their adhesives for the conditions under which they are to be employed, and also understand the necessity of taking into account the moisture content of the wood at the time of fabrication as well as in service. The analysis presented in this paper to a large extent substantiates analytically what has been found in a general way to be true by experience, as pointed out in several of the discussions.

As to the question of the nature of the mathematical analysis employed we should like to quote the following paragraph of the paper which had been inserted after the discussors had seen the paper but before the authors had seen the discussions.

"Equations [1] and [2*a*] have been obtained by minimum energy methods. A stress distribution was assumed of known shape in the direction across the thickness of the laminations while the variation of the stresses in the direction of the glue line was determined entirely by the energy method. A further approximation was made by saying that the state of stress was plane. It is believed that the values of the maximum stresses will be changed slightly only if plane strain is assumed instead of plane stress but this should still be confirmed by calculation. With regard to Equation [2*b*] it should be said that this is a plausible generalization of Equation [2*a*] for the isotropic material. Equation [2*b*] has however not been obtained by solving directly the problem for the anisotropic material. It is hoped that the theoretical work leading to the above results may be continued so as to cover the points mentioned as well as to obtain results which go beyond those given here."

The authors are fully conscious of the fact that they have only made a beginning and that much more remains to be done before

the problem of this paper can be considered to be fully solved. They are confident, however, that within the limitations stated the mathematical results will be confirmed by future work.

Early experiments with the use of thin interlayers had led to the belief that stress relief might be obtained thereby. However, continued exposure to alternate wet and dry cycles proved this belief to be in error. It did appear that the interlayers relieved to an appreciable extent the shear stresses, but did not relieve the normal stresses induced during the drying cycle. Statements regarding the use of thin veneer interlayers were omitted at the oral presentation, but unfortunately were in the earlier copies offered to the discussors. These statements are also absent from the final paper.

Considering the question presented regarding the classification of glues into waterproof and water resistant grades, the paper attempts to indicate that glues may be completely waterproof but not able to withstand the normal or shearing stresses imposed during changing moisture content of a laminated beam. Proper selection of the glue must necessarily be dependent on the wood specie as well as the beam size, lamina size, and service conditions to be expected. It would appear that a glue for spruce need not be anywhere near as strong as a glue for oak. Conditions for use of the glue also must enter into consideration, so that the possibility of using a lower temperature of cure, or a less expensive glue with the less stress inducing joints may be worthy of consideration by the timber fabricators.

Tests of Oil-Film Journal Bearings for Railroad Cars

By S. J. NEEDS,[1] PHILADELPHIA, PA.

This paper presents results of a special series of tests made with fitted and broached railway-car journal bearings as part of the program of the Journal Bearing Development Committee of the Association of American Railroads. Three recently developed bearings, each containing less bronze than the old standard A.A.R. bearing, were tested. Comparative tests were run with bath, pad, and waste-pack lubrication. Quantitative effects of fitted and broached bearing surfaces and method of lubrication on operating temperature, frictional power loss, and the all-important minimum film thickness are given. An attempt has been made to separate the friction of the bearing from that of the pad or waste-pack lubricator. Total frictional power losses of the box are given in terms of drawbar pull.

Introduction

In December, 1941, the Association of American Railroads organized a journal-bearing development committee[2] for the purpose of studying and redesigning the railway-car journal bearing. The main object of this committee was conservation of nonferrous metals. According to recent figures (1)[3] there are about 250,000 tons of nonferrous metals in the journal bearings of railroad rolling stock in this country. Any possible reduction in weight of these bearings, consistent with safety, would mean important tonnages of copper, lead, and tin available for other war purposes. Since there are some 2,000,000 cars in service, and the rate of bearing renewal is about two bearings per car per year, any possible saving would soon become apparent.

The first steps taken by the A.A.R. Journal Bearing Development Committee were to redesign the standard journal bearing and to invite bearing manufacturers and other interested parties having ideas on how strategic metals could be saved, to submit bearings for test. These tests were conducted in the laboratory of the Railway Service and Supply Corporation in Indianapolis under the direction of a full-time resident committee. Bearings successfully passing the laboratory tests were further tested in actual service.

Two obvious methods of reducing the amount of strategic metal in a bearing are to reduce its dimensions, or to replace the metal by some material more plentiful. Since there appears to be no satisfactory substitute for babbitt, reduction in quantity is the only open course. It has been found possible, however, to lighten the bronze back considerably, also to replace a large percentage of the bearing bronze with ferrous metals.

The old standard $5^1/_2 \times 10$ bearing weighed about $25^1/_2$ lb. The bronze back weighed $20^1/_2$ lb, and the $^1/_4$ in. thickness of babbitt lining averaged about 5 lb. In redesigning this bearing the committee reduced the weight of the back by about $2^1/_2$ lb and took the same weight from the lining. Thus 12 per cent of the bronze and 50 per cent of the babbitt were saved. These changes in design not only save 5 lb of metal but have resulted in mechanical improvements by eliminating several sources of trouble in the old bearing. One feature of this improved bearing is the depressed back, patented by E. S. Pearce (2), which applies the load near the ends of the bearing instead of at random, thus permitting the bearing to bend axially and conform to the flexure of the loaded journal. The back is now machined, which increases contact area and thus improves heat transfer to the wedge. Another improvement is broaching or boring the babbitt bearing surface to a uniform diameter which provides a smooth machined surface for the journal instead of the rough cast surface permissible in the old design. The improved bearing, known as the A.A.R. Emergency, D23-5/29/42, is shown in Fig. 1.

A design submitted by the author's company removes another $3^1/_2$ lb from the bronze back of the $5^1/_2 \times 10$ bearing by using ribs instead of solid construction. In this design the load is applied to the back on four properly located bosses. Due to the greater flexibility of the back and the predetermined points of load application, the bearing can flex circumferentially as well as axially. This results in increased bearing area and greater film thickness, which reduces unit load and rate of wear. Thus the $20^1/_2$-lb back of the old A.A.R. bearing has been lightened some 6 lb, or about 30 per cent, with improved bearing characteristics and performance. This bearing is shown in Fig. 2.

A bearing which goes considerably further in conservation of bronze has been submitted by the Railway Service and Supply Corporation. This design, called the "V" bearing (3), abandons the one-piece idea and employs a relatively thin bronze shell fitted into a malleable-iron back. In the $5^1/_2 \times 10$ size this composite bearing requires only 5.7 lb of bronze, as compared with 14.4 lb in the flexible-back bearing, 17.7 lb in the A.A.R. improved bearing, and 20.5 lb in the old A.A.R. bearing. The malleable-iron back weighs somewhat less than $9^1/_2$ lb. This bearing is shown in Fig. 3.

Many other bearings were submitted to the committee for test. The three mentioned are typical examples of practical substitute bearings. They performed well in service and were selected for a special series of tests herein described, to determine their operating characteristics. A full report of the committee's work is beyond the scope of this paper.

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[1] Service Manager, Kingsbury Machine Works, Inc. Mem. A.S.M.E.

[2] Membership of this committee includes Chairman W. I. Cantley, Mechanical Engineer, Association of American Railroads; C. B. Bryant, Assistant to Vice-President, Research and Tests, Southern Railway System; J. W. Hergenhan, Assistant Engineer, New York Central System; J. R. Jackson, Engineer of Tests, M. P. Lines; L. B. Jones, Engineer of Tests, Pennsylvania Railroad; J. Mattise, General Air Brake Inspector, C.&N.W. Railway; S. J. Needs, Kingsbury Machine Works, Inc., A.S.M.E. Representative; V. C. Barth, C.&N.W. Railway, R. V. Brinkworth, New York Central System, and J. M. Wingert, Pennsylvania Railroad, members of resident committee.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed jointly by the Research Committee on Lubrication and the Railroad Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

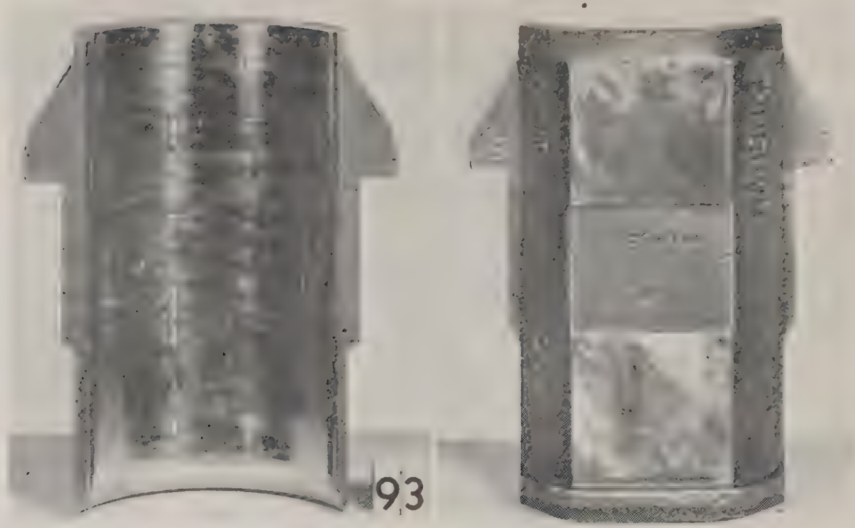

Fig. 1 A.A.R. Emergency Bearing 93 (D23-5/29/42)
(Fitted to test journal under load.)

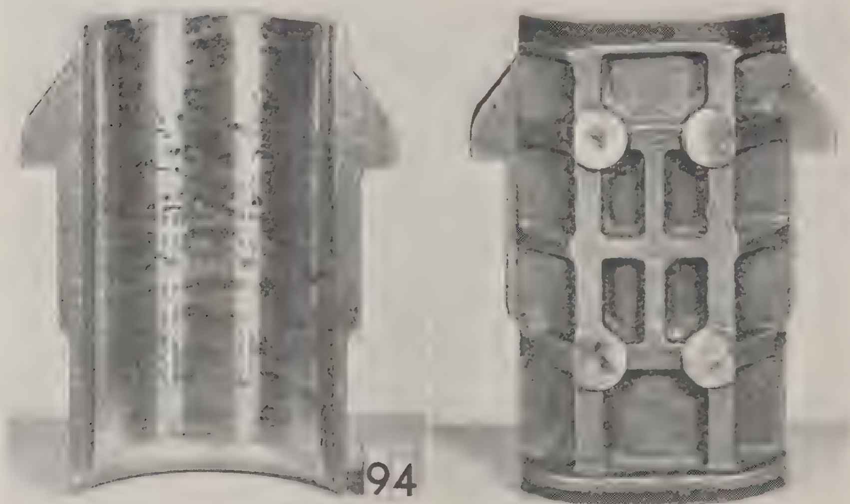

Fig. 2 Kingsbury Flexible-Back Bearing 94
(Fitted to test journal under load.)

Few Improvements in History of Railway-Car Bearings

Despite the importance of the railway-car bearing and the great number replaced annually, little effort has been made to improve its performance. The basic design of the bearing in use today dates back to the early days of the American railroads. Lack of progress has been due to several factors, perhaps most important of which is the fact that the bearing behaves very well in service without requiring too much attention. Considering the vast number of bearings in operation, actual failures are few and far between. The only machined surface required on the prewar bearing was the bore to which the babbitt lining was applied. In many cases the bearing surface and back were used as cast. For example, the lining of the $5^1/_2 \times 10$ bearing was rough-cast to $5^9/_{16}$ in. diam. Corresponding journal diameter could vary from $5^1/_2$ in. when new to 5 in. before the axle was condemned. The $5^9/_{16}$-in-diam rough-cast babbitt bearing surface was applied to the journal with permissible diametral running clearance from $^1/_{16}$ in. to $^9/_{16}$ in. After establishing load-carrying area by wear, the bearing would run for an average of 4 years under loads often exceeding 1000 psi of actual bearing area, lubricated by oily waste usually contaminated by appreciable quantities of dirt, grit, and water. The journal was finished by roller-burnishing, a process hardly conducive to a geometrically perfect surface. In the general case part of the contamination in the lubricant embedded itself in the babbitt and aided by sidesway, the bearing became an effective lap which improved the journal

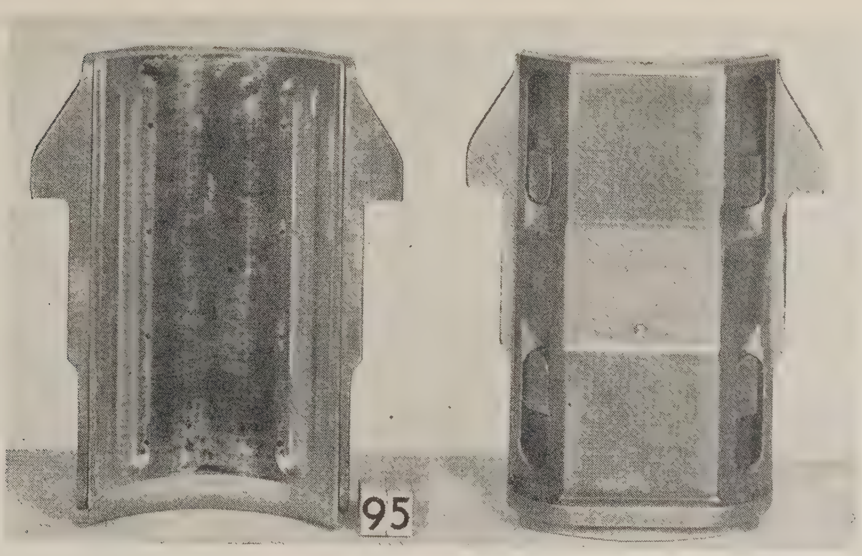

FIG. 3 RAILWAY SERVICE & SUPPLY CORPORATION "V" BEARING 95
(Fitted to test journal under load.)

surface. As bearing area increased with wear, film thickness and load capacity increased and the bearing became safer. After the initial wearing-in period the rate of wear dropped to 0.001 to 0.002 in. per 1000 miles. At the end of its average life of 4 years the bearing was removed for one of many reasons but seldom because of crown wear, since only three in every hundred bearings removed are worn out in the crown.

Considering the geometrical perfection of oil film designed into bearings for modern high-speed machinery, the behavior of the crudely constructed railway journal bearing is most impressive. Occasionally there might be an epidemic of hotboxes but year in and out the general performance has been satisfactory. Hence aside from economic considerations, there appeared to be no urgent necessity for changes.

Regardless of its record, however, the railway-car journal bearing is susceptible of improvement. One reason for improvement is the fact that in 1943 there were about 135,000 "set-outs" due to bearing trouble in freight cars, with resultant loss of time, service, and revenue. It is obvious that the frictional drag of the waste-pack consumes power which could be used in hauling useful load. It is also clear that a better fit between journal and bearing would result in less load per unit of bearing area, increased oil-film thickness, and greater load capacity.

Minimum film thickness is the criterion of bearing safety. In operation a properly fitted bath-lubricated bearing is completely separated from its journal by an oil film of appreciable thickness, except at the moments of starting and stopping. Thus wear is reduced and bearing safety increased. These considerations are not new. Fitted journal bearings are common on European railway cars and replacement of waste pack by bath lubrication has been made successfully abroad (4). The problem has also received some attention here.

To evaluate the possibilities in this direction the committee undertook comparative tests of fitted and broached bearings lubricated by bath, pad, and waste pack. The object of this paper is to describe these particular tests and to draw some general conclusions therefrom.

THE TESTING MACHINE

The machine used for the journal-bearing tests is shown in Fig. 4 and is fully described elsewhere (5). Essential elements of this machine consists of a horizontal axle supported by two roller bearings with a $5^1/_2 \times 10$-in. test journal overhung as in the conventional railway car. The axle is connected through a flexible coupling to a 25-hp d-c motor which draws its power from the city lines through a motor generator set. The machine is able to start under full test load and maintain constant rpm

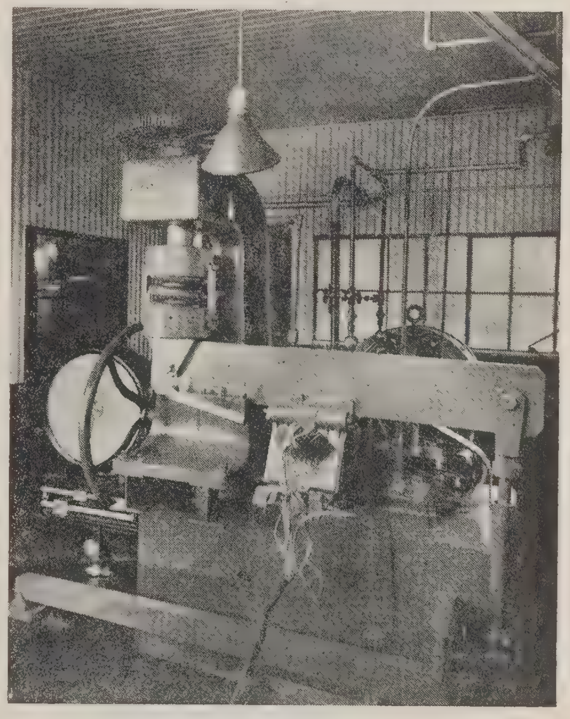

FIG. 4 JOURNAL-BEARING TESTING MACHINE

corresponding to any speed from zero to above 100 mph. An automatic recorder makes a continuous record of speed. A recording wattmeter gives continuous indication of power applied to the motor and more precise readings may be made by ammeter and voltmeter.

The bearing under test is mounted in a conventional box loaded by a beam across the top much as in regular service. Load is applied to the beam by direct-reading scales through an amplifying lever arrangement. During all tests herein described the load was held constant at 16,375 lb. A dust guard was applied to the inside end of the box to control oil leakage. All parts of the box assembly conform to A.A.R. standards.

Temperatures of journal, bearing, wedge, and box were measured by thermocouples located in a plane near the longitudinal center line of the assembly. Journal temperature was measured in a central hole in the journal drilled from the outer end. Bearing temperature was measured in a similar hole drilled in the center of the bronze back parallel with the bearing surface. Wedge temperatures were measured on each side, and box temperatures were measured at several points around the inner side.

Diameter of the test journal was 5.4385 in. A departure from usual railroad practice was made by grinding the journal to size instead of finishing by roller-burnishing. The journal was not reground for these particular tests. An entirely satisfactory surface was obtained with no measurable decrease in diameter by light honing with a fine hard stone.

Bearings, Lubrication, and Oil

Bearings included in this study are shown in Figs. 1, 2, 3, and 5. Bearing 93 is the A.A.R. emergency design D23-5/29/42, manufactured by the National Bearing Metals Corporation. Bearing 94 is the flexible-back design manufactured by the Pennsylvania Railroad from a drawing furnished by the author's company. Bearing 95 is the "V" design of the Railway Service and Supply Corporation, and Bearing 92 is the A.A.R. emergency design manufactured by Magnus Metals Corporation and known as the "Magnus Special."

Analyses of the backs show the metals are closely alike. Average contents are 73.6 copper, 19.8 lead, 5.9 tin, and 0.5 zinc. To eliminate any possible difference in results due to the linings, bearings 93, 94, and 95 were babbitted at the St. Louis plant of the National Bearing Metals Corporation at the same time with metal from the same pot. This babbitt contained 87.1 lead, 8.1 antimony, 4.7 tin, 0.1 arsenic, and a trace of copper. Bearing 92 was lined, by Magnus Metals Corporation, with a metal composed almost entirely of lead. Its contents were 97.6 lead, 1.7 tin, 0.4 calcium, and 0.2 magnesium. The alkali earth was added to harden the lead.

The four bearings were bored or broached to the journal diameter and fitted by hand-scraping under full test load, thus eliminating effects of journal and bearing flexure due to load. Bearings 92, 93, and 94 were each $5^3/_8$ in. long in the direction of motion. This is equivalent to an angular length of 113.2 deg. Actual bearing area was 45.7 sq in. and since the total load in all tests was 16,375 lb, the three bearings carried a unit load of 358 psi of actual bearing area. Bearing 95 was compound-broached, and its fitted length was $2^5/_8$ in. between the two longitudinal grooves. This is equivalent to an angular length of 55.4 deg. Actual bearing area was 22.3 sq in., and the unit load was 735 psi. All bearing surfaces were $8^1/_2$ in. wide at right angles to the direction of motion.

When fitting the bearing surfaces it was noted that the alkali-earth-hardened metal was appreciably harder than the babbitt. This is also shown in the illustrations of the fitted bearing surfaces. Fig. 5 shows the hardened lead lining spotted much as if the metal were tin-base babbitt. Figs. 1, 2, and 3, however, show the spotted areas extending over practically the entire surfaces. This intimate contact was brought about by wear and plastic flow of the high spots under load. The scraper removed most of the metal, and short circumferential back and forth movements of the dry journal under load produced sufficient wear and flow to fit the bearing to the journal. Scratches in the journal, however slight, were imprinted on the bearing surfaces. In fact this metal was so soft that gentle rubbing with clean cheesecloth would produce scratches. Its Brinell hardness was 17.8, as compared with 23.8 for the alkali-earth-hardened lead. Brinell hardness of tin-base babbitt (85–10–5), is about 23.9.

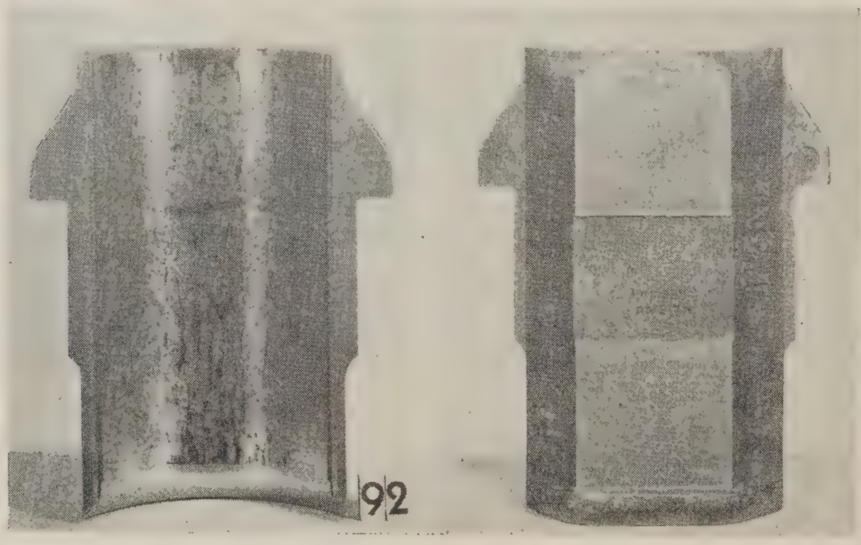

Fig. 5 A.A.R. Emergency Bearing 92; "Magnus Special"
Lined with alkali-earth-hardened lead.)

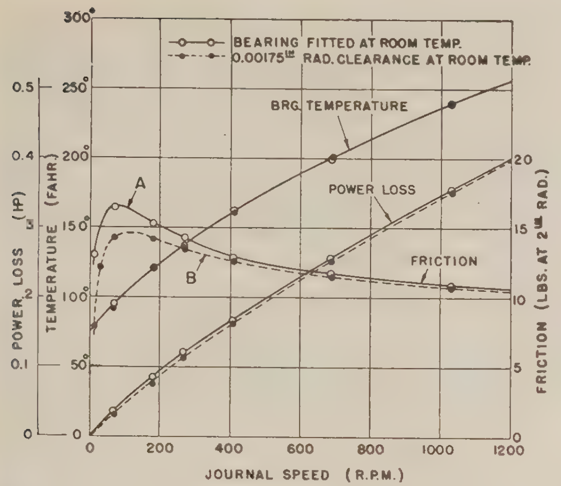

FIG. 6 EFFECT OF BEARING CLEARANCE ON FRICTION AND POWER LOSS

(120-deg centrally loaded bearing; 4 in. diam × 4 in. wide. Load 16,760 lb = 1200 psi of projected bearing area. Atreco XB oil, viscosity 14.4(10)$^{-6}$ lb sec per sq in. at 100 F.)

When bronze bearings are fitted to a steel shaft at a given temperature they remain fitted only at that temperature. Bearings 92, 93, 94, and 95 were fitted at about 75 F. Since operating temperatures were well above room, the bearings automatically provided their own running clearance by differential expansion of bronze bearing and steel journal. The importance of this clearance is shown in Fig. 6 which refers to a bath-lubricated 120-deg bronze bearing 4 in. diam × 4 in. long, carrying a load of 1200 psi of projected bearing area. Curve *A* shows variation of friction with speed when the bearing is running with a shaft fitted at room temperature. Curve *B* shows friction variation when running with a shaft 0.0035 in. smaller in diameter than the shaft to which the bearing was fitted. At higher speeds the temperature rise provides running clearance for the fitted bearing, and the increased clearance with the smaller shaft is not particularly effective in reducing friction. At lower speeds, however, the temperature is not high enough to cause much difference in expansion of bearing and journal and these surfaces remain at substantially equal radii. Under such conditions the friction of the fitted bearing is appreciably greater than that of the clearance bearing. At 10 rpm, friction with the fitted bearing is about 70 per cent greater than with the clearance bearing.

In addition to fitting the bearing surfaces, attention was given to the backs and wedge to avoid possible torsion of the bearings when loaded. Some time was also spent fitting the insert of bearing 95 to its malleable-iron back.

In all tests with bath lubrication the oil level was set at $^3/_4$ in. above the bottom of the journal at the start of each run. Oil circulation, thermal expansion, and leakage altered this level during the test but the quantity of oil in the box was kept substantially constant by replacing the small amount lost with fresh oil.

The journal lubrication pad used in the tests was developed by the Railway Service and Supply Corporation. It is described elsewhere (6) and has been successfully used in railroad service. Oil level with the pad was set at $^5/_8$ in. below the bottom of the journal at the start of each test.

The waste pack used for lubrication in these tests weighed 8 lb. It consisted of $3^1/_4$ lb of oil for each pound of new oven-dried cotton waste.

Oil used in the tests was neutral reclaimed car oil. Viscosity was 348 S.S.U. at 100 F, 154 sec at 130 F, and 53.8 sec at 210 F. Specific gravity was 0.9123 at 60 F and 0.8877 at 130 F. Oil from the same barrel was used throughout the study.

TEST OF BEARING WITH ALKALI-EARTH-HARDENED LEAD LINING

Bearing 92, with the alkali-earth-hardened lead lining, was first run bath-lubricated at 40 mph, which is equivalent to 374 rpm of a 36-in-diam wheel. Bearing temperature became constant at 176 F and the test was discontinued at the end of 6 hr. Upon inspection the bearing surface was found practically unchanged. Fitted spots in the central bearing area seemed to have lost some of their polish but were still quite distinct. Several tiny holes were found in the bearing surface near the lug end from which oil seeped as the bearing cooled. If all oil is removed from the bearing surface seepage will continue even at this writing, more than a year after the bearing was tested.

An 8-hr run was made at 40 mph the following day. Final bearing temperature was 196 F. There was no change in appearance of the bearing surface but a longitudinal seam had opened about $1^3/_4$ in. from the entering edge, running parallel with the bearing axis to within about 1 in. from each end. This seam corresponded to the edge of the large babbitt groove which in this design is cast in the back. No doubt the disturbed metal around the edges of this seam had interfered with action of the oil film and accounted for the higher operating temperature.

A run was made next day at 80 mph (747 rpm). The power-input curve was very uneven, and several times the normal motor input of 3.5 kw increased momentarily to more than 7 kw. After about 3 hr of such operation the bearing temperature reached 300 F and then began to decrease. At 6 hr the temperature had dropped to 275 F, and the machine was unloaded and stopped. Upon removal the bearing area was found badly

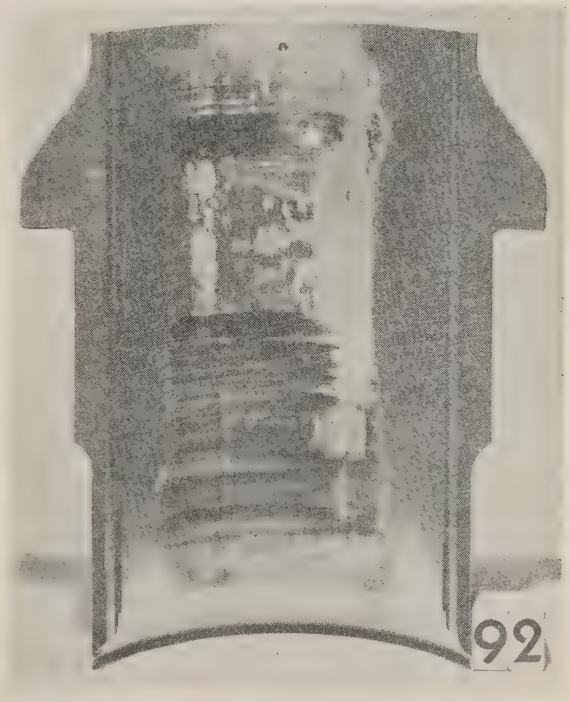

FIG. 7 CORROSION AND WIPING OF ALKALI-EARTH-HARDENED LEAD BEARING METAL

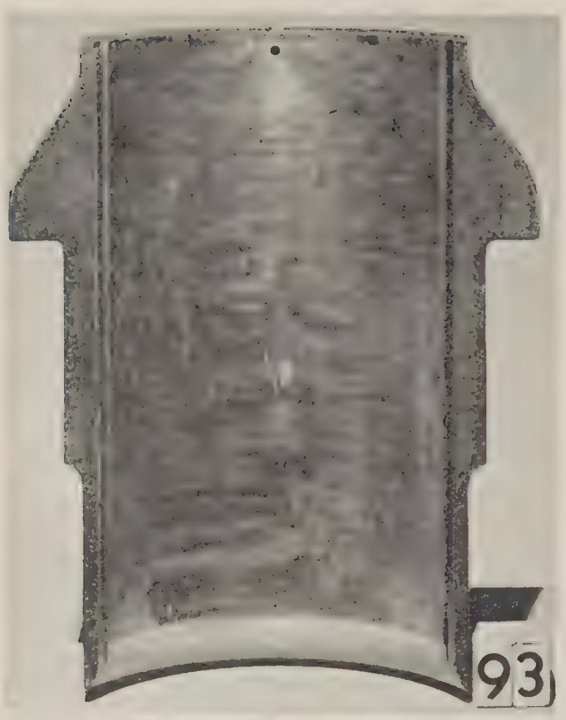

Fig. 8 Fitted A.A.R. Emergency Bearing 93 After Tests
Note that scraped pattern is still visible after bearing has "traveled" 3680 miles, indicating complete separation of bearing and journal.)

corroded and in some spots the corrosion had caused wiping, as shown in Fig. 7. Cause of corrosion remains to be determined but the only obvious difference between this run which resulted in failure due to corrosion, and the previous runs in which no corrosion occurred, was an increase in bearing temperature due to increased speed. Corrosion of high-lead alkali-earth-hardened linings had been observed in earlier tests of broached waste-packed bearings. This lining was included in the present study to discover whether or not the phenomenon would reappear when bearing and journal were more widely separated by the thicker oil film of the fitted bearing.

Tests of Babbitt-Lined Bearings

(a) *Fitted Bearings.* Procedure in all tests with babbitt-lined bearings followed the same routine. All bearings were run bath-lubricated before continuing with pad and waste pack. Fresh oil was used with each change in method of lubrication of each bearing. An 8-hr run was made with each bearing at each speed. Machine and bearing were permitted to cool overnight. Approximately one half the test period was required to warm the parts and reach stabilization temperature at which rates of heat generation and dissipation became equal. From then on there was little, if any, temperature change. No special cooling was provided for the box, hence operating temperatures were appreciably higher than would obtain in service where the box is air-cooled as the car moves along. The machine was started and stopped with the load removed to avoid possibility of surface abrasion in the absence of an oil film.

The fitted bearings are shown after the tests in Figs. 8, 9, and 10. Each bearing had "traveled" 3680 miles and except for circumferential scratches, due to insoluble particles passing

Fig. 9 Fitted Kingsbury Flexible-Back Bearing 94 After Tests
(Note that scraped pattern is still visible after bearing has "traveled" 3680 miles, indicating complete separation of bearing and journal.)

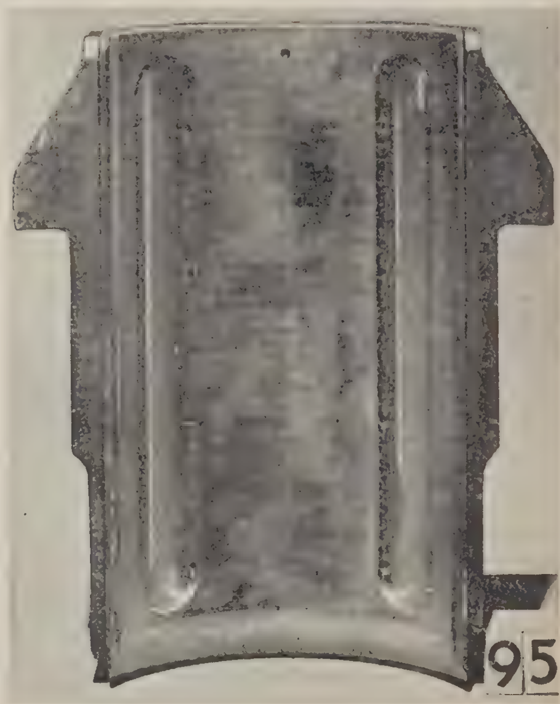

Fig. 10 Fitted Railway Service and Supply Corporation "V" Bearing 95 After Tests
(Note that scraped pattern is still visible after bearing has "traveled" 3680 miles. Scratches are somewhat more pronounced than in bearings 93 and 94 due to thinner oil film.)

TABLE 1 RAILROAD-CAR $5^1/_2 \times 10$ JOURNAL BEARINGS; BEARINGS FITTED UNDER LOAD[a]

SPEED	40 M.P.H. (373.6 R.P.M.)									80 M.P.H. (747.2 R.P.M.)									100 M.P.H. (934 R.P.M.)								
BEARING	93			94			95			93			94			95			93			94			95		
LUBRICATION	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE
JOURNAL TEMP.	168	195	225	164	200	232	176	207	240	234	261	307	230	270	322	223	279	330	270			275			255		
BEARING TEMP.	166	192	221	164	199	231	174	201	231	232	256	301	229	267	320	219	267	316	266			274			251		
BOX TEMP.	151	142	138	144	141	131	154	147	135	200	171	168	198	173	163	191	189	165	226			231			218		
ROLLER BRG. TEMP.	130	133	137	130	132	135	134	132	138	158	158	164	159	156	168	155	167	165	174			175			172		
ROOM TEMP.	90	90	93	90	90	92	94	90	91	95	88	91	92	88	95	89	95	90	96			96			98		
JOURNAL TEMP. RISE	78	105	132	74	110	140	82	117	149	139	173	216	138	182	227	134	184	240	174			179			157		
BEARING TEMP. RISE	76	102	128	74	109	139	80	111	140	137	168	210	137	179	225	130	172	226	170			178			153		
BOX TEMP. RISE	61	52	45	54	51	39	60	57	44	105	83	77	106	85	68	102	94	75	130			135			120		
MOTOR INPUT (WATTS)	1849	1886	1909	1817	1857	1966	1870	1852	1964	3090	3024	3090	3117	2990	3106	3139	2976	3231	3984			4040			3871		
MIN. OIL FILM (IN) ①	60	46	35	61	42	32	31	23	20	45	40	33	47	37	30	29	21	18	35			31			23		

① MULTIPLY BY $(10)^{-5}$

TEMP. AND TEMP. RISE IN DEG. F.

[a] Load 16,375 lb; journal diameter 5.4385 in. Stabilized running condiditions, 8-hr test runs.

TABLE 2 RAILROAD-CAR $5^1/_2 \times 10$ JOURNAL BEARINGS; BEARINGS BROACHED TO $5^{17}/_{32}$ IN. DIAMETER; DIAMETRAL CLEARANCE 0.093 IN.[a]

SPEED	40 M.P.H. (373.6 R.P.M.)									80 M.P.H. (747.2 R.P.M.)									100 M.P.H. (934 R.P.M.)								
BEARING	93-A			94-A			95-A			93-A			94-A			95-A			93-A			94-A			95-A		
LUBRICATION	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE
JOURNAL TEMP.	172	199	218	167	195	230	170	209	225	213	257	297	207	262	304	215	275	322	241			229			235		
BEARING TEMP.	170	194	213	167	194	230	168	201	216	211	250	288	207	260	302	212	263	307	240			229			232		
BOX TEMP.	148	138	138	142	126	125	144	138	131	176	158	165	172	158	155	178	176	167	198			187			197		
ROLLER BRG. TEMP.	137	140	140	136	131	135	137	134	137	159	155	163	155	160	165	162	161	165	171			165			172		
ROOM TEMP.	100	106	98	98	90	90	100	91	93	94	95	94	98	91	95	103	96	95	99			95			102		
JOURNAL TEMP. RISE	72	93	120	69	105	140	70	118	132	119	162	203	109	171	209	112	179	227	142			134			133		
BEARING TEMP. RISE	70	88	115	69	104	140	68	110	123	117	155	194	109	169	207	109	167	212	141			134			130		
BOX TEMP. RISE	48	32	40	44	35	35	44	47	38	82	63	71	74	67	60	75	80	70	99			92			95		
MOTOR INPUT (WATTS)	1811	1721	1829	1747	1752	1910	1790	1879	1836	3016	2889	2970	2902	2908	2995	2916	2970	3044	3736			3634			3569		
MIN. OIL FILM (IN) ①	16	13	11	20	16	12	14	11	10	16	13	11	20	15	12	14	10	8	15			20			14		

① MULTIPLY BY $(10)^{-5}$

TEMP. AND TEMP. RISE IN DEG. F.

[a] Load 16,375 lb; journal diameter 5.4385 in. Stabilized running conditions, 8-hr test runs.

through the oil films, there is little change in appearance of the surfaces. Original scraped patterns are clearly visible indicating complete separation of bearing and journal during the tests. Scratches on the left or trailing side of bearing 95 seem somewhat more marked than in the other bearings. This would be expected since the film thickness of bearing 95 was less because of its shorter bearing arc. Results of the tests are given in Table 1.

(*b*) *Broached Bearings.* Upon completion of the fitted-bearing tests the bearings were broached to $5^{17}/_{32}$ in. diam. To identify the broached bearings the letter A was added to the bearing number. The same shaft was used as in the previous tests, hence the diametral running clearance of the broached bearings was almost exactly $^3/_{32}$ in. The oil used was from the same source as in the previous tests. The bearings were first run bath-lubricated at 40, 80, and 100 mph, followed by pad- and waste-pack-lubricated runs at 40 and 80 mph. Test results are given in Table 2.

Figs. 11, 12, and 13 show the bearings after the tests. Due to load concentration, a worn or crown area was found in the center of each bearing. The greater the difference between journal and bearing diameters, the shorter the arc of contact and the heavier the load concentration. When difference between journal and bearing diameters is appreciable, application of load causes local deformation of the bearing surfaces, possibly accompanied by plastic flow of the relatively soft bearing metal. Thus the line contact of cylindrical bearing surfaces of different radii is increased to contact over an appreciable area. If the bearing surfaces are smooth and sufficient area is in contact, rotation of the journal introduces an oil film which separates bearing and journal and permits running without wear. If sufficient area is not in contact, wear will continue until the contact area is large enough to permit formation of a load-carrying film. Figs. 11, 12, and 13 show these effects. Under load but not running, intimate contact of the bearing surfaces occurred over an area somewhat less than is shown worn in the illustrations. Running under load, a greater area than that worn was effective in carrying the load. The circumferential scratches covering the worn areas were made by insoluble particles passing through the films. In general, the outline of the worn area is more distinct on the left than on the right. Journal rotation was from right to left and the oil film was thinner on the left or trailing side.

Other things being equal, length of crown area in direction of motion depends on the stiffness of the back and the method of loading the bearing. With an absolutely rigid back, bearing area could be developed only by elasticity, wear, or flow of bearing surfaces. In these tests the bearings showed no signs of plastic flow

Fig. 11 Broached A.A.R. Emergency Bearing 93-A After Tests

when examined after the runs. The mean crown length of bearing 93-A was found to be 21 deg by planimeter measurements of tracings of worn area. Due to controlled points of loading which permit flexure in the circumferential direction, the mean crown length of bearing 94-A is $27^1/_2$ deg. Mean crown length of bearing 95-A is $17^1/_2$ deg due, no doubt, to the rigidity of its malleable-iron back. Some area on either side of the worn crown probably is effective in supporting load-carrying film. In the absence of definite information it is assumed that this area is of the order of 20 per cent of the worn crown area. Hence, for calculation, useful length of oil film has arbitrarily been taken as 25 deg for bearing 93-A, 32 deg for bearing 94-A, and 21 deg for bearing 95-A. On this basis the same total load of 16,375 lb gives unit loadings of 1625, 1270, and 1930 psi, respectively.

The foregoing runs were repeated after rebroaching the bearings $^3/_{32}$ in. larger than a new shaft diameter of 5.406 in. In addition to the usual tests, runs were made, pad- and waste-pack-lubricated, at 100 mph. The rebroached bearings are designated by the letter B. Due to rebroaching, babbitt thickness was reduced to about $^3/_{32}$ in. and a porous spot appeared in the lining of bearing 93-B. Results were somewhat erratic with this bearing and are not included with the results of the other bearing tests given in Table 3. Comparison of Tables 2 and 3 shows operating temperatures of the A and B bearings in good agreement.

Average results for the three fitted bearings and the five broached bearings are given in Table 4. Calculated bearing losses refer to the test bearing only. Calculation of these frictional power losses is necessary since this information cannot be found directly from the testing machine. For example, consider the fitted bearing running bath-lubricated at 40 mph. Under stable temperature conditions average power input to the motor is 1845 w, from Table 4. Curves are available giving power losses at various speeds and roller-bearing temperatures when the

Fig. 12 Broached Kingsbury Flexible-Back Bearing 94-A After Tests

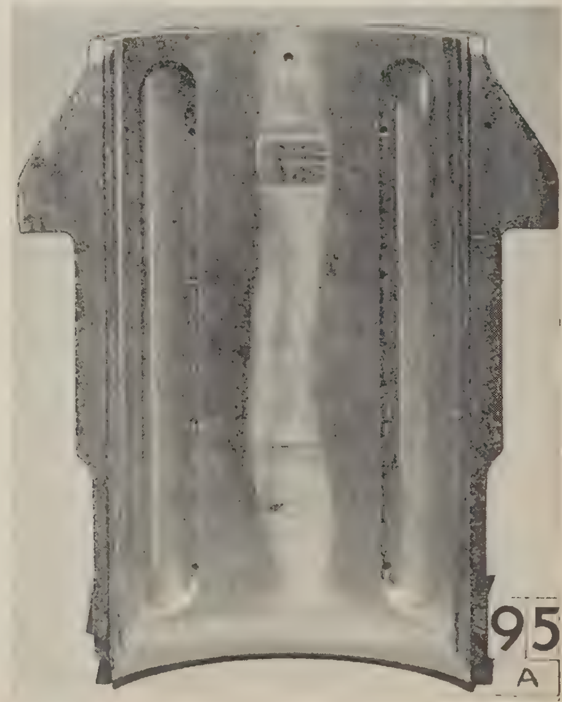

Fig. 13 Broached Railway Service & Supply Corporation "V" Bearing 95-A After Tests

TABLE 3 RAILROAD-CAR $5^{1}/_{2} \times 10$ JOURNAL BEARINGS; BEARINGS BROACHED TO 5.498 IN. DIAMETER; DIAMETRAL CLEARANCE 0.092 IN.[a]

SPEED	40 M.P.H. (373.6 R.P.M.)						80 M.P.H. (747.2 R.P.M.)						100 M.P.H. (934 R.P.M.)					
BEARING	94-B			95-B			94-B			95-B			94-B			95-B		
LUBRICATION	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE
JOURNAL TEMP.	166	199	230	169	205	227	206	264	326	208	302	305	218	289	353	223	351	396
BEARING TEMP.	165	197	228	165	195	216	206	261	323	205	285	289	218	284	349	220	331	375
BOX TEMP.	139	134	128	144	130	126	171	147	159	172	165	152	179	158	163	181	185	177
ROLLER BRG. TEMP.	134	140	138	134	135	137	164	165	165	160	158	168	170	172	170	171	173	173
ROOM TEMP.	89	98	93	93	90	88	93	94	97	93	92	89	95	97	88	94	97	88
JOURNAL TEMP. RISE	77	101	137	76	115	139	113	170	229	115	210	216	123	192	265	129	254	308
BEARING TEMP. RISE	76	99	135	72	105	128	113	167	226	112	193	200	123	187	261	126	234	287
BOX TEMP. RISE	50	36	35	51	40	38	78	53	62	79	73	63	84	61	75	87	88	89
MOTOR INPUT (WATTS)	1869	1857	1860	1821	1822	1907	2855	2806	2976	2910	3040	3008	3299	3295	3649	3426	3634	3816
MIN. OIL FILM (IN) ①	18.0	13.7	11.1	15.5	12.0	10.3	19.7	14.0	10.4	15.8	10.0	9.7	20.2	13.9	10.7	16.0	9.1	8.1

① MULTIPLY BY $(10)^{-5}$

TEMP. AND TEMP. RISE IN DEG. F.

[a] Load 16,375 lb; journal diameter 5.406 in. Stabilized running conditions, 8-hr test runs.

TABLE 4 AVERAGE OF TEST RESULTS FROM TABLES 1, 2, AND 3; $5^{1}/_{2} \times 10$ RAILROAD-CAR JOURNAL BEARINGS; FITTED AND BROACHED BEARINGS, BATH, PAD, AND WASTE-PACK LUBRICATED

	FITTED BEARINGS							BROACHED BEARINGS								
SPEED	40 M.P.H.			80 M.P.H.			100 MPH	40 M.P.H.			80 M.P.H.			100 M.P.H.		
LUBRICATION	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	BATH	PAD	WASTE	BATH	PAD	WASTE	BATH	PAD	WASTE
JOURNAL TEMP.	169	201	232	229	270	320	267	169	201	226	210	272	311	229	320	375
BEARING TEMP.	168	197	228	227	263	312	264	167	196	220	208	264	302	228	308	362
BOX TEMP.	150	143	135	196	178	165	225	143	133	130	174	161	160	188	172	170
ROLLER BRG. TEMP.	131	132	136	157	160	166	174	136	136	137	160	160	165	170	173	172
ROOM TEMP.	91	90	92	92	90	92	97	96	95	92	96	94	94	97	97	88
JOURNAL TEMP. RISE	78	111	140	137	180	228	170	73	106	134	114	178	217	132	223	287
BEARING TEMP. RISE	77	107	136	135	173	220	167	71	101	128	112	170	208	131	211	274
BOX TEMP. RISE	59	53	43	104	88	73	128	47	38	38	78	67	66	91	75	82
OIL FILM VISCOSITY ①	2.02	1.24	0.814	0.823	0.546	0.344	0.540	2.07	1.26	0.90	1.06	0.54	0.37	0.81	0.35	0.24
MOTOR INPUT (WATTS)	1845	1865	1946	3115	2997	3142	3965	1808	1806	1868	2920	2923	2998	3533	3465	3733
					*										*	
CALC. BRG. LOSS (HP)	0.242	0.189	0.173	0.441	0.332	0.274	0.548	0.190	0.149	0.125	0.387	0.277	0.230	0.485	0.324	0.266
LUBRICATOR LOSS (HP)		0.080	0.204			0.203			0.038	0.145		0.114	0.261		0.070	0.487
TOTAL BOX LOSS (HP)	0.242	0.269	0.377	0.441		0.477	0.548	0.190	0.187	0.270	0.387	0.391	0.491	0.485	0.394	0.753

① UNITS LB. SEC. PER SQ. IN.–MULTIPLY BY $(10)^{-6}$

* AVERAGE MOTOR INPUT PROBABLY LOW.

TEMP. AND TEMP. RISE IN DEG. F.

machine is unloaded and no test bearing is in the box. According to these curves, 1238 w are required to run the machine when unloaded. With no test-bearing load the roller bearings carry only the weight of the shaft, hence the 607 w difference between power input with and without test-bearing load represents friction of the test bearing and dust guard, and added friction due to the test load on roller bearings and motor. By calculation, test-bearing friction was 0.242 hp or 181 w. This leaves 426-w loss due to dust-guard friction and load on roller bearings and motor. Since the total of these unknown losses is some $2^{1}/_{4}$ times the power loss of the test bearing, it is obvious that there is no hope of arriving at test-bearing power losses directly from motor input.

It is possible, however, to estimate power losses due to pad and waste pack by assuming that at a given load, speed, and roller-bearing temperature, the machine losses will remain constant. With this assumption any difference in power input in the tests with bath, pad, and waste-pack lubrication can be attributed to variation of friction in the box. From Table 4 the power input for the fitted bearing running bath-lubricated at 40 mph is 1845 w. Calculated power loss is 0.242 hp at the bearing temperature of 168 F. The same bearing, running pad-lubricated at the same speed, requires a power input of 1865 w. In the latter case calculated power loss for the bearing alone is 0.189 hp at the operating temperature of 197 F. Difference in box loss between pad and bath lubrication is 1865 — 1845 = 20 w or 0.027 hp. Therefore total box loss is 0.242 + 0.027 = 0.269 hp, and since the power loss of the bearing alone is 0.189 hp, the difference, or 0.080 hp, represents the power loss due to the pad. In like manner power loss in the waste-packed box at 40 mph is 0.173 hp for the bearing and 0.204 hp for the waste pack at the test temperature of 228 F.

Applying this reasoning to the pad-lubricated fitted bearing at 80 mph, it is found that the total power loss for bearing and pad is 0.049 hp less than the calculated loss for the bearing alone. Apparently, observed value of motor input is low. This is not the case, however, with the broached bearings, and since power

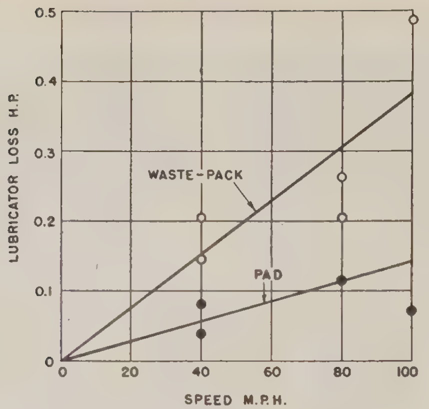

Fig. 14 Frictional Power Loss of Pad and Waste-Pack $5^1/_2 \times 10$ Bearings
(Load 16,375 lb.)

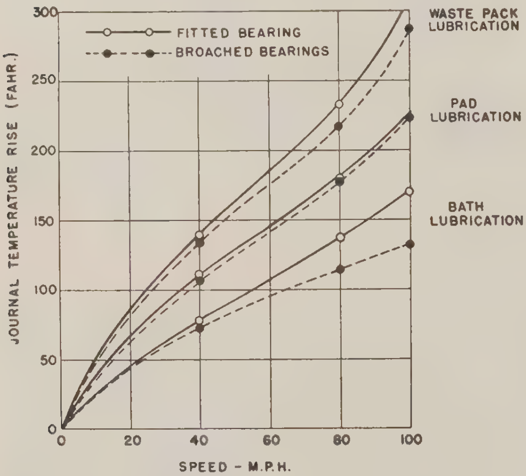

Fig. 15 Journal Temperature Rise With Speed
($5^1/_2 \times 10$ bearings; load 16,375 lb; average room temperature 93 F.)

losses of the lubricators do not depend on the bearings, the same pad loss would be expected with fitted and broached bearings. To arrive at pad and waste-pack power losses from the test results, lubricator losses in Table 4 are plotted in Fig. 14. Maximum departure of observed points from the curves represents errors in average motor input of about 2 per cent. Since box losses are a relatively small part of motor input, small errors in power measurement are greatly magnified when computing total box loss.

Discussion of Results

Variation of journal temperature rise with speed is shown in Fig. 15, where the solid lines refer to the fitted bearings and broken lines to the broached bearings. Journal temperature rise in all cases is somewhat less with broached than with fitted bearings. Since friction losses of pad and waste pack are independent of the bearing, lower operating temperatures indicate that the broached bearing operates with less friction than the fitted bearing. Calculation also shows this to be true. Against this slight advantage in friction, however, the broached bearing has the serious disadvantages of considerably less oil-film thickness and higher unit loading due to its shorter angular length. Because of its thinner film the reserve load capacity of the broached bearing is much less than that of the fitted bearing. Greater film thickness and reserve load capacity are the real advantages of the fitted bearing.

Maintenance of sufficient oil-film thickness is the deciding factor in successful bearing operation. When due to overload, overheating, or low speed, the oil film becomes thinner than the heights of the irregularities on the bearing surfaces, these high spots are no longer completely separated and metallic wear occurs. Wear may also be caused by particles in the oil too large to pass freely through the film. All wear is accompanied by temperature increase, and if heat is generated more rapidly than it can be dissipated the bearing eventually overheats and fails. If, however, an oil film thick enough to separate the bearing surfaces completely can be maintained under all conditions, metallic wear will be eliminated and the bearing cannot fail. Factors tending to thick films are high viscosity, high speed, and low unit loading.

In the absence of metallic contact, rate of heat generation depends on rate of oil-film shear. With a given load, rate of shear increases with speed, and heat thus generated reduces film viscosity. Operating temperatures will rise until the gradient between oil film and box is sufficient to dissipate the heat as rapidly as it is generated. This stable temperature may be so high that the increase in film thickness due to speed is more than offset by the decrease in thickness from reduced viscosity; and in that case, film thickness decreases with increasing speed. In these tests this decrease was more marked with the fitted than with the broached bearings, as shown by the curves of calculated oil-film thickness in Fig. 16. In service, higher speeds are accompanied by induced air currents which cool the box, thus maintaining lower operating temperatures and thicker oil films.

Effect of speed and method of lubrication on frictional power loss are shown in Fig. 17, in which average curves are plotted for fitted and broached bearings, bath, pad, and waste-pack lubricated. Power losses in the box are made up of bearing and lubricator frictions. Dust-guard friction and losses due to oil churning are comparatively small and are neglected. In each graph the lower curve represents bearing friction calculated for the conditions of the test. With bath lubrication the only source of power loss in the box is the bearing itself, hence there is but one curve in the graphs for bath lubrication. With pad and waste-pack lubrication, however, friction losses due to the lubricators are added to the bearing friction. These additional losses are represented by the shaded areas in Fig. 17, superimposed on the curves for bearing friction. The upper curves then represent bearing and lubricator friction or the total friction of the box. With each of the three types of lubrication, mean power loss in the box averages 16 per cent greater with the fitted than with the broached bearing. This agrees with the lower-operating-temperature curves for the broached bearings shown in Fig. 15.

From comparison of the shaded areas in Fig. 17, and from the curves in Fig. 14, it is interesting to note that the friction of the waste pack is 2.7 times the friction of the pad. Comparison of shaded and unshaded areas shows the waste-pack friction to be 115 per cent of the bearing friction. Similarly, friction loss due to the pad is 37 per cent of the bearing friction. From this it appears that for all practical purposes the friction of the waste pack is about the same as the friction of the bearing, and the friction of the pad is approximately $^1/_3$ that of the waste pack.

In addition to the friction of pad and waste pack, the heat-insulating properties of these lubricators add considerably to the higher operating temperatures. This is clearly shown in Fig. 18, which gives average results for the three fitted bearings at 80 mph. In each test curves were plotted from temperature readings around the longitudinal center line of the box after equi-

librium temperature had been reached. Mean temperature, found from each curve by planimeter, is listed in the tables as box temperature. Temperatures at the top of the box are little influenced by method of lubrication. Temperatures at sides and bottom, however, are greatly affected by the heat-transmission characteristics of the lubricator. When the bearing is bath-lubricated, agitation by the rotating journal facilitates transfer of heat from journal to box where it is dissipated to the atmosphere. In this case the oil is acting as a coolant as well as a lubricant.

With pad lubrication there is some oil cooling of the journal but the bath is considerably less agitated due to the presence of the pad, and heat transfer between journal and box is less efficient than with the bath. Rubbing of the pad on the journal adds to the heat generated by the bearing and this, together with poorer heat-transfer characteristics from journal to box, accounts for the higher operating temperature. Because of the higher operating temperature, however, oil viscosity has dropped to the point where the sum of bearing and pad frictions is about the same as the friction of the bath-lubricated bearing. Since total friction is the same, the higher operating temperature of the pad-lubricated bearing must be due entirely to the heat-insulating characteristics of the pad.

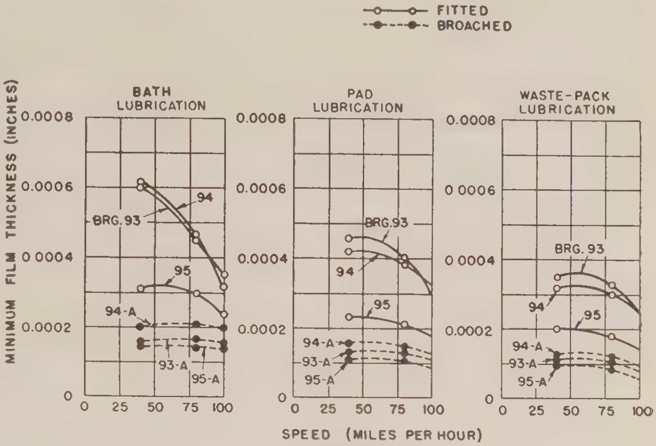

Fig. 16 Variation of Calculated Minimum Oil-Film Thickness With Speed
($5^1/_2 \times 10$ bearings; load 16,375 lb.)

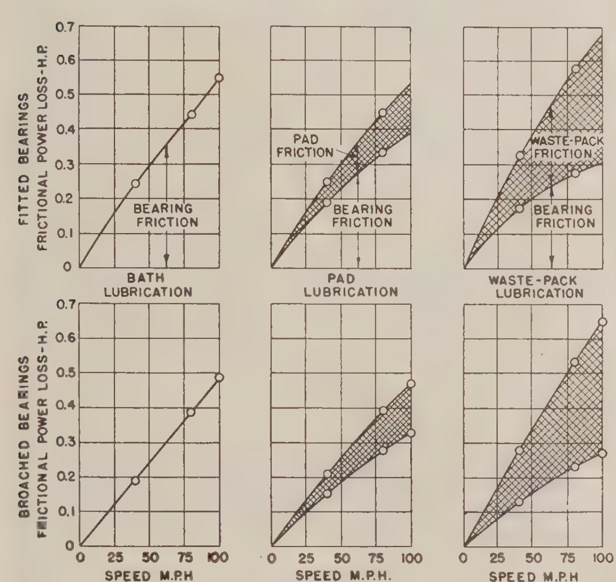

Fig. 17 Variation of Frictional Power Loss With Speed
($5^1/_2 \times 10$ bearings; load 16,375 lb.)

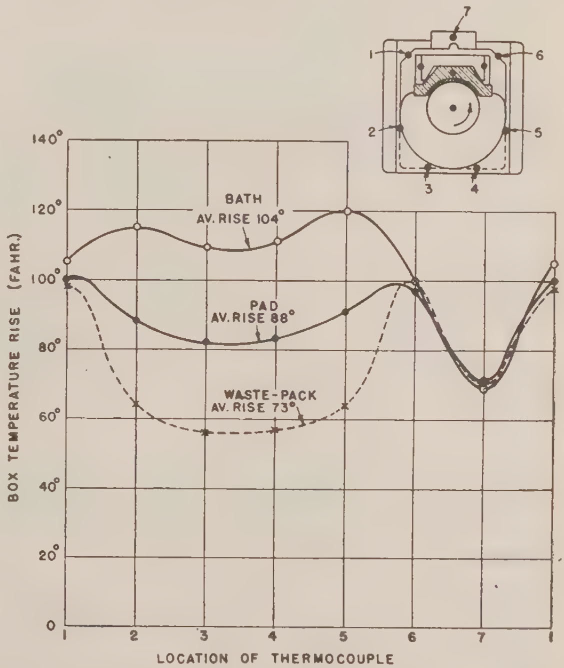

Fig. 18 Temperature Variation Around Longitudinal Center of Box With Bath, Pad, and Waste-Pack Lubrication
($5^1/_2 \times 10$ fitted bearings at 80 mph; load 16,375 lb; average room temperature 91 F.)

The situation becomes more pronounced with the waste pack. In this case there is little, if any, cooling effect of the oil. Friction between waste and journal is relatively high, and the pack acts as an effective heat insulator between journal and box, as shown by the lower box temperature rise. As temperature increases, rate of oil viscosity drop with temperature decreases and decrease in bearing friction becomes less rapid. The sum of waste-pack and bearing frictions becomes greater than with bath or pad lubrication, and temperature gradient between journal and box increases in order to dissipate the additional heat. Higher operating temperatures are undesirable, even when accompanied by reduced bearing friction, since film thickness and bearing safety are less at higher temperatures.

Variation of box temperature rise with journal temperature rise is shown in Fig. 19, for the three methods of lubrication. With perfect heat transmission between journal and box the temperatures would be the same at these points. This condition is represented by the dot and dash line A, the slope of which is unity. Slopes of the other curves show clearly the relative heat-insulating effects of the three methods of lubrication. From Fig.

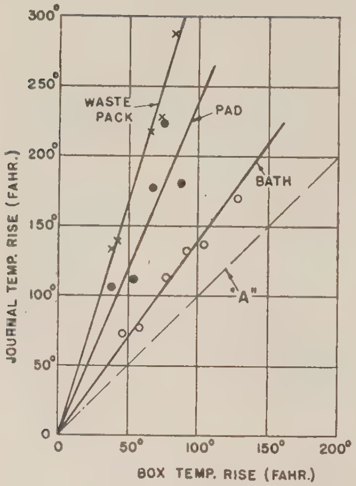

Fig. 19 Temperature Gradients Between Journal and Box
($5^1/_2 \times 10$ bearings; load 16,375 lb; average room temperature 93 F.)

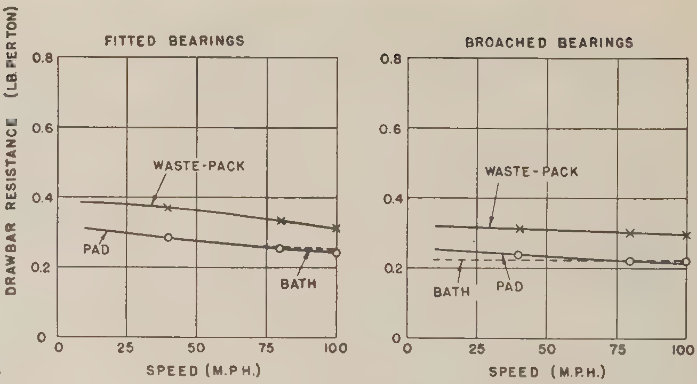

Fig. 20 Variation of Drawbar Resistance With Speed
($5^1/_2 \times 10$ bearings; load 16,375 lb.)

19 average temperature rise of the box is 72 per cent of the temperature rise of the journal with bath lubrication, 42 per cent with pad, and 30 per cent with waste pack. The importance of this relationship in maintaining cool bearings is obvious.

Friction losses given in Fig. 17 are readily converted into drawbar resistance. Variation of drawbar resistance with speed is given in Fig. 20 for the fitted and broached bearings, bath, pad, and waste-pack lubricated. Resistance is expressed in pounds per ton carried by the bearing and, as usual, speed is given in miles per hour. A 36-in-diam wheel is assumed. Since frictional power losses with bath and pad lubrication are closely alike, drawbar resistance is also nearly the same. Due to higher power losses with the waste-packed bearings, average drawbar resistance is about $^1/_3$ greater than with the other two methods of lubrication.

Conclusions

Fitted and broached bearings have been tested at 40, 80, and 100 mph. Data at 20 and 60 mph would have been useful in plotting the curves and for confirming results at the other speeds. On the basis of our present information, however, the following conclusions appear warranted:

1 Pad and waste pack are capable of supplying sufficient oil to maintain complete fluid-film lubrication at speeds at least as high as 100 mph.

2 The railway-car bearing is capable of carrying its maximum load without failure at temperatures at least as high as 375 F.

3 Operating temperatures and frictional power losses are somewhat less with the broached than with the fitted bearing. These advantages are offset by the fact that under similar operating conditions the oil-film thickness of the broached bearing averages only $^1/_3$ that of the fitted bearing. Hence the broached bearing has less reserve load capacity and safety.

4 In a waste-pack-lubricated box, friction of the waste pack is approximately the same as the friction of the bearing.

5 Friction of the pad used in these tests is about $^1/_3$ that of the waste pack.

6 Total friction with bath and pad lubrication are nearly the same, due to the higher operating temperature caused by the pad.

7 Total friction with waste-pack lubrication is higher than with bath or pad despite the higher operating temperatures due to the friction and blanketing effect of the waste pack.

8 Because of its lower operating temperature the bath-lubricated bearing has greater oil-film thickness, hence greater reserve load capacity and safety. This is particularly important at high speeds.

9 Frictional power loss and drawbar resistance of the waste-pack-lubricated bearing are approximately $^1/_3$ greater than with bath or pad lubrication.

10 Method of lubrication has a profound effect on the heat-dissipation characteristics of the box and the temperature gradient between journal and box.

From the foregoing discussion and conclusions it is clear that the crown, or useful load-carrying area of the bearing, should be as large as possible since it results in reduced unit load, increased film thickness, and less wear. It would be interesting to check these conclusions in actual service on a car or tender making considerable mileage per month. The same type bearing could be applied to all journals. One half should be fitted and the other half broached. Wheels, axles, and trucks should be in good condition so that the test would not be terminated by bearing end wear or broken collars. Periodic inspections and careful measurements of wear should give some interesting and impressive information on bearing reliability and life.

Acknowledgment

The author greatly appreciates the assistance of the Resident Committee and the splendid co-operation of Mr. Pearce and his staff. Acknowledgment is especially due L. G. Taylor and J. Roller for the careful manner in which they conducted the series of test runs reported in this paper.

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Discussion

P. G. Exline.[4] The writer is particularly interested in this paper because of some work of a similar nature carried out at the laboratories of the Gulf Research & Development Company during 1942. A standard A.A.R., 5 × 9 journal and bearing was used for a series of tests to compare the performance of a bearing having a normal diametral clearance of $^1/_{16}$ in. with that of a bearing having 0.005 in. clearance. The load, 30,000 lb, was above normal, while the speed range was equivalent to $2^1/_2$ to 30 mph on a 36-in. wheel.

Measurements were not made of the coefficient of friction, but the bearing temperatures indicated a qualitative agreement with the data reported by the author.

It was noted that after a period of operation at 30 mph, the standard-clearance bearing would often show signs of distress when the speed was reduced to $2^1/_2$ to 5 mph. This did not occur with the 0.005-in-clearance bearings, indicating a higher factor of safety. The standard-clearance bearing also took a longer time to accommodate itself to a reversal of the direction of rotation even though it had been well run-in in the forward direction. In this respect, the low-clearance bearing behaved as one which had been well run-in for both directions of rotation.

The test journal was mounted substantially the same as the Pearce machine, in that it was supported by a spherical roller bearing at the wheel seat. This bearing was cooled by adjusting the temperature of the stream of oil pumped through it. It was found that changes in the temperature of this bearing could influence the temperature of the test bearing. It seems evident that the wheels of a railroad car can act as cooling fins and dissipate an important fraction of the heat developed in the oil film.

The author's results and those of the writer suggest that a properly designed railroad bearing with bath or oil-ring lubrication and having adequate protection against oil loss and tampering would provide trouble-free performance with less maintenance than the present bearings.

M. D. Hersey.[5] This report not only tells how the problem of conservation of bronze and babbitt was solved, but offers in connection therewith a large amount of carefully checked information on the design and performance of railroad-car bearings. Of special interest is the use made of the theory of lubrication to calculate the "all-important" minimum oil-film thickness, and also the friction loss, with the aid of actual temperature measurements.

Might it not add to the permanent value of this record if the author would refer to the formulas or curves used for these calculations? The necessary procedure is not entirely obvious to the writer, and doubtless many other students of lubrication would appreciate an outline of the steps in a completely worked example.

J. R. Jackson.[6] It was the writer's privilege to be associated with the author during the test program treated in this paper. The data made available through this research constitute a distinct contribution to the knowledge of railroad-car journal-bearing design and lubrication. Furthermore they indicate the direction for the betterment of the journal-box assembly of the future by utilizing more completely the advantages which an oil-lubricated solid-type bearing has over the present conventional waste-pack journal box.

It is perhaps difficult to understand why the present conventional design of railway-car journal-box assembly has been retained down through the years, and why advantage has not been taken of advances in the art of bearing lubrication to replace the waste-pack originally used and continued without essential change on the American railroads for a hundred years. The answer is simplicity and reliability together with long-established standards and practices of operating cars in interchange service, and the relatively large number of units in use.

It probably should be stated that the simple expedient of removing the waste pack and filling the bottom of the journal box with oil, as was done during the laboratory tests, would not work out in service because the oil could not be retained in the box. Bath or flood lubrication would necessitate a redesign of the conventional box to insure oil retention. This question of oil retention in the conventional journal box is also pertinent, in lesser degree, to some forms of pads or wick-type lubricators which have had limited service trials on railway equipment in this country. Completely redesigned journal-box assemblies, employing flood lubrication and other refinements, including provision for oil retention, have been developed and used in considerable numbers on some roads on equipment operated in on-line service. These have not found sufficient favor to be considered for adoption as standard in interchange freight service.

The increasing use of roller bearings for passenger equipment and their development for freight equipment will no doubt result in a reconsideration of the design, service, and economics of railway-car journal bearings in the immediate future. It is felt that this paper is a distinct contribution to the literature on this subject, and that a similar line of investigation of the roller bearing for railway equipment should be carried out.

L. B. Jones.[7] In studying this report it should be borne in mind that the comparison between broached and fitted bearings was made with broached bearings which were just "starting out," so to speak. In other words, in actual service the broached bearing will continue to enlarge its contact area until it approaches the condition of the fitted bearing, with resultant improvement in oil-film condition.

While the tests were run mainly to determine oil-film conditions, it will be of interest to railroad men to note that the A.A.R. standard journal lining gave a good account of itself, and nothing developed in the test to indicate the necessity for a change in its composition.

The two principal indictments of the time-honored waste pack, pointed out by Mr. Pearce in his comments on this paper, are again high-lighted in this report. The actual friction of the waste pack is equal to, or greater than, the bearing friction; and it also acts as a thermal insulator to discourage radiation of heat from the journal.

The drawbar resistance in pounds per ton due to the friction load on the journal, as indicated in Fig. 20 of the paper, shows that the journal friction is a relatively small part of the total resistance of loaded freight cars on level track, as determined by dynamometer tests. It therefore appears that improvements in journal-box performance will pay their largest dividends in hot-box insurance rather than in decreasing the rolling resistance of the train.

F. K. Mitchell.[8] The author prefaces his remarks on the his-

[4] Section Engineer, Gulf Research & Development Company, Pittsburgh, Pa. Mem. A.S.M.E.

[5] Research Associate in Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Fellow A.S.M.E.

[6] Engineer of Tests, Missouri Pacific Lines, St. Louis, Mo. Mem. A.S.M.E.

[7] Engineer of Tests, The Pennsylvania Railroad, Altoona, Pa. Mem. A.S.M.E.

[8] Assistant General Superintendent of Motive Power and Rolling Stock, New York Central System. Mem. A.S.M.E.

tory of the improvements on railway-car bearings by the statement, "Despite the importance of the railway-car bearing and the great number replaced annually, little effort has been made to improve its performance." If he intended to say by that remark that the bearing itself had undergone very little improvement prior to the war period, his statement is unquestionably true, but the insinuation that little effort has been made to improve the performance of the bearing cannot be substantiated. Over the years, journal boxes, dust guards, wedges, and other related parts used in the assembly, of which the journal bearing is a part, have been subjected to a great deal of experimentation, and many important facts have been learned thereby, and certain very productive changes worked out and put into practice.

It is essentially true that the most extensive changes in the bearing itself have come about since the beginning of the war, and are children of necessity. These changes involved not only the bearing but also the lining and were promulgated through the necessity of minimizing the amount of critical materials required to produce a bearing of a certain size. As to the lining, the efforts in this respect have been confined generally to reducing its thickness and to producing a lining made of substitute material, both in order to save babbitt. It is generally conceded that the reduction in the thickness of the lining accomplished the required end. Substitute lining described by the author as a metal composed almost entirely of lead, alkali-earth-hardened, has not been entirely satisfactory, which fact cannot be discounted. On the other hand, the results so far obtained in the use of such a lining certainly do not warrant the complete discarding of the idea of its use. On the contrary, it is evident from the experience to date that further experimentation and development of such a lining offer worth-while possibilities.

The second series of major changes in the bearing itself, made to reduce the amount of critical material in the bearing back, has been enlightening, and also seems to be producing the desired results. There are still some features in connection with the various redesigned backs which have been offered for this purpose that need further study. These studies will no doubt be undertaken by the committee, of which the author is a member, in due time. Further comment in this regard will be made later.

It is unfortunate that the laboratory test equipment which was used to make the study under discussion does not duplicate in a practical manner the actual service conditions to which bearings are, in fact, subjected. It will be noted that the bearings were subjected to a constant uniform load of 16,375 lb. In actual service the load, of course, varies. Furthermore, in actual service the bearing is subjected to severe vertical and longitudinal shocks which the test equipment as used was unable to duplicate. This, it is felt, in itself materially affects the practicability of the information developed. It is pointed out by the author, under "Discussion of Results," "maintenance of sufficient oil-film thickness is the deciding factor in successful bearing operation." Therefore it would appear quite evident that the maintaining of such an oil film under the conditions imposed by the test apparatus is an entirely different problem from that which is found in actual service where, even under normal conditions, the bearing is constantly being disturbed by vertical shocks originating from uneven rail joints, etc., and longitudinal shocks arising from brake applications, slack action, etc.

While discussing the matter from the point of view of oil film, it should also be understood that with the laboratory test equipment used in these experiments, the journal being the driving instead of the driven element, the greater bearing clearance occurs on the rising side of the journal, whereas in actual practice the journal is driven by the bearing and the greater clearance is on the opposite side of the bearing. Hence under laboratory conditions the oil film produced by any method of lubrication, and under any bearing conditions, would be more favorable than under circumstances of actual usage where the clearance is the minimum on the rising side of the journal.

These facts in themselves make questionable the acceptability of some of the results obtained in this series of tests. They certainly do so to the point of discrediting the definite conclusion that "pad and waste pack are capable of supplying sufficient oil to maintain complete fluid-film lubrication at speeds at least as high as 100 mph, and that "the railway-car bearing is capable of carrying its maximum load without failure at temperatures at least as high as 375 F." Here it might also be noted that because many failures of bearings result from causes other than heat generated, the design of the back is an important factor, and unless it is shown that the back, however physically composed, is not only so designed as to distribute uniformly the load over the bearing area, but also is of such strength as to prevent breakage, then it cannot be concluded that the railway-car bearings tested were capable of carrying their maximum load without failure; this regardless of temperature. It is quite obvious that no failures from causes other than heating would be expected when the tests were conducted with the laboratory equipment which was used in these experiments, and the answer to this question was not obtained for any of the bearings under test.

It might also be said in passing that in the observation regarding heat transfer from the bearing to the related parts of the bearing-and-box assembly two confusing statements are incorporated in the report. The first is that the depressed-back (E. S. Pearce patent) bearing applies the load near the ends of the bearing instead of at random. Actually, this bearing applies the load over all but the one third of the area in the center. The application of the load near the extreme ends of the bearing has long since been proved to be mechanically incorrect. It is further stated that the back of the so-called "depressed-back bearing" is now machined. It will be found on check of the records that only a few railroads actually machine the back of the so-called "depressed-back bearing" for bearings used in passenger service, and that generally no roads are machining the depressed-back bearing used in freight service.

As to the transfer of heat, because of the greater contact area and therefore the improved heat transfer to the wedge, it will be interesting to note that bearings 93 and 95 have the large back area referred to, whereas bearing 94 has only four small bearing areas, nowhere comparable to that offered by bearings 93 and 95, yet the tests show that bearing 94 produced journal-temperature, bearing-temperature, and box-temperature rises at various speeds which generally lay between that of bearings 93 and 95. No explanation for this fact either in the case of the fitted bearing or the broached bearing is offered.

Regarding the question of whether or not the broached and unfitted bearing is more satisfactory than the broached and fitted bearing: The author concludes, and the data developed indicate, that better results can be expected from a fitted bearing. While this is interesting, and no doubt true, it has no practical value for the reason that the expense and delay involved in fitting the bearing, particularly to freight cars, would in no wise be offset by the results obtained.

The data developed and conclusions reached in connection with the relative values of bath, pad, and waste lubricating systems would indicate that the committee has concluded from the results obtained that waste-packed lubrication is less satisfactory than pad lubrication and bath lubrication. This is no doubt true if it were not for the practical side of the problem. Certainly bath lubrication is a far cry from being practical at present. Pad lubrication is more or less in its infancy and offers considerable possibilities, yet, at the moment, the writer knows of no practical means of avoiding excessive loss of lubricant in the present freight-

car box assembly where either bath or pad lubrication is used. Here again a factor arises which the test equipment used by the committee does not bring out. The entire matter does emphasize the necessity for further experimentation and development along the desirable lines indicated by the tests.

As to the effect on drawbar resistance, neither the data having to do with the friction of the broached or unbroached bearing nor pad nor waste-packed boxes are of much practical value.

It is worthy of note that speeds for use in connection with these experiments (40, 80, and 100 mph) were selected. While 40 mph represents pretty closely the present freight-train movement, the immediate future no doubt will mean speeds for freight-train movement between the figures of 40 and 80, throughout which range the report is silent as to what might be expected. The author comments on this fact in his conclusions. It is regrettable that data were not also secured for speeds of 50, 60, and 70 mph.

While as previously indicated, the data submitted in this report are interesting, although in a great many respects somewhat impractical, it is respectfully suggested that either test equipment which will more closely simulate actual operational conditions should be designed and used for such a series of experiments, or final conclusions should be based on actual experience in the field.

E. S. Pearce.[9] On certain phases of railroad journal operation this paper sets out most clearly two points of general and practical interest:

1 The very low power loss chargeable to the conventional journal-box assembly.

Power loss in journal friction as a factor in total train resistance is a particularly timely subject, as the railroads are contemplating operation at higher maximum and higher average speeds. The values given by the author cast considerable doubt on the advisability of substitution of roller bearings for solid bearings as a matter of sound economics.

2 The separation of the power loss between bearing and lubricating medium.

To those confronted with the everyday problems of journal operation, the significance of packing friction losses relative to those of the bearing alone will be of considerable practical interest.

Analyzing the author's paper in the light of the two major disclosures, obviously, much of constructive value to the railroads in the operation of a conventional journal-box assembly would result from a further exploration of this subject through a lower range of speeds and temperatures in combination with various methods of lubrication, lubricants, and bearing construction now available.

The accepted formula for calculating freight-car resistance on straight level track is[10]

$$R = 1.3 + \frac{29}{w} + 0.045V + \frac{0.0005AV^2}{wn} \quad \ldots\ldots\ldots [1]$$

Fig. 21 of this discussion has been superimposed upon the author's Fig. 20 demonstrating what a small portion of total resistance can be charged to journal friction at the load and speeds used in the paper.

The values given by the author for power losses are roughly one tenth of that given by the first two terms of Equation [1] of this discussion.

Journal operating temperatures in this paper are at a level much higher than the generally accepted limit for safe performance in service. This is due in large part to operation in the still

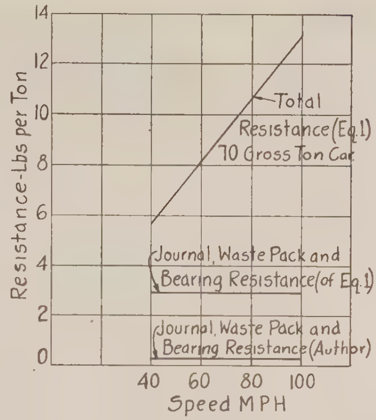

Fig. 21 Relation of Journal Resistance to Total Resistance of a Freight Car

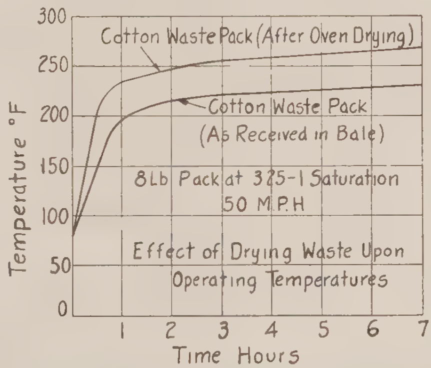

Fig. 22 Effect of Drying Waste Upon Operating Temperatures

air of the laboratory. Pearce[11] shows the magnitude of this difference at lower speeds than covered in this paper.

Another contributing factor to the high temperature obtained with the waste pack was the use of "oven-dried" waste as so designated by the author. In investigations of this kind, in the interest of accurate duplication of results, oven-drying of waste is done to offset the variation in moisture content and the effect on temperature which would otherwise exist between various samples of waste. In preparing the waste for test purposes it is dried at a temperature of 220 F. The effect of this drying upon operating temperatures is shown in Fig. 22 of this discussion.

The contribution of the lubricating medium to the total power loss of the bearing assembly has never been fully appreciated. The placing of a waste pack in a journal box is very similar to the installation of an opposed bearing, as there is an upward pressure of the waste pack against the lower half of the journal. Fig. 23 shows how this contributes to the elevation of the running temperature as the amount of packing at the same oil-waste ratio, is varied in the same journal box.

To verify the power losses of the lubricating medium, as shown in Fig. 14 of the paper, a study was made of the reduction in power input to the driving motor by the momentary lowering of the lubricating medium from contact with the lower half of the

[9] President, Railway Service and Supply Corporation, Indianapolis, Ind. Mem. A.S.M.E.

[10] "The Steam Locomotive," by R. P. Johnson, Simmons-Boardman Publishing Corporation, New York, N. Y., 1944, p. 184.

[11] "Temperature Distribution and Sources in the Conventional Railway Journal Box," by E. S. Pearce, Trans. A.S.M.E., vol. 62, 1940, pp. 633–638.

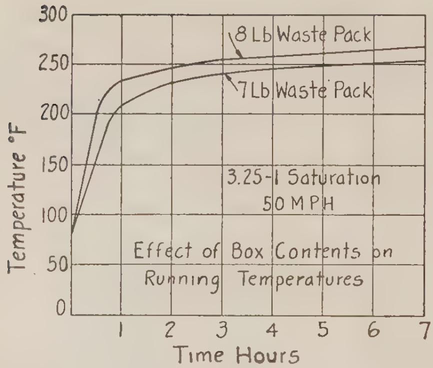

Fig. 23 Effect of Box Contents on Running Temperatures

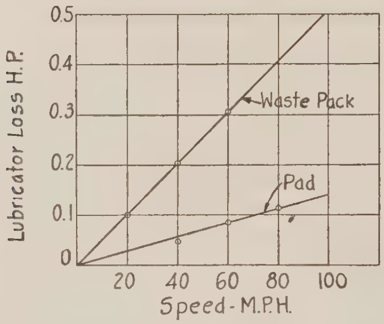

Fig. 24 Frictional Power Loss of Pad and Waste Pack, as Determined by Momentarily Dropping Lubricator From Contact With Journal

journal. This was accomplished by the use of the bearing described by Pearce[12] as bearing No. 18, this bearing being capable of operation for sustained periods on the stored oil. This time interval was quite adequate to take the necessary instrument readings. The box used was split on its horizontal center line and mounted on a jack so that it could be quickly lowered and again raised. Results are shown in Fig. 24 of this discussion, and comparison with the author's Fig. 14 indicates that the calculated loss of the lubricating medium is in good general agreement with the experimental findings.

R. J. Shoemaker.[13] In connection with the results obtained with pad lubrication reported in this paper, we may state that one of the major trunk-line railroads of this country has been using pad lubrication (Magnus type) very successfully for a number of years past in connection with oil lubrication of driving-box brasses of main passenger and freight power, also on main-line passenger cars, locomotive tender, and locomotive truck and trailer brasses.

By the use of pad lubrication, considerable economy of operation has been accomplished as shown by reduction in the number of hotboxes, increased service life of bearings, and corelated parts, reduction in friction, etc., as compared with results obtained with conventional lubrication formerly used.

With regard to the economy of materials accomplished in the tests of new types of bearings reported in the paper, the author states, "Since there appears to be no satisfactory substitute for babbitt, reduction in quantity is the only open course." The author further states that in redesigning the prewar type of journal bearing, the babbitt metal was reduced 50 per cent in weight, or in other words, reducing the lining metal from $^1/_4$ to $^1/_8$-in. thickness, as was done in the A.A.R. emergency-type bearing used in the early years of the war. However, in view of the unsatisfactory performance in service of this $^1/_8$-in. lining, the babbitt metal was later restored to its original $^1/_4$-in. thickness, as now used by the railroads in the present type of A.A.R. emergency journal bearing.

With regard to a satisfactory substitute for babbitt metal, we may mention that our company developed and put on the market a number of years ago an alkali-earth-hardened lining metal for bearings known as "Satco" metal, which is having widespread and successful use as a substitute for lead- and tin-base babbitts, block tin, etc., in various types of railway and other bearings.

This metal is a lead-base alloy containing from 95 to 98 per cent lead with the balance calcium, tin, and other hardeners. It contains no antimony or copper. During the late war great economies were effected by the use of Satco bearing metal in vital materials such as tin, copper, antimony, etc.

Satco metal has a melting point approximately 150 F higher than that of babbitt metals with a correspondingly greater hardness at elevated temperatures. Tables 5 and 6 of this discussion show comparative physical properties of Satco metal, A.A.R. babbitt, tin-base babbitt and antimonial lead at normal and elevated temperatures.

Table 5 Degrees Brinell Hardness of Various Bearing Metals

(500 kg load)

Deg F	Satco	Tin-base babbitt	A.A.R. babbitt	Antimonial lead
70	23.8	22.2	17.8	15.0
212	17.2	12.9	9.5	6.2
300	12.0	7.6	4.2	*
400	7.7	*	*	*

* Indicates metal too soft to measure hardness using 500-kg load.

Table 6 Compressive Strength of Various Bearing Metals

(Pounds per square inch; 25 per cent deformation)

Deg F	Satco	Tin-base babbitt	A.A.R. babbitt	Antimonial lead
70	16300	17200	15600	13000
212	10200	7500	6100	5000
300	7000	4000	2700	1100
400	3700	2000	1200	*

* Metal too soft to determine compressive strength due to excessive deformation.

Because of its greater heat resistance Satco-lined bearings perform satisfactorily at temperatures which cause failure by melting in babbitt-lined bearings, consequently the alloy is more resistant to "waste grabs" which are a frequent cause of failure in railway bearings.

In addition to the various types of railway-car and locomotive bearings, Satco metal is also used as a lining in main and connecting-rod bearings and motor-support bearings of Diesel engines. The successful performance of the alloy in this service has been reported by L. M. Tichvinsky.[14]

The author further reports that the lining metal in the "magnus special" bearing tested by the committee showed indications of corrosion during the 80-mph schedule. In this connection we wish to state from our experience with Satco-lined bearings, that corrosion is the exception rather than the rule. In the railway field particularly, corrosion of Satco linings is of rare occurrence.

Under certain conditions, however, particularly with lubricating oils which are compounded with free fatty acids, all types of lining metals regardless of their composition will corrode to some extent.

[12] "Locomotive and Car Journal Lubrication," by E. S. Pearce, Trans. A.S.M.E., vol. 58, 1936, pp. 37–45.

[13] Magnus Metal Corporation, Chicago, Ill.

[14] "Diesel-Engine Bearings," by L. M. Tichvinsky, *Mechanical Engineering*, vol. 67, 1945, pp. 297–308.

Author's Closure

The discussion has brought to light some additional experimental work on the subject. It is hoped that Mr. Exline plans to publish the results of the railway-car journal bearing studies conducted in the laboratories of the Gulf Research and Development Company.

Data on lubricator power losses furnished by Mr. Pearce are most interesting inasmuch as they check the results given in the paper by an entirely independent method. With waste-pack lubrication, Fig. 24 shows the losses somewhat greater than the author's Fig. 14. Losses with the pad are in close agreement. Mr. Pearce's direct method of obtaining his data is probably somewhat more accurate than the author's indirect approach. The results, however, are in good agreement as has been pointed out, and may be said to verify some of the conclusions reached in the paper. Mr. Jones's comment that journal friction is a relatively small part of the total car resistance is well illustrated in Fig. 21.

Detailed procedure in the calculation of bearing oil-film thickness and friction, referred to by Mr. Hersey, was purposely omitted to reduce the length of the text. Basic data for these calculations will be found in previously published papers by Kingsbury[15] and Needs.[16]

The test results bring out the very interesting fact that despite the relatively large clearances of the railway journal bearing, clearances far beyond those permitted in other machinery, the bearing friction is slightly less than with the more normal clearances obtained by running the fitted bearings at operating temperatures. In other words, it appears that the friction of the bearing itself is about as low as can be expected and improvement can be hoped for only through method of lubrication. In this respect, statements by Mr. Jackson and Mr. Shoemaker that some of the railroads of this country have been using flood and pad lubrication for a number of years are significant.

A comparison of the test results given in the paper with similar data on roller bearings for railroad equipment as suggested by Mr. Jackson would be very interesting indeed and it is hoped that the Committee will be able to release such information in the not too distant future.

Mr. Shoemaker's data on variation of bearing-lining hardness with temperature are particularly interesting. From the fact that bearing 95-B carried a load of 1930 psi at 375 F with "A.A.R. babbitt" there appears to be ample babbitt hardness safety factor when operating at the usual loads and temperatures encountered in service.

That part of the discussion which is critical stresses the fact that a bearing under test in the laboratory is not meeting actual service conditions, and for this reason the test results are not of practical nature. When planning the tests the difference between laboratory and service conditions was realized. For example, it was not expected that the machine in which the tests were run would give any information on the ability of the bearing to withstand collar cracking forces. Such information is readily obtained by placing the bearing in service. The main objects of the laboratory tests were to investigate bearing friction and method of lubrication under steady comparable conditions and to compare these characteristics with those of the fitted bearing on the assumption that the fitted bearing was the best obtainable for the service. Since all successful bearings in actual service operate by virtue of their oil films, comparisons under comparable conditions of various methods of feeding these oil films are entirely valid, hence practical, regardless of whether the data are accumulated in the laboratory or on the railroad. In order to make the tests comparable they had to be run under steady conditions and since there are very few stretches of track where a train can run 80 or 100 mph for eight consecutive hours, the difficulty of obtaining our data from service is apparent. In some respects the conditions of the laboratory tests were more severe than the bearing would meet in service. Proof of this is the fact that every bearing surviving the laboratory test gave a good account of itself in service.

One or two other points raised by Mr. Mitchell can be answered more specifically. It is quite true that in a railroad car the two opposite bearings push the axle and wheels along. The horizontal component of the load which provides this force is essentially small as compared with the vertical component due to the weight of the car and its load. However small the horizontal load may be it does cause the bearing center line to lead the journal center. This, however, has no effect whatever on formation of the oil film. Only the converging portion of the oil film can possibly generate positive load-carrying pressures. In the testing machine the load-carrying area is in the center of the bearing as shown in Figs. 11, 12, and 13. Because the bearing must carry the axle and wheels along in actual service the load-carrying bearing area will be shifted from the center a very slight distance opposite the direction of rotation. Under comparable operating conditions the films will be exactly the same otherwise.

Regarding heat transfer from bearing to wedge, the test results seem to indicate that the smaller contact area of bearing 94 is not a matter of concern. As Mr. Mitchell has pointed out, the operating temperatures of bearing 94 are much the same as the temperatures of the other two bearings under the same operating conditions. This would be expected inasmuch as friction is approximately the same with each of the three bearings. It will be noted from the tables, however, that the temperatures of bearings 93 and 95 are generally several degrees lower than the journal. With bearing 94 this difference is only a degree or so, the smaller difference being due to less contact area between bearing and wedge.

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[15] "Optimum Conditions in Journal Bearings," by A. Kingsbury, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-7, pp. 123–148.

[16] "Effects of Side Leakage in 120-Degree Centrally Supported Journal Bearings," by S. J. Needs, Trans. A.S.M.E., vol. 56, 1934, paper APM-56-16, pp. 721–732.

Dimethyl-Silicone-Polymer Fluids and Their Performance Characteristics in Unilaterally Loaded Journal Bearings[1]

By J. E. Brophy,[2] R. O. Militz,[3] and W. A. Zisman[4]

This investigation concerns the practicability of using the dimethyl silicone polymer fluids as lubricants in unilaterally loaded journal-bearing machines. A specially designed bearing machine was used having forced-feed lubrication up to pressures of 26 psi. It was operated at a speed of 1725 rpm, and the oil-sump temperatures varied from 115 to 130 F, while the bearing temperatures varied from 150 to 240 F. Plain or chrome-plated high-carbon-steel shafts were used in conjunction with bearings of copper-lead, bronze, tin-base babbitt, cast iron, commercial brass, aluminum (17S), copper and Alfin metal. Cast-iron or steel bearings used in conjunction with steel journals were entirely unsatisfactory when lubricated with the silicone fluid. All of the nonferrous bearings performed satisfactorily with the silicone fluid. It was found that an organic silicon-containing film was formed during the long, slow, break-in process. The need for such a long break-in could be eliminated by the application of a technique of pretreating or lacquering the bearings with silicone oil before assembling the test machine. The method of pretreatment is described briefly. It was also discovered that such pretreated bearings were able to carry much higher than normal loads when operated in contact with petroleum lubricating oil.

Introduction

This is part of a larger investigation made by the Naval Research Laboratory during the war at the request of the Bureau of Ships. It concerns the physical and chemical properties of the silicone polymer fluids (or polyorganosiloxanes) and their application to, and development for, naval lubrication problems. As the dimethyl silicone polymer fluids have the smallest temperature coefficients of viscosity of any of the available silicones (or of any other known pure liquids in the viscosity range of lubricants), and as they are especially stable to oxidative decomposition, they have been the most thoroughly studied, and they alone are discussed here.

The dimethyl silicone fluids are included among the various new silicones developed and manufactured by both the Dow Corning Corporation and the General Electric Company and are now commercially available from each supplier under his own trade name.[5,6] At the start of this investigation only one manufacturer was in commercial production and able to supply large enough quantities of the desired fluid; hence all the data given here relate to several large batches of a well-reproduced fluid made by the Dow Corning Corporation.

Comparisons with the latest available and comparable grades of dimethyl silicone polymer fluids manufactured by both producers showed them to behave identically as lubricants. The fluid used here was specially stripped of volatiles, making it noninflammable to incendiary fire. It was identified by the producer as Dow Corning fluid, series 500, batch 369–62–69, and its viscosity-temperature characteristics are summarized as follows:

Temperature, deg F	Kinematic viscosity, centipoises
210	29.8
130	55.6
100	73.5
0	275.0
—20	390.0
—40	600.0

Prior to the investigation of this laboratory all of the lubrication tests reported by the producers and by the numerous other laboratories having access to these fluids had led to the conclusion that the silicones were poor lubricants. However, these fluids were considered so promising in other respects and the need for low- and high-temperature lubricants and noninflammable hydraulic fluids was so great that it was deemed unsound to drop them from consideration on the basis of the usual quick tests for load-carrying capacity.

After some search, a simple technique was found suitable for observing quickly some of the lubricating peculiarities of silicones. This is described in another paper.[7] Using this technique to separate the promising metal combinations from those obviously having no promise, it was found that whereas a number of nonferrous metals were suitable for bearings, steel or cast iron were uniquely unpromising. It was therefore decided to study these extreme types of metal combinations in both a bearing machine and a hydraulic pump system. The results obtained in hydraulic systems are described elsewhere,[7] while the first results obtained in bearing machines are reported here.

Since the theoretically best understood and most common type of lubrication occurs in the unilaterally loaded journal-bearing machine, it was considered to be the appropriate machine for studying load-carrying capacity (or seizure load) as affected by the fluid, the break-in procedure, and the bearing composition. By going from light loads with full hydrodynamic conditions

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[1] The opinions or assertions contained in this paper are the authors' and are not to be construed as official or reflecting the views of the Navy Department.

[2] Associate Mechanical Engineer, Lubrication Section, Naval Research Laboratory, Washington, D. C. Mem. A.S.M.E.

[3] Lieutenant, U.S.N.R., Lubrication Section, Naval Research Laboratory, Washington, D.C.

[4] Head Lubrication Section, Naval Research Laboratory, Washington, D. C. Mem. A.S.M.E.

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[5] "The Silicones—A New Plastics Family," *Plastics*, vol. 2, January, 1945, pp. 112–115.

[6] "The Organo-Silicon Polymers," by E. G. Rochow, *Chemical and Engineering News*, vol. 23, 1945, p. 612.

[7] "Dimethyl-Silicone-Polymer Fluids and Their Performance Characteristics in Hydraulic Systems," by V. G. Fitzsimmons, D. L. Pickett, R. O. Militz, and W. A. Zisman, published on pages 361–369, of this issue of the Transactions.

prevailing, to heavy loads with intermittent metallic contact, it was believed that important peculiarities of the fluid could be noted. It was necessary to design a special bearing machine for this work.

Owing to the originally limited availability and high cost of the silicone fluid, this machine was of small capacity. Nevertheless it was desirable to be able to employ commercially available insert bearings and to secure adequate freedom from shaft deflections and bellmouthing of bearings, which meant that good bearing alignment and accurate boring and finishing of bearings were essential. Several models were built, the final one used for this investigation being shown in Figs. 1 to 4, inclusive.

Due to the poor lubricating value of the silicones in steel-on-steel or steel-on-cast-iron sliding loaded surfaces, the forced-feed oil system of the machine had to be operated with a low-pressure low-flow-rate gear pump designed to avoid such loading. The machine used was equipped with a specially designed gear pump having bronze bearings. Later models of the machine were equipped with Pesco 1P-349N aviation gear pumps run at low speeds.

Experimental Equipment and Methods

Fig. 1 shows the bearing machine completely assembled with weights on the pan of the loading arm. Fig. 2 is a partial test assembly of the bearing machine revealing the placement of the bearings, journal, and thermocouple. Fig. 3 is an assembled view of the support bearings, test bearing, journal, and flinger. Fig. 4 is a view of the component parts of the test assembly shown in Fig. 3.

The small motor A beneath the drain pan, Fig. 1, drives a special vertical, bronze, gear pump B. The pump is immersed in its own reservoir C to minimize the volume of fluid needed. The high-pressure feed line extends from the side of the pump to the aircraft-type Skinner filter D. One copper tube from the filter feeds the support bearings. This tube and its end connection to the support block are shown in Figs. 2, 3, and 4. Another copper tube from the filter goes to a needle valve and thence to a rotameter-type flowmeter E. The fluid pressure is measured at the outlet of the flowmeter. A coiled copper tube connects the flowmeter with the test-bearing holder F. The load on the bearing is applied through the long beam G and a secondary lever system that is connected to the test-bearing holder.

The method of supplying the lubricant to the bearings is shown in Figs. 3 and 4. The flare fitting on the block H is connected to both support bearings by drilled holes. The small hole at the top of the test bearing J shown in Fig. 4, is the supply orifice for the lubricant. The test holder is drilled to connect the flare fitting and the bearing. A spring-loaded thermocouple is screwed into the bottom of the test-bearing holder at point K and contacts the outer surface of the test bearing.

Plain bearings with one hole drilled top center (opposite loaded area) were used. The bore was 0.687 in. and the length of the bearing was $^1/_2$ in. The bearing clearance (bore diameter minus journal diameter) was held to 0.0010 in. $\pm$ 0.0001 in. A special boring technique, developed and made available to this laboratory by Russell Dayton of the Battelle Memorial Institute, was used to assure true cylindrical holes in the test bearing and support bearings. The journals were of chromium-plated high-carbon steel with the exception of that used in run N, in which case it was copper-plated steel. The bearings were pressed into the test-bearing holder and the support block. If rough-boring was necessary, it was accomplished before pressing the bearings into the holder and block. Finish-boring was accomplished in the lathe in a special fixture using the boring technique mentioned previously. This fixture was built to isolate the eccentricities of the lathe spindle from the work and

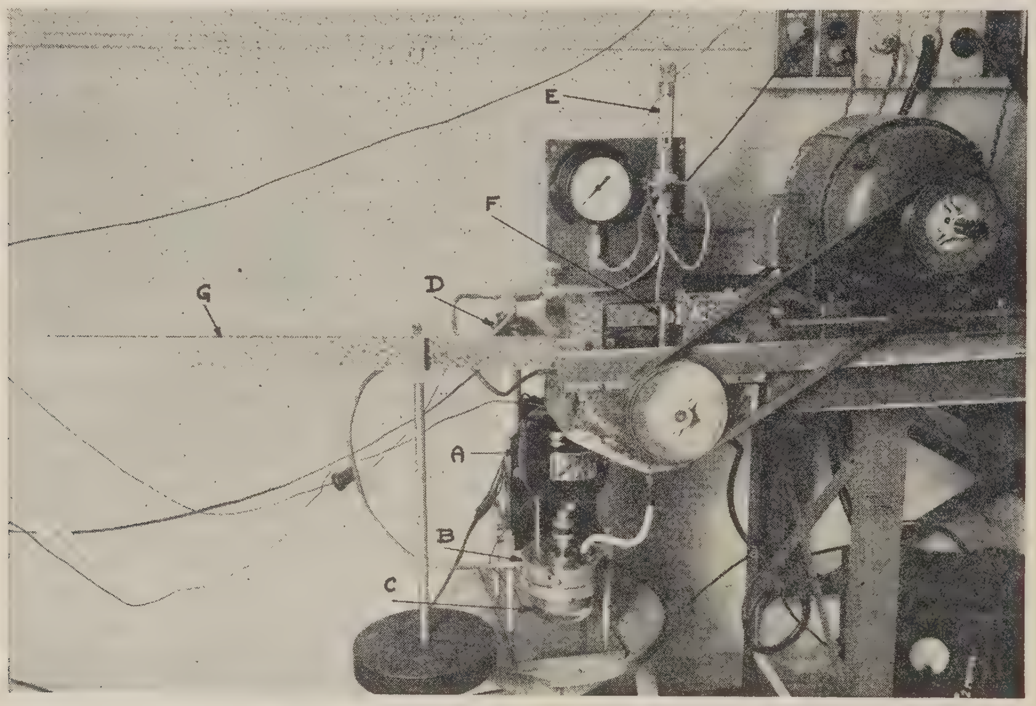


Fig. 1 Bearing-Test Machine, Assembled for Test Run

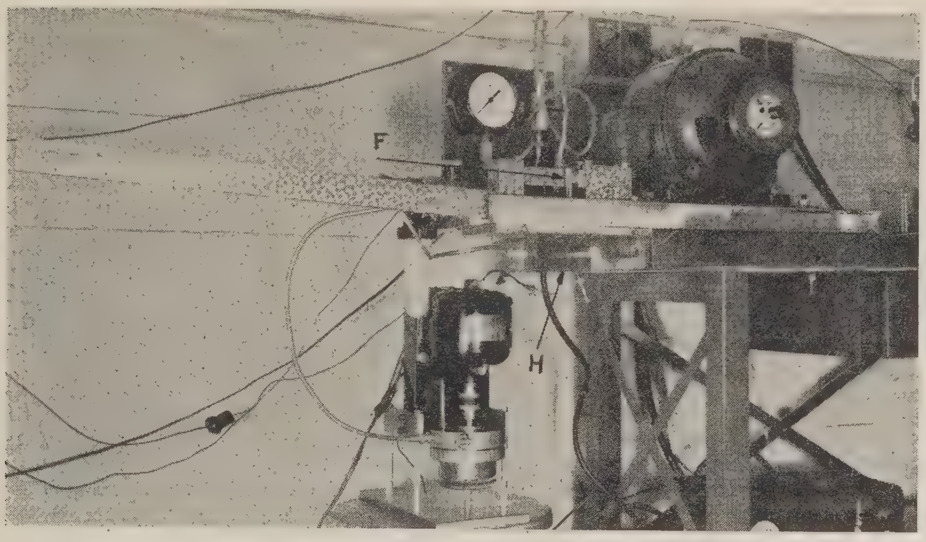


Fig. 2 Bearing-Test Machine With Oil Pan and Drive Pulley Removed

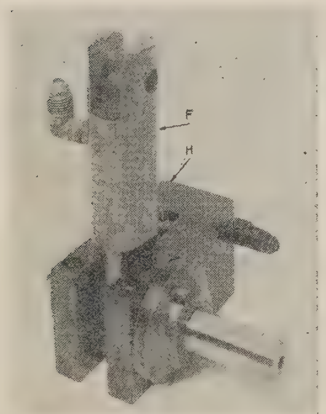


Fig. 3 Assembly of Support and Test Bearings

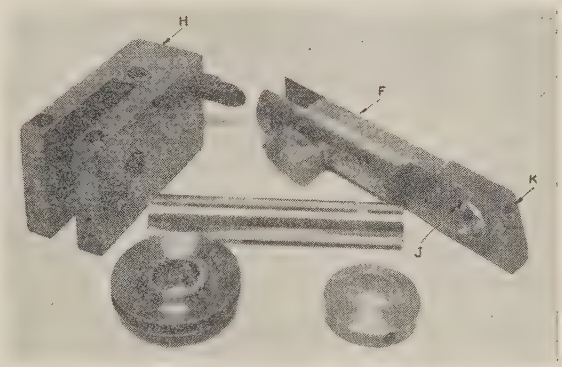


Fig. 4 Component Parts of a Bearing Test

to insure reproducible finishings in the bearing. All parts in the test setup, including the bearings, were washed either in unleaded gasoline or in benzene and blown dry with clean air. All bearing surfaces were wet with the silicone fluid prior to assembly in the machine.

Due to the low surface tension of the silicone fluid, approximately 20 dynes per cm, it was difficult to prevent leakage of the fluid from the system. In the first bearing machine developed for this work it was found that the standard labyrinth-type seal, used so much in modern equipment, was entirely unsatisfactory. Flinger rings were found to work if properly shaped. As for the standard synthetic-rubber seal rings, it was found necessary to change the composition of the rubber to a silicone plasticized Hycar compound, described in the accompanying paper on hydraulics.[7]

The gear pump was operated to deliver the silicone fluid to the test bearing at a pressure of approximately 26 psi. No attempt was made to maintain the same ambient temperature or bearing temperature throughout the various runs. Bearing temperatures ranged from 152 to 240 F, while temperatures of the oil in the sump ranged from 118 to 129 F. The journal speed throughout was 1725 rpm. The load in all the tests was limited to a maximum of approximately 6000 psi. This limit was imposed by the design of the machine, since higher loads would cause excessive shaft deflection and bellmouthing of the bearings.

After some experimentation, several procedures were adopted for breaking in the bearings. A slow break-in consisted of an initial overnight run of 16 hr at 500 psi, and thereafter an increase in the load of 100 psi per hr. This load was continued until failure occurred, or the load limit of the machine was reached (6000 psi). If no seizure occurred the load was cycled between 500 and 6000 psi. One run of 157 hr duration was made. Once broken in, the bearings were run 8 hr per day at 6000 psi after a 15-min warm-up run at 500 psi. The 157-hr run was stopped because of a failure in the power supply to the oil-pump motor. A fast break-in had only 2 hr initial operation at 500 psi, and the load was increased 500 psi every 15 min. A very fast break-in has been used in which the bearing was operated at 500 psi for 2 hr and then loaded to 3000 psi and increased to 6000 psi 15 min after that. The bearing was then cycled between 500 and 6000 psi.

The temperature of the test bearing was taken at the outer

surface of the bearing through the bottom of the holder as previously described. Temperatures were measured and recorded continuously on a Leeds & Northrup potentiometer having a range of 0 to 350 F, and a time of full-scale travel of 27 sec. With this type of instrument it is impossible to follow flash temperatures or temperatures at final seizure, but general operating temperatures were recorded very well. The temperature of the bearing rose slightly after each increase in load, but fell in a few minutes to an equilibrium temperature slightly higher than that for the previous load. This temperature-recording arrangement has proved satisfactory for observing incipient seizure. However, in the runs described in this paper, no difficulties with incipient seizure were found in the silicone-lubricated runs over the load range used.

The bearing metals used included cast iron, babbitt, a high lead and a high-tin bronze, copper-lead alloys containing high, medium, and low amounts of lead, copper-nickel silicide, copper (hard), commercial brass (half-hard temper), aluminum (17S), and Alfin metal. The chemical analyses and sources are given in Table 1. It should be understood that these materials are not the only ones of promise but simply all those investigated to date.

Some Load-Carrying Characteristics of Chemically Untreated Bearings

The first bearing machine built was tried with bearings of cast iron, 80-10-10 bronze, and Delco bushing (copper-lead), and steel-backed babbitt. The nonferrous bearings were satisfactory over the temperature range of 140 to 180 F up to the load-carrying limit of the machine (only 2200 psi). The cast-iron bearings operated under no load but failed promptly at loads of only 250 psi. Therefore a new machine was built permitting loading up to 6000 psi.

Using the break-in procedure already described, copper, Federal Mogul F-16, aluminum (17S), and Alfin alloy bearings were investigated in the journal-bearing machine shown in Figs. 1 to 4, inclusive, using the silicone fluid. The summarized results are given in Table 2, as runs A, B, C, and D. In all cases these metals, once broken in, operated as satisfactory bearings at loads up to the load limit of the test machine (6000 psi). Of course, this conclusion is restricted to the given range of bearing temperatures, oil-flow rates, and ZN/P values (Z, N, and P having the usual meanings, i.e., Z = viscosity; N = rpm of journal; P = bearing load per unit of projected area). The quantity PV (in which P = the load and V = journal velocity in feet per second) used to predict the performance of a bearing with a petroleum oil is given in Table 2. No significance can yet be given to these values for the silicone fluids until more data are collected. In all cases no significant change (± 1 per cent or less) in the viscosity of the fluid resulted.

In each of these runs, as in the earlier bearing tests mentioned, it was found that the slowly broken-in bearings not only developed high load-carrying capacities, but also retained them when the original silicone fluid was replaced by new and unused silicone oil. An obvious inference was that the bearing surface had been chemically or otherwise altered. A careful inspection of each bearing surface following the slow break-in procedure revealed that each loaded surface was coated with a thin transparent film or lacquer roughly 0.0001 in. thick and varying from water-white to pale yellow in color. Such a lacquer had also been found on the bronze bushings of the Pesco high-pressure gear pumps used in the hydraulic pump.[7]

It is now well recognized that bearings carefully run-in on petroleum lubricants slowly develop a dark lacquer coating on the loaded-bearing surface. As it was considered possible that the high load-carrying capacity obtained in a slow break-in with silicone oil was due to the formation of the observed film or

TABLE 1 ANALYSES OF BEARING METALS USED

Bearing metal	Cu	Pb	Sn	Zn	Sb	Fe	Ni	Al	C	Si	Mg	S	P	Mn
Delco bushing[a]	88.92	3.79	4.01	3.28	0.01	0.01	..	...	..	..	...	...	..	..
A.S.T.M.-B62-36	...	...	...	...	..	..	..	...	..	..	...	...	..	..
F-16 (copper-lead)[b]	74.63	19.04	5.47	0.6	0.1	0.5	..	...	..	..	...	...	..	..
F-1 (bronze)[b]	86.95	0.1	9.5	2.5	0.1	0.1	0.05	0.05	..	..	...	...	..	..
A.S.T.M. B60-36	...	...	...	...	..	..	..	...	..	..	...	...	..	..
Commercial brass (half-hard temper)	60.2	2.69	...	37.0	..	0.05	..	0.1	..	..	...	...	..	..
Copper bar (hard)	99.9	...	...	...	..	..	..	...	..	..	...	...	0.9	..
Aluminum (17S)	4.0	...	...	...	..	..	..	95.0	..	..	0.5	...	..	0.2
Alfin metal[c]	0.05	...	6.32	...	..	0.89	..	93.0	..	0.07	...	...	..	..
Copper-nickel-silicide[d]	96.0	...	...	...	..	..	2.19	...	..	0.8	...	...	..	..
Babbitt	3.77	0.07	89.2	...	7.64	0.01	..	...	..	..	...	0.088	0.40	..
Cast iron	...	...	...	...	..	...	..	...	3.36	1.2	...	...	..	0.48
Bronze	79.9	10.0	10.0	0.02	0.01	..	..	...	..	..	...	...	..	..

[a] Manufactured by Delco Appliance Division, General Motors Corporation (Pc. No. 5029150).
[b] Manufactured by Federal-Mogul Corporation.
[c] Manufactured by Al-Fin Corporation, Jamaica Long Island, N. Y. (obtained through courtesy of N.A.C.A.).
[d] Manufactured by P. R. Mallory & Company, Inc. (known as Mallory No. 53).

TABLE 2 RESULTS OF JOURNAL-BEARING TESTS WITH SILICONE FLUID

Run no.	Bearing material[a]	Bearing load,[b] psi	Bearing temp.,[c] deg F	Oil-sump temp., deg F Initial	Final	$\frac{ZN}{P}$	PV	Treated bearings	Type of break-in
A	F-16	6000	165	118	122	11.5	31400	No	Slow
B	Aluminum (17S)	5300[d,e]	145	120	127	15.5	27200	No	Slow
C	Alfin	5700[d,e]	163	127	127	12.1	29700	No	Slow
D	Copper	6000[d]	163	122	122	11.5	31400	No	Slow
E	Delco	6000	152	118	127	12.7	31400	Yes	Fast
F	Delco	500[d]	...	...	...	..	2590	No	Fast
G	Delco	6000	240	120	122	6.8	31400	Yes	Fast
H	Delco	6000	146	122	125	13.4	31400	Yes	Fast
J	Delco	6000	190	120	125	9.3	31400	Yes	Fast
K	Delco	6000	137	118	125	14.5	31400	Yes	Fast
L	Commercial brass	6000	196	116	120	9.0	31400	Yes	Fast
M	F-1	6000	152	115	122	12.7	31400	Yes	Fast
N	Cold rolled steel[g]	5500[d]	135	116	118	16.6	25500	Yes	Fast
V	Cast iron	2000[d]	115	120	120	53.6	10800	Yes	Fast
W	Cast iron	2500[d,f]	126	116	118	37.8	13500	Yes	Fast

[a] All shafts except where noted were chrome-plated high-carbon steel.
[b] Load limit of test machine was approximately 6000 psi.
[c] Highest steady-state temperature prior to seizure.
[d] Bearing seized at this load.
[e] Run terminated due to accidental failure of power to oil-supply pump.
[f] Incipient seizure at 1521 psi.
[g] Copper-plated high-carbon-steel shaft, support bearings—Delco.

lacquer, it was decided to try to produce artificially such films on bearings and investigate the resulting change in the break-in and load-carrying characteristics of the bearings lubricated with silicone fluids.

Silicone Lacquers and their Use in Silicone-Fluid-Lubricated Systems

Previous research of this laboratory on the oxidation stability of the dimethyl silicone polymer fluids had shown them to be very stable at temperatures up to approximately 300 F. At higher temperatures some formaldehyde and formic acid were evolved, and eventually the oil thickened until it set to form a water-white transparent gel. The rate of oxidation and of viscosity increase was considered negligible for most purposes at temperatures of 300 F, but at higher temperatures they became increasingly more serious. In the temperature range of 300 to 500 F and in the presence of the common metals, Cu, Fe, Al, bronze and babbitt, the rate of oxidation was decreased somewhat. However, pure lead and pure tin were found to be catalysts for accelerating the oxidation and gelation of the oil. From this work and other related research of this laboratory it was concluded that the films on the bearings, as described previously, were silicon-containing organic lacquers or thin films formed during contact of the metal with the products of the silicone-oil oxidation reaction.

The test bearing metals and also the supporting bearings were immersed in beakers of the silicone oil and were heated in the open air for long periods of time at 300 F or higher. The desired lacquers eventually formed, the time required varying with the temperature and metal. Numerous variations of this method of forming lacquers on the bearings have been tried and developed. The method used in preparing the bearings throughout the test discussed in this report involved the use of temperatures of 300 F for as long as 100 hr, or 500 F for only 24 hr. Longer times of exposure were necessary for steel than for nonferrous metals.

With the exception of run F, all of the runs E to N, inclusive, in Table 2 were made with lacquer-coated bearings previously prepared by the chemical method described. After the lacquer treatment the bearings were allowed to cool, were washed with benzene, and were then assembled in the test machine. Following this, the test fluid was added and the bearing test was run according to the loading schedule indicated in Table 3. It will be noted from Table 2 that all these treated bearings carried the maximum load of the machine (6000 psi), while the break-in time was reduced to only 2 hr. Also no significant changes (± 1 per cent or less) in the viscosity of the fluid resulted.

In run F an untreated Delco bearing was used, and it was initially loaded up to and held at 500 psi. After $1^1/_4$ hr operation at that load seizure occurred. This is to be contrasted with the five equally or more rapidly loaded runs, E, G, H, J, and K, on the same bearing metals up to maximum loads of 6000 psi. The low load seizure in run F indicates the need for a slow break-in to obtain load-carrying capacities in excess of 1000 psi. It can now be concluded that a silicone film or lacquer formation on the bearings is necessary for the operation at high unit loads of silicone-fluid-lubricated journal bearings.

In runs L and M on commercial brass and F-1 (high tin-bronze) bearings, respectively, each was chemically treated by the method described, and each could be rapidly and successfully loaded to the maximum load-carrying capacity of the machine. In run N a copper-plated high-carbon-steel journal, rotated in a previously lacquered, cold-rolled steel, test bearing, operated successfully under rapid loading until seizure occurred at 5500 psi. This is to be compared with the very low loads (250 psi) at which seizure occurred with steel-on-cast-iron silicone-lubricated bearings as described in the earlier tests.

Runs V and W were made using lacquer-treated bearings of cast iron and journals of chromium-plated high-carbon steel. Seizures occurred at loads of approximately 2000 and 2500 psi. In another paper,[7] it is shown that Pesco high-pressure gear pumps, fitted with cast-iron bushings, failed in a few minutes at a hydraulic pressure of 600 psi. Therefore, although the silicone-lacquer treatment did not increase the load-carrying capacity of cast-iron bearings to values as high as it did the nonferrous bearings studied, a considerable and perhaps valuable increase did result. This increase in ability to carry loads is especially significant when one considers the very rapid loading schedule followed (see Table 3).

Silicone-Lacquered Bearings and Their Use With Petroleum Oils

The advantages found in silicone-fluid systems in treating bearings with silicone lacquer suggested the investigation of the value of using such treated bearings in petroleum-oil-lubricated systems. In runs P, Q, R, S, and T (see Tables 4 and 5), Navy Symbol 1080 petroleum oil was used for the bearing tests. This is a high V. I. nonadditive aviation-engine lubricant, and the batch used had the following viscosity-temperature characteristics:

15.5 centipoises at 210 F
74.5 centipoises at 130 F
360.0 centipoises at 77 F
2700.0 centipoises at 32 F

The viscosity is identical with that of the silicone fluid used at 148 F, which is not far from a mean of the bearing temperatures encountered.

Before beginning runs P and Q, the bearings were lacquered with a silicone oil by the chemical methods already described. Runs R, S, and T were made without any preliminary chemical treatment of the bearings. It was found that the lacquered

TABLE 3 TREATMENT AND LOADING SCHEDULE AND FOR JOURNAL-BEARING TESTS WITH SILICONE FLUID

Run no.	Bearing material	Lacquer treatment	Loading schedule	
A	F-16	None	500 psi for $15^1/_2$ hr	100 psi increments in load 1 hr after each equilibrium temperature was reached
B	Aluminum	None	500 psi for 17 hr	Same loading as in A
C	Alfin	None	Same as in A	
D	Copper	None	500 psi for 19 hr	Same loading as in A
E	Delco	Bearings cooked in silicone oil 150 hr at 300 F	500 psi for 2 hr	250 psi increments in load every 15 min
F	Delco	None	500 psi load	Bearing seized after $1^1/_4$ hr
G	Delco	Bearings cooked in silicone oil 24 hr at 430 F	500 psi for $2^1/_4$ hr	500 psi load increments every 15 min
H	Delco	Bearing cooked in silicone oil 24 hr at 380 F	500 psi for 2 hr	3000 psi load increments every 15 min
J	Delco	Bearing cooked in silicone oil 24 hr at 380 F	Same as run H	
K	Delco	Bearings cooked in silicone oil 44 hr at 300 F	500 psi for 2 hr	6065 psi added immediately after break-in
L	Commercial brass	Cooked in silicone oil 89 hr at 300 F	500 psi for 2 hr	Load increase same as run G
M	F-1	Cooked in silicone oil 108 hr at 300 F	500 psi for $2^1/_2$ hr	Load increase same as in run G
N	Cold-rolled steel	Shaft and bearing cooked in silicone oil Bearings—137 hr at 300 F Shaft—168 hr at 300 F	Same as in run E	
V	Cast iron	Shaft and bearings cooked in silicone oil Bearings—178 hr at 300 F Shaft—167 hr at 300 F	500 psi for $2^3/_4$ hr	Same load increase as in run G
W	Cast iron	Bearings cooked in silicone oil 8.5 hr at 500 F	Same as in run G	

TABLE 4 RESULTS OF JOURNAL-BEARING TESTS USING NAVY SYMBOL 1080 PETROLEUM OIL

Run no.	Bearing material[a]	Bearing load, psi	Bearing[b] temp, deg F	Oil-sump temp, deg F Initial	Final	ZN/P	PV	Treated bearing	Type of break-in
P	Commercial brass	5800[c]	158	110	112	10.0	31000	Yes	Fast
Q	Delco	4300[c]	147	101	102	18.3	22900	Yes	Fast
R	Delco	2100[c]	126	127	129	58.8	11300	No	Slow
S	Delco	2600[c]	143	141	124	30.6	13900	No	Fast
T	Delco	2300[c]	136	136	136	40.7	12600	No	Fast

[a] All shafts were chrome-plated high-carbon steel.
[b] Highest steady-state temperature prior to seizure.
[c] Seizure load.

TABLE 5 TREATMENT AND LOADING SCHEDULE FOR JOURNAL-BEARING TESTS WITH N.S. 1080 PETROLEUM OIL

Run no.	Bearing material	Lacquer treatment	Loading schedule	
P	Commercial brass	Bearing cooked in silicone fluid 96 hr at 300 C	72 psi for 2 hr	250 psi load increments every 15 min
Q	Delco	Bearing cooked in silicone fluid 118 hr at 300 F	500 psi for 2 hr	Load increase same as in run P
R	Delco	None	72 psi for 16 hr	Load increments of 250 psi 1 hr after each equilibrium temperature was reached
S	Delco	None	72 psi for 4 hr	Load increased 250 psi every 30 min
T	Delco	None	72 psi for $2^1/_3$ hr	Load increased as in run S

bearings did not seize during a fast break-in until loads were used of 5800 psi with brass and 4300 psi with Delco bearings, respectively. In contrast, the unlacquered Delco bearings seized at from 2100 to 2600 psi, depending upon the loading schedule followed during the break-in. Therefore, with a petroleum-oil-lubricated system, the silicone-lacquer bearing treatment was found to increase considerably the permissible rate of break-in and the seizure load.

Discussion of Experiments

Due to the care used in boring, aligning, and finishing the bearings and journal, reasonably good reproducibility was obtained, and seizure loads did not vary more than approximately 10 per cent from the mean. In this work high enough values of ZN/P were selected to remain usually in the region of hydrodynamic lubrication. It is of immediate interest to learn how much above 6000 psi are the seizure loads for the various nonferrous metals studied. Inasmuch as increases in seizure loads with lacquer formation of from 100 to 200 per cent were found, there is no doubt of the reality of the improvement obtained with lacquering.

Considerable variations were found in the maximum bearing temperatures before seizure. However, this is not unexpected because of the wide range of loading schedules followed during the break-in operations. These temperature variations are not considered as important in silicone-fluid as in petroleum-fluid-lubricated systems, due to both the smaller temperature coefficient of viscosity and the greater oxidation stability of the silicone fluid. As in the hydraulic work,[7] no significant viscosity changes occurred during the bearing runs not accounted for by initial loss of volatiles. This eliminates shear instability of the fluid as a cause of the peculiar lubricating properties of steel-on-steel or cast-iron bearings.

Conclusions

A number of the common bearing metals have been found very promising for use in unilaterally loaded journal bearings, lubricated with the silicone fluid. Among them are copper-lead, bronze, commercial brass, babbitt, copper, aluminum (17S), and Alfin alloy. These were especially effective with chromium-plated high-carbon-steel journals. Cold-rolled steel bearings operated with copper-plated high-carbon-steel journal bearings also showed promise. These nonferrous bearings have been successfully operated at the load limit of the bearing machine used (6000 psi), with load increments of as much as 5500 psi applied during cycling tests.

For the maximum load-carrying capacity the bearings should have either a long gradual break-in or be suitably treated with a silicone lacquer prior to operation. The long, slow break-in process formed a silicon-containing organic film on the bearing. It was found that such film-coated bearings had load-carrying capacities comparable to those of artificially silicone-lacquered bearings.

Bearing systems of steel on steel and steel on cast iron were not satisfactory for operation with silicone fluids. However, a very slow break-in process or a preliminary treatment of the bearings with silicone lacquer enabled them to stand loads of up to approximately 2000 psi. Suitable bearing-lacquer coatings have been formed by completely immersing the bearings in dimethyl silicone polymer fluid (viscosity at room temperature of 50 to 100 centipoises or more may be used) and maintaining them at a temperature of from 300 to 500 F in the presence of air. The time of exposure varied with the metal, being longest for steel and least for copper. Exposures of from 100 hr were needed at 300 F and 24 hr at 500 F.

Acknowledgments

The skillful mechanical and experimental work of John Larson MM 2/c and the able machine work of William Kostowicz MM 2/c were of considerable aid in this work. Their interest and suggestions are much appreciated. Analyses of the test oils before and after the bearing runs were made through the courtesy of Carl Hechmer, Lieutenant, U.S.N.R., and Eleanor Bried, Sp (X) 2/c. Chemical analyses of the bearings listed in Table 1 were made through the co-operation of the Analytical Section of the Metallurgy Division.

Dimethyl-Silicone-Polymer Fluids and Their Performance Characteristics in Hydraulic Systems

By V. G. FITZSIMMONS,[1] D. L. PICKETT,[2] R. O. MILITZ,[3] and W. A. ZISMAN[4]

This investigation concerns research on the polymethyl siloxanes or dimethyl-silicone polymers, and their uses as lubricants and hydraulic fluids. It is shown that these new fluids give excellent results when operated in a standard aircraft-type Pesco gear pump. It was found that the silicones when used in a gear pump fitted with cast-iron bushings caused serious wear and disintegration of the bushings at pump pressures of only 600 psi. It is concluded that the loaded surfaces of steel sliding on cast iron or on steel are not adequately lubricated by the silicone fluid, whereas steel sliding on bronze is effectively lubricated. A simple manually operated slider-and-plate test method was discovered capable of permitting the discrimination between good, bad, and indifferent bearing combinations for use with silicone fluids. Results with 156 combinations of metals are summarized. There is also given a means of relating these observations to the practical problem of selecting suitable metals for use in lubricating the sliding parts of machinery with silicones. The results are related to observations on the choice of suitable metals for gear pumps, piston pumps, and journal bearings. It is shown that the same conclusions relative to steel sliding on steel or on bronze are reached from tests with the Vickers piston pump.

Introduction

EARLY in the war the Naval Research Laboratory was requested by the Bureau of Ordnance to investigate the possibility of using the new silicone-polymer fluids (or polyorganosiloxanes) as recoil and damping fluids. When difficulties due to the inflammability of present aircraft hydraulic fluids became apparent, the Bureau of Aeronautics also became interested in the development of these fluids for use in aircraft hydraulic systems. A description of some of the properties of these fluids by the two producers will be found in the literature.[5,6]

Many of the physical and chemical properties of interest in the evaluation of hydraulic fluids were investigated by this laboratory prior to commencing tests in hydraulic equipment. It was concluded that of the various silicone-polymer fluids, the dimethyl silicones were the most desirable because they had the smallest temperature coefficients of viscosity of any pure fluids known, and also had unusual stability to oxidation. The resistance of the dimethyl-silicone fluids to various fire hazards was studied, and it was concluded that they were adequately resistant to the fire hazards of interest in this investigation. Laboratory tests and M-1 incendiary-bullet firing tests on properly stripped silicones verified that, under such fire hazards, these fluids would not ignite nor would the liquid spray propagate a flame front. This made it of interest to study the operation in hydraulic equipment of a carefully stripped silicone fluid.

Both manufacturers of the silicones were requested to prepare one sample each of the fluid having a viscosity at 210 F of approximately 25 to 30 centistokes. The viscosity-temperature characteristics obtained in each case were within the following narrow ranges:

Temperature, deg F	Viscosity, centistokes
210	26–30
130	52–56
100	68–74
0	259–275
—20	385–390
—40	600–610

The two fluids submitted were used throughout this investigation and no important differences in behavior were found. Other less viscous fluids can be obtained having the advantage of being much less viscous at lower temperatures. Thus a silicone fluid has since been obtained having the advantages that, while the viscosity at 210 F is 18 centistokes at 0 F it is 160 centistokes, and at —40 F it is only 360 centistokes. Such a single hydraulic fluid can at once satisfy all present military requirements in so far as viscosity, volatility, and pour point are concerned. The high viscosity at 100 F of the fluid used here was considered desirable at the commencement of this investigation because the tendency of the silicones to leak out of the hydraulic system and the inflammability were decreased by using the highest possible viscosity consistent with approximating both the low-temperature ordnance and aircraft hydraulic specifications, O.S. 2943 and AN-VV-O-366b.

Experimental Equipment

At the commencement of this investigation the available supply of silicone fluids was very limited and hence the laboratory hydraulic tests were made in systems operated with the small-size high-pressure pumps used in aircraft. These were the 1P-349-N Pesco gear pump and the Vickers model No. PF-2713-10, constant-delivery aircraft piston pump.

The Pesco pump has nitrided gears running in high-lead bronze bushings. The bushings are pressure-loaded against the sides of the gears. Any wear resulting from a test is therefore noticed on the face of the gears and the sliding surface between the thrust face of the bushings and the side of the gears.

The Vickers pump is a 7-piston constant-delivery pump. Several points are observed for wear, one being the hardened-steel

[1] Assistant Technologist, Lubrication Section, Chemistry Division, Naval Research Laboratory.

[2] Associate Technologist, Lubrication Section, Chemistry Division, Naval Research Laboratory.

[3] Lieutenant, U.S.N.R., Lubrication Section, Chemistry Division, Naval Research Laboratory.

[4] Head, Lubrication Section, Chemistry Division, Naval Research Laboratory. Mem. A.S.M.E.

[5] "The Silicones—A New Plastics Family," *Plastics*, January, 1945.

[6] "The Organo-Silicon Polymers," by E. G. Rochow, *Chemical and Engineering News*, vol. 23, 1945, p. 612.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Navy Department or of this Society.

universal link on which are pivoted, on steel pins, four hardened-steel knuckles. Through two steel retainers this assembly serves to connect the drive shaft to the bronze cylinder block. As a result of the angle (10 deg) between the drive shaft and cylinder block, each revolution of the shaft causes the knuckles to rotate 10 deg around the steel pins of the universal link. Bronze bearings at either end of the universal link are lubricated by the fluid pumped being forced through a small hole in the link. The action of the piston in the cylinder block and the sliding of the cylinder block on the valve plate are also observed. Five ball bearings, one thrust loaded, two radial loaded, and two with a combination load, serve to make this pump very desirable as almost every type of wear is represented.

The hydraulic system used is shown in Fig. 1. The reservoir A was designed to produce a long liquid path within the sump from inlet to outlet in order to facilitate deaeration of the fluid. Two sizes of sumps were used, depending upon the amount of the test fluid available. When the smaller was used the total capacity of the hydraulic system was 1.25 gal, while with the larger sump it was 3 gal. Thermometer T was used for reservoir temperature measurements. The line C was $^{5}/_{8}$ in. OD 0.035-in-wall copper tubing. Line D was a standard flexible high-pressure rubber hose with $^{13}/_{32}$-in. ID conforming to the specification AN-863-8-21. All other tubing was $^{1}/_{2}$-in. OD and 0.035-in. wall copper. Standard 24S-T aluminum alloy or equivalent AN-type flare fittings were used throughout. The driving motor used, M, was a General Electric type K, frame 225, 220/440-volts, 3-phase, rated at 5 hp at 3600 rpm. Pressure was controlled by a Vickers No. C-167-E relief valve H. The gage I was used on the high-pressure side of the valve. This was a Lonergan gage with a range up to 3000 psi. Gage J was used to measure the pressure drop across the relief valve, and gage K measured the pressure drop across the filter L. This filter was either a Purolator aircraft-line type, using a paper element impregnated with what is reported to be a phenol-formaldehyde resin, or a Skinner aircraft-instrument-type filter, using a paper-disk element enclosed in a cloth sheath. The hydraulic fluid was cooled in flowing through cooler M_1, which consisted of 20 ft of $^{3}/_{8}$-in. copper tubing inserted in 20 ft of $^{3}/_{4}$-in. copper tubing and then coiled in a helix. The test fluid passed through the space between the inner and outer tubes, and

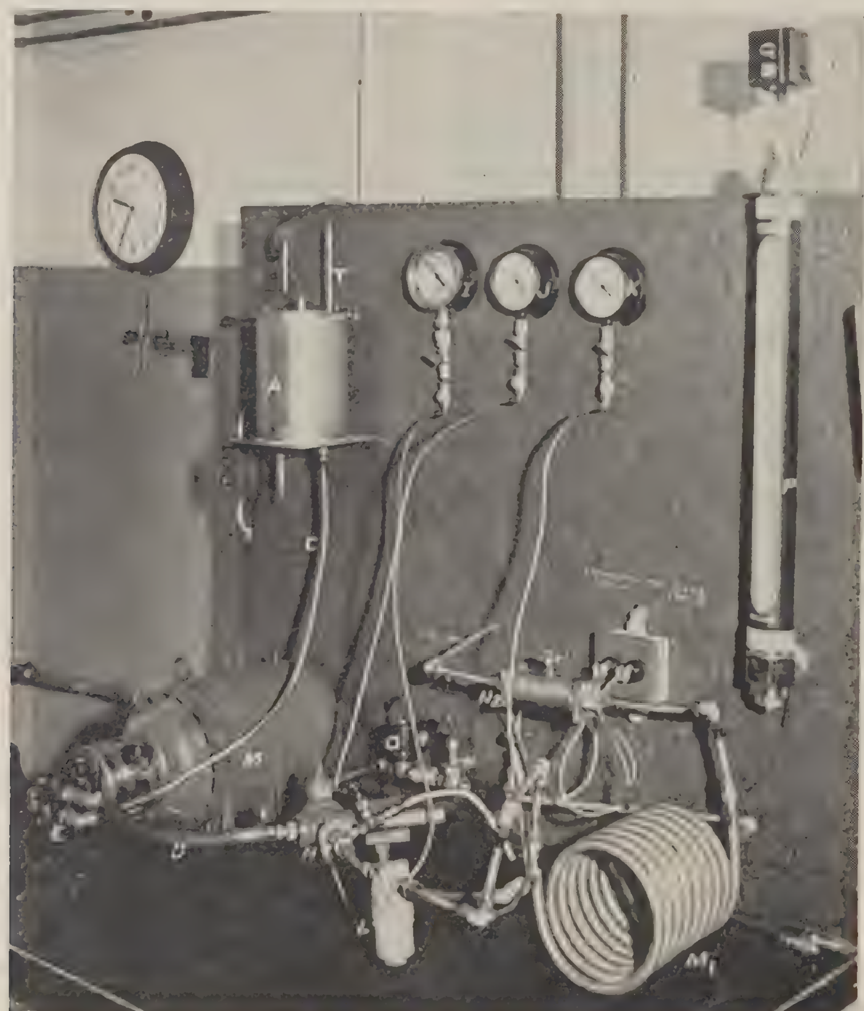


Fig. 1 Hydraulic Test Setup Used

a flow of cooling water passed through the inner $^3/_8$-in. tube. Where necessary the helix was immersed in a cooling bath.

The device N_1 was a Fenwal thermoswitch which was connected with the motor control, the purpose being to shut off the driving motor if the temperature of the hydraulic fluid exceeded the desired test temperature. This device was in the nature of a safety shutoff in case of water failure to the cooler. The device N_2 was also a Fenwal thermoswitch which controlled the sump oil temperature through a solenoid O in the water-supply line to the cooler. The test fluid then passed through P, a Fisher and Porter rotameter Stabl-vis type 12 P7, capable of measuring flow rates from 0 to 5 gpm. On top of the rotameter a Minneapolis-Honeywell Pressuretrol Q was installed. This was a safety device to switch off the power to the drive motor if for any reason the pump failed to deliver fluid. It was caused to operate by the slight pressure in the line from the rotameter to the reservoir which was generally in the neighborhood of 1 or 2 psi. When this pressure was not present the switch moved to the "off" position, thereby breaking the circuit to the drive motor. The fluid left the rotameter through a line which entered the side of the reservoir near the top and then directed the flow against the wall of the container. This caused the test fluid to take a circular path to the pump inlet, thereby allowing any entrapped air bubbles the greatest possible time to escape to the surface of the fluid. The liquid velocity was not sufficient to cause the fluid to vortex in the reservoir.

Operating Technique

Before each test the entire hydraulic system was dismantled and cleaned with suitable solvents. Repeated washing with unleaded gasoline was sufficient to clean the system if previously operated with the silicone fluid. Where polymer-thickened petroleum hydraulic fluids had been used previously, it was necessary to wash all parts in unleaded gasoline to remove the petroleum fluid and to follow that with successive washings with methyl cellosolve and acetone. The cleaning procedure was completed with drying the system with a forced draft of clean air. In using these fluids precautions were necessary owing to the pronounced ability of the dimethyl silicones to creep over surfaces and to leak through mechanical joints. All tubing flares had to be made as nearly perfect and tight as possible, and it was found valuable to use wherever practicable a pipe-thread sealing compound. An available material which was found satisfactory was "Tite-seal," a product of the Radiator Specialty Company of Charlotte, N. C.

Before each run and before a gear pump was placed on the test stand, it was fitted with a new set of bushings and gears. Both the gears and the bushings were thoroughly washed in benzene, dried, and weighed to the nearest tenth of a milligram on an analytical balance. As the pump was assembled, all moving parts and all O-ring seals were wetted with the fluid to be tested. When the system was assembled and secured, it was charged by filling the reservoir and turning the pump slowly until the entire system was filled. The drive motor was then started and the fluid circulated with no restriction offered by the Vickers relief valve until it could be ascertained that the system was operating satisfactorily. The Vickers relief valve was then adjusted to the desired pressure and as the fluid began to heat up, the adjustments on the thermoswitches were made.

Most of the runs were made for 100 hr of continuous operation or until a failure caused discontinuance. Samples of the hydraulic fluid were taken at the end of each run and the viscosity, neutralization number, and precipitation number were measured, the latter two measurements being made only when the reference petroleum fluids were used. The gears and bushings were removed from the pump, washed, and weighed at the end of each run to determine the amount of wear occurring during the test period. The rest of the system was examined for any unusual manifestations such as corrosion, solvent effects, and deposits.

Piston-pump performance was characterized by observations of the flow rate at the beginning and end of each run, appearance of metallic particles in the test fluid or the filter, and condition of the working parts, as evidenced by weight losses, appearance, and fits. Due to the size and mass of the working components as well as to the presence of many crevices impossible to clean perfectly, measurements of weight losses were not considered satisfactory for observing wear.

A new filter core was used for each run and in some cases where a Skinner filter was used, a cloth bag covered the core. After each run the filters were examined for deposits. In some of the runs thermocouple readings were taken to determine temperatures after the fluid had passed through the pump and again after passing through the relief valve. These thermocouple installations were temporary in that the temperature was found to rise 10 deg F passing through the valve. The results were consistent for a number of runs, thus making such observations superfluous and hence they were not made routine observations. The approximate number of cycles which the fluid made per run was calculated by multiplying the time of the run by the flow in gallons per minute and then dividing by the system capacity. The flowmeter readings were not corrected for density or viscosity of the fluid. These readings were all made at running temperature for the purpose of determining losses in volumetric efficiency of the pumps at a determined pump speed. These readings were approximately 4 per cent higher than the actual flow rates.

The reference petroleum hydraulic fluids tested had the following viscosity-temperature characteristics:

Temperature, deg F	Viscosity, centistokes AN-VV-O-366b	O.S. 2943
210	5.16	10.0
130	10.1	...
100	14.1	29.1
0	90.3	223.5
—25	...	556.3
—40	406.4	...

Packings for Systems Operated With Dimethyl-Silicone-Polymer Fluids

If the dimethyl-silicone-polymer fluids are to be used in hydraulic systems, it may be necessary to change the composition of the packings. It was found essential to change the composition of the O-ring seals which are so commonly used especially in aircraft. Whatever the ingredients used as plasticizers for the synthetic rubber of the O-ring seals, these plasticizers gradually were extracted by the silicone fluids, and as a result the rings shrank and became brittle. This shrinkage was of such magnitude as to cause serious leakage from the pumps and other hydraulic components using such rings. As might be expected, the solubility of the plasticizers was increased with time and temperature. It was found that serious leakage occurred after 24 hr of pump operation at 180 F sump temperature. However, it was found possible to run the pump for 500 hr continuously at 140 F without serious leakage, but after that run had been completed the seals had become so brittle that they shattered when an attempt was made to remove them. Even in systems operating at lower temperatures than 140 F, there may be trouble with present O-ring packings after long exposures occurring in peacetime practice or in long storage. There was evidence that the solubility effect was present even at room temperature.

A solution to the problem of the shrinkage of O-ring seals was found. The same solution will probably care for difficulties with

many other types of hydraulic packings. It seems that the most commonly used plasticizers of synthetic rubber are all soluble to some extent in dimethyl-silicone fluids. Therefore the High Polymer Section of this laboratory compounded a new packing with a small amount of the silicone hydraulic fluid under test. The rubber used was basically a Hycar compound having a durometer reading of 60 and a tensile strength of 2000 psi, an elongation of 325 per cent, and a freezing point of —60 F. A small percentage of graphite was also included in the manufacture of this material to reduce the friction of these seals on moving parts. It was found that O-rings made from this stock gave completely satisfactory performance in pumps using silicone fluids for 100 hr continuous running at from 180 to 200 F, the highest temperatures employed in this investigation. It was found later that suitable gaskets necessary for journal-bearing experiments[7] could be made from the same material.

Behavior of Silicone Fluids in Gear Pumps

The results obtained on the reference petroleum oils and on the silicone fluid are given in Tables 1 and 2. The sometimes variable finish and hardness of the gears and bushings furnished by the manufacturers of the pumps caused irregular weight measurements during the break-in of new gears as can be seen when comparing the gear weight losses on all runs, especially runs P-20 and P-21 of Table 2. It can be noticed from runs P-19, P-20, and P-21 that when the same set of gears was re-used the weight losses became less, thus indicating that at least part of the weight loss was not due to the fluid alone but rather to mechanical variables. These three runs are the only ones in which gears were re-used, and so it can be concluded that the weight losses in the other runs represent the highest wear rates to be expected in the history of a given pump operated on clean oil.

It was noticed that when using the petroleum "ordnance hydraulic fluid" O.S. 2943, the weight losses were much greater than when the silicone fluid was used. Values of approximately 0.015 g per gear per 100 hr, using the new gears and bushings with petroleum fluid, should be compared with values of often much less than 0.009 g using the silicone fluid. It was also found that higher rates of bushing wear occurred when petroleum was used than with silicone.

It is believed the cause of the low wear rates on the silicone-lubricated pressure faces of the bronze bushings was the formation of a protective resin consisting of cross-linked polymers developed by oxidation of the fluid. Early research on the oxidation of silicones had shown that gels and eventually cross-linked polysiloxane resins resulted. Similar deposits were obtained by this laboratory when strips of polished metal were immersed in the silicone fluid during the oxidation tests.

Although it was found difficult to detect the presence of gel in the silicone fluid because of the small difference in the refractive indexes of gel and fluid, careful examination under a microscope definitely revealed the presence of particles of gel in the fluid

TABLE 1 GEAR-PUMP RUNS ON ORDNANCE AND AERONAUTICAL HYDRAULIC PETROLEUM OILS

	AN-VV-0-366b		O.S. 2943 (PRL 1081)	
	P-60	P-63	P-29	P-31
Time in hrs.	100	100	100	100
Sump temp. °F	180	180	140	140
PSI high	1500	1500	1500	1500
PSI low	38-40	37-45	65-40	65-45
Flow GPM:				
Start	3.08	2.97	3.03	3.01
Finish	3.04	2.92	3.04	3.04
Cycles approx.	18,000	16,000	18,180	18,120
Wt. loss grams:				
Drive gear	.0439	.0535	.0115	.0075
Driven gear	.0372	.0592	.0178	.0052
Max. one bushing	.1782	.0102	.0574	.0044
Total bushings	.4452	.0204	.1873	.0079
Visc. cs.:				
Start	13.37	13.37	28.39	29.10
Finish	7.884	8.485	12.70	14.45
Filter type	Purolator Line type	Purolator Line type	Purolator Line type	Purolator Line type
Appearance of oil after run	--	--	Normal, clear	Normal, clear
Gears & bushings	New	Broken in 100 hrs.	New	New

TABLE 2 GEAR-PUMP RUNS ON DIMETHYL-SILICONE-POLYMER FLUID

	Run No.								
	P-11	P-13	P-18	P-19	P-20	P-21	P-22	P-23	P-25
Time in hours	48	50	41	100	100	100	100	100	500
Sump temp. °F	180	180	180	180	180	180	200	200	140
PSI high	1500	1500	1500	1000	1500	1500	1500	1500	1500
PSI low	135	110	150	150	175	140-155	160	165-195	210-300
Flow GPM:									
Start	3.24	3.25	3.35	3.24	3.34	3.35	3.24	3.36	3.51
Finish	3.27	3.35	3.24	3.24	3.34	3.34	3.12	3.35	3.43
Cycles approx.	9,360	9,750	8,118	19,440	20,040	20,040	19,080	20,100	105,000
Wt. loss grams:									
Drive gear	.0066	--	.0117	.0034	.0048	.0006	.0092	.0044	.0030
Driven gear	.0045	--	.0072	.0000	.0020	.0011	.0041	.0000	.0028
Max. one bushing	.0034	--	.0068	.0081	.0010	.0010	.0086	+.0004	.0168
Total bushings	.0042	--	.0133	.0171	.0024	.0020	.0137	+.0010	.0208
Visc. cs.:									
Start	--	69.35	69.27	69.27	69.27	71.11	69.27	71.11	71.11
Finish	--	68.51	68.23	68.58	69.23	72.05	70.39	72.34	70.16
Filter type	Skinner no bag	Purolator line type	Skinner with bag	Skinner with bag	Skinner no bag*	Skinner with bag	Skinner no bag**	Skinner with bag	Skinner no bag
Appearance of oil after run	Cloudy gel	Light amber, clear	Hazy	Hazy	Hazy	Clear, light amber	Hazy amber	Clear amber	Clear deep yellow
Gears & bushings	New	New	New	New	From P-19	From P-19&P-20	New	New	New

* After 40 hours.
** After 60 hours.

[7] "Dimethyl-Silicone-Polymer Fluids and Their Performance Characteristics in Unilaterally Loaded Journal Bearings," by J. E. Brophy, R. O. Militz, and W. A. Zisman, published on pages 355–360 of this issue of the Transactions.

which had been run for 100 hr. In every run some soft black gel was found on the filter element. In the 500-hr run, P-25, there was enough gel formed in the fluid so that it could be scraped from the filter walls with a spatula, permitting the recovery of 10 ml. Apparently the only reason for the fluid casting out the gel under normal conditions was that the gel would attach itself to minute particles of more dense material in the fluid and would thereby become heavy enough to be thrown out at points where flow direction changed abruptly such as happened in the filter.

Some attempts were made to concentrate the gel formed by the use of a centrifuge but this method proved unsatisfactory. As a result of the formation of gel in the silicone fluid, it was found that a definite build-up of back pressure occurred on the Skinner filters fitted with bags, accompanied by a gradual clogging of the pores by slime. The rate of clogging, however, was not sufficient to cause trouble in normal operation. Using either the purolator filter or the Skinner filter without a bag, serious clogging of the filter occurred after 500 hr of continuous operation at 140 F and 1500 psi operating pressure.

It will be noticed in Table 2 that there was no significant viscosity change (less than 2 per cent) in the silicone fluid after having been pumped at a pressure of 1500 psi continuously for as long as 500 hr (or 105,000 cycles). For comparison, the petroleum fluid O.S. 2943 decreased in viscosity to less than 50 per cent of its original value after 18,000 cycles.

A set of cast-iron bushings (3.3 per cent carbon content) was made to fit the Pesco 1P-349-N pump to replace the usual bronze bushings. Results with O.S. 1113 petroleum hydraulic oil as well as the silicone fluid are given in Table 3. When operated on

TABLE 3 GEAR-PUMP RUNS USING CAST-IRON BUSHINGS

	Hydraulic fluid			
	Silicone		OS-1113 petroleum oil (commercial)	
Run No.:	P-14	P-14A	P-14B	P-14C
Time in hours	0.25	0.33	2.0	1.0
Sump temp. °F	180	180	180	180
PSI high	600 (failed)	600 (failed)	800 (satisfactory	800 (satisfactory
PSI low	110	110	110	110
Flow GPM:				
Start	3.80	--	3.60	3.60
Finish	2.20	--	3.60	3.60
Cycles approx.	75	--	--	--
Filter type	Purolator line type	Purolator line type	Purolator line type	Purolator line type
Appearance of oil after run	Black with iron particles	Black with iron particles	Clear	Clear
Gears and bushings	New	New	New	New

silicone fluid, the pump ran normally at 200 psi for 15 min. Therefore the pressure was increased gradually. At 600 psi the pump began to fail rapidly and the fluid became black with such finely dispersed particles of iron that they were not retained by the filter, while the volumetric efficiency of the pump fell approximately 33 per cent in 5 min, leading to the discontinuance of the entire run. A microscopic examination revealed that much gel had formed in the silicone fluid. Several more runs were made using new cast-iron bushings and pump failures were obtained. The roughened appearance of a set of the cast-iron bushings following such a disastrous run is evident in Fig. 2.

Finally, similar runs were made using the Navy Ordnance petroleum hydraulic fluid O.S. 1113. It was found that the pump operated well at a pressure of 800 psi. When examined after 2 hr of operation, there was only a slightly galled spot on the bushings as evidence of wear.

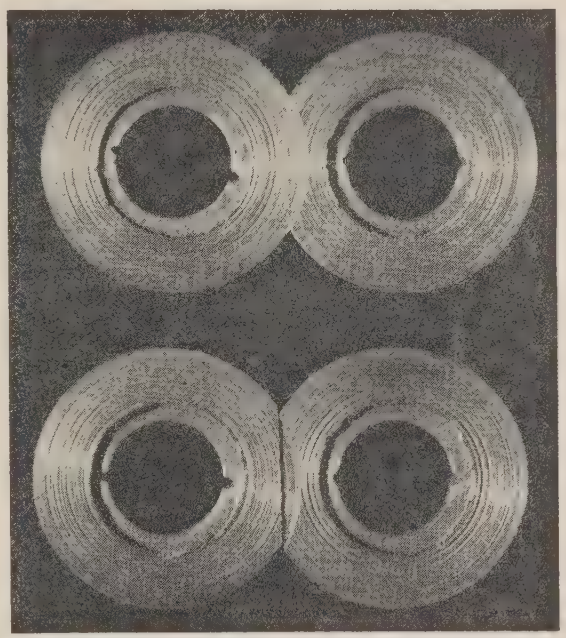


FIG. 2 CAST-IRON BUSHINGS AFTER 0.25 HR IN SILICONE

SELECTION OF BEARING METALS FOR USE WITH SILICONE LUBRICANTS

The contrast in the high rate of wear of silicone-lubricated surfaces of steel rubbing on cast iron, and the negligible rate of wear of steel on bronze in the gear-pump runs described, suggested the possibility that other combinations of rubbing metal surfaces might exist which were also well lubricated by this fluid. It was evident that an undesirably large amount of shop and time-consuming testing machine work would be needed if the large number of metal combinations of interest were tried as pump bushings or test bearings.

In endeavoring to duplicate the serious rate of wear and eventual seizure at low pressures of steel sliding on polished cast iron or on polished steel, a simple technique was found valuable in the rapid sorting out of the good, bad, and indifferent metal combinations lubricated with the silicone fluid. This "slider-and-plate" method consisted in sliding manually a flat cylindrical and polished metal slider having a diameter of 1 in. upon a lubricant-covered, flat, polished metal plate $1^1/_2$ in. wide and 6 in. long while exerting a downward pressure on the slider. The edge of the cylinder was rounded off to make the cylinder operate as a sled and so invite formation of an oil wedge between slider and plate.

In the middle portion of each long back-and-forth stroke of the slider a condition of thick-film lubrication prevailed unless the unit load had become high enough to cause metal-to-metal contact. The maximum load obtainable by exerting pressure manually was approximately 20 psi with the 1-in. slider. Loads of around 40 psi could be obtained using sliders of smaller diameter, although the need for preventing rocking did not permit the use of sliders much smaller than $^5/_8$ in. diam.

The existence of serious wear was determined by the sense of touch, by an inspection of the sliding metal surfaces, by observations of the fluid for deterioration, and especially for the presence of abraded metal. This method may not be equally useful, however, in examining fluids of entirely different chemical compositions.

A series of these simple tests was made, using thirteen different sliders and twelve different plates. The metals used included bronze (80-10-10), brass (commercial), copper, aluminum, magnesium, tin, babbitt (lead-base), cold-rolled steel, stainless steel, and steel plated with nickel, chromium, cadmium, zinc, and silver. This permitted observing 156 combinations of bearing metals. For comparison, a sample of naval specification petroleum oil NS-1047 was used. This high V.I. nonadditive oil had viscosities of 6.3 centistokes at 210 F, 41 centistokes at 100 F, and 2300 centistokes at 0 F.

It was chosen because at the test temperature (68 F), it had the same viscosity as the silicone fluid used. The plated metal surfaces were buffed to a high polish before use in these tests. The other metals were polished, using successively finer grades of emery paper and ending with 4/0 grade paper. All surfaces were previously cleaned with hot c.p. toluene. Although the degree of surface polish somewhat affected the results of this test in so far as doubtful cases were concerned, it was found that the differentiation between the relatively good and bad cases was not changed.

The results of an attempt to classify the observations using the slider-and-plate tests are presented in Tables 4, 5, and 6. In each table the qualitative results are indicated for the silicone oil and for the comparison oil. From the relative performance of all the combinations of lubricated metals tried, three groups were made. The good metal combinations are listed in Table 4. None of these exhibited any adverse characteristics. The bad combinations are listed in Table 5. The doubtful ones, listed in Table 6, had some bad features, such as relatively high friction or the presence of a metallic sludge in the oil after rubbing, but these characteristics were not considered pronounced enough to classify them as definitely bad.

TABLE 4 GOOD SILICONE-LUBRICATED METAL COMBINATIONS AND CORRESPONDING RATINGS USING PETROLEUM OIL N.S. 1047

Slider	Plate	Corresponding petroleum test result
Brass	Brass	Good
	Nickel	Good
	Cadmium	Good
	Tin	Good
	Zinc	Good
	Aluminum	Doubtful
	Chromium	Good
Bronze	Babbitt	Good
	Nickel	Good
	Copper	Good
	Chromium	Good
	Cadmium	Good
	Zinc	Good
	Aluminum	Good
	Silver	Good
Babbitt	Brass	Good
	Silver	Good
	Nickel	Good
	Chromium	Good
	Cadmium	Doubtful
	Zinc	Good
Nickel	Tin	Good
	Silver	Good
	Chromium	Good
	Aluminum	Good
Zinc	Brass	Good
	Tin	Doubtful
	Nickel	Good
	Cadmium	Good
	Zinc	Good
	Aluminum	Good
Magnesium	Tin	Doubtful
	Nickel	Doubtful
	Chromium	Doubtful
	Cadmium	Doubtful
Aluminum	Brass	Bad
	Copper	Bad
	Magnesium	Bad
	Nickel	Good
	Chromium	Good
	Zinc	Good
Cadmium	Brass	Good
	Copper	Good
	Babbitt	Good
	Tin	Good
	Stainless Steel	Good
	Silver	Good
	Nickel	Good
	Chromium	Good
	Cadmium	Good
	Zinc	Good
	Aluminum	Good
Silver	Brass	Good
	Copper	Doubtful
	Babbitt	Good
	Tin	Doubtful
	Stainless Steel	Good
	Nickel	Good
	Chromium	Good
	Zinc	Good
	Aluminum	Good
Tin	Copper	Good
	Babbitt	Good
	Tin	Doubtful
	Stainless Steel	Good
	Nickel	Good
	Chromium	Good
	Aluminum	Doubtful
Cold rolled Steel	Nickel	Doubtful
	Cadmium	Doubtful
Copper	Aluminum	Good
	Brass	Good
	Copper	Good
	Babbitt	Good
	Magnesium	Good
	Tin	Good
	Silver	Good
	Nickel	Good
	Chromium	Good
	Cadmium	Doubtful
	Zinc	Good

In contradiction to the original conclusion of earlier investigators that silicone fluids were not good lubricants, the results of this study revealed 81 bearing-metal combinations with which they may be found to be useful lubricants. The original reports were based upon the failure of silicone fluids to lubricate steel on steel where petroleum oils behaved very well. It is of interest to point out here that there are cases where the N.S. 1047 petroleum oil behaved poorly, while the silicone fluid produced good lubrication. From Table 4 may be cited such cases as aluminum on brass; copper on magnesium; magnesium on tin, nickel, chromium and cadmium; and cold-rolled steel on nickel and cadmium.

In contrast, the following metal combinations were lubricated well by N.S. 1047 but were not well lubricated with the silicone fluid: Steel, aluminum, copper, and nickel sliding on steel; nickel on brass and on nickel; tin on cadmium, and many others given in Tables 5 and 6.

It is apparent that each particular oil fails to lubricate certain definite metal combinations and that specificity of lubrication is a phenomenon not limited to silicone fluids alone. For example, the difference in the load-carrying capacity between N.S. 1047 oil and white mineral oil or a pure hydrocarbon like dodecylbenzene could be discerned, but in no case was there the serious scratching and seizure encountered with the silicone fluid.

TABLE 5 BAD SILICONE-LUBRICATED METAL COMBINATIONS AND CORRESPONDING RATINGS USING PETROLEUM OIL N.S. 1047

Slider	Plate	Corresponding petroleum test result
Brass	Magnesium	Bad
	Silver	Bad
Tin	Silver	Good
Magnesium	Brass	Doubtful
	Copper	Doubtful
	Babbitt	Doubtful
	Magnesium	Doubtful
	Aluminum	Doubtful
Cold rolled Steel	Aluminum	Good
	Stainless Steel	Good
Copper	Stainless Steel	Good
Aluminum	Tin	Bad
	Silver	Doubtful
	Stainless Steel	Good
Babbitt	Aluminum	Doubtful
Nickel	Copper	Good
	Babbitt	Good
	Cadmium	Good
	Zinc	Good
	Stainless	Good
Chromium	Tin	Doubtful
	Silver	Bad
	Nickel	Bad
	Chromium	Bad
	Cadmium	Bad
	Zinc	Bad
	Babbitt	Bad
	Copper	Bad
	Magnesium	Bad
	Brass	Bad
	Aluminum	Bad
	Stainless Steel	Bad

TABLE 6 DOUBTFUL SILICONE-LUBRICATED METAL COMBINATIONS AND CORRESPONDING RATINGS USING PETROLEUM OIL N.S. 1047

Slider	Plate	Corresponding petroleum test results
Brass	Babbitt	Good
	Copper	Doubtful
	Stainless steel	Doubtful
Babbitt	Magnesium	Good
	Tin	Good
	Stainless steel	Good
	Babbitt	Doubtful
	Copper	Doubtful
Nickel	Brass	Good
	Magnesium	Doubtful
	Nickel	Good
Zinc	Copper	Doubtful
	Babbitt	Doubtful
	Magnesium	Doubtful
	Stainless steel	Doubtful
	Silver	Good
	Chromium	Doubtful
Cadmium	Magnesium	Doubtful
Aluminum	Cadmium	Bad
	Aluminum	Doubtful
	Babbitt	Bad

Slider	Plate	Corresponding petroleum oil results
Bronze	Tin	Good
	Brass	Good
	Magnesium	Good
	Stainless steel	Good
Silver	Magnesium	Doubtful
	Silver	Good
	Cadmium	Doubtful
Tin	Brass	Doubtful
	Magnesium	Doubtful
	Cadmium	Good
	Zinc	Good
Magnesium	Stainless steel	Doubtful
	Silver	Doubtful
	Zinc	Doubtful
Cold rolled steel	Brass	Doubtful
	Copper	Doubtful
	Babbitt	Doubtful
	Magnesium	Doubtful
	Tin	Doubtful
	Silver	Doubtful
	Chromium	Doubtful
	Zinc	Doubtful

In the case of steel on steel, the N.S. 1047 lubricated specimens were given the maximum manual loading (40 psi with a $^5/_8$-in. slider) and still no tendency toward seizure was noticed. Similar plates when lubricated with silicone oils and placed under loads of only 5 psi could be felt to scratch as though an abrasive were present. Slightly higher loads caused deep scratches or even sudden seizures, and with loads of 30 to 40 psi, movement of the slider almost instantly became an impossibility due to the seizure. In spite of such pronounced differences as this, in individual cases with most of the combinations of metals, the silicone fluid and the N.S. 1047 behaved alike. Thus in Table 4, of every five cases where silicones were found to be good, four were also good with the petroleum oil. From Tables 5 and 6, of every five cases in which silicones were found to be bad or doubtful, petroleum was found to give bad or doubtful results in three.

A qualitative relation can be established between the predictions from these simple tests and the results of tests with other more complex machines such as the journal-bearing machine.[7] In the case of the unilaterally loaded journal-bearing machine, the flat plate of these experiments corresponds to the shaft, and the slider corresponds to the bearing. Thus the slider (like the bearing) will accumulate thermal effects and reaction products which, depending upon the oil, may help or hinder lubrication. In contrast, each element of surface area of the plate (like the journal) will bear the load only periodically, allowing better distribution of the heat and chemical products generated. By similar considerations it is believed that these results can be interpreted for many applications to machinery and mechanisms of a more complex nature.

In trying to relate these slider-and-plate observations to some of the performance characteristics to be expected of more complex machines, another factor must be taken into account. The sliding test does not permit ascertaining the effect of break-in. For example, it has been found in the journal-bearing machine that a steel shaft rotating in a bronze bearing does very well if, and only if, a proper break-in procedure is followed.

In the Pesco pump runs it was found that bronze bushings rubbing against the face of steel gears worked well but developed lacquer coatings. This particular combination of metals belongs to the doubtful class in Table 6 because a slight sludge was noticed. Under careful break-in conditions in a pump or journal bearing,[7] this initial wear is avoided, a protective lacquer is formed, and subsequently the bearing is able to stand higher loads than with petroleum oil under similar conditions. Similar observations apply to the case of a brass slider upon steel, and may apply to other combinations. In agreement with this conclusion, when the slider and plates were coated with silicone lacquer by the method described in the companion paper, reference[7] no difficulties were encountered with seizures or scratching in the slider test.

Behavior of Silicone Fluids in Piston Pumps

Prior to observing the performance of the silicone fluid in the Vickers piston pumps, a 300-hr test run at 1500 psi and 180 F sump temperature was made on the petroleum oil AN-VV-O-366b, to provide reference data. Observations and a thorough inspection were made after 100, 200, and 300 hr of continuous operation, and the results are given in Table 7. The wear rates with this petroleum fluid were negligible. The general pump condition remained excellent throughout the test, while the oil remained clear, although an appreciable darkening in color was observed. The viscosity steadily decreased as usual due to shear breakdown of the polymer thickener. The viscosity decrease in the first 5000 cycles was approximately 24 per cent, and as usual the rate of viscosity breakdown rapidly became less after the first 5000 cycles.

TABLE 7 PISTON-PUMP RUNS ON PETROLEUM OIL AN-VV-O 366b

(Sump temperature 180 F)

	Start	After 100 hrs.	After 200 hrs.	After 300 hrs.
Test pressure (psi)	1500	1500	1500	1500
Pressure drop across relief valve (psi)	1450	1450	1450	1450
Flow rate GMM	2.94	2.68	2.85	2.84
Approx. number of cycles	0	5360.	11390.	16450.
Visc. cs. at 100°F	14.27	10.73	9.637	9.58
Percent visc. change	0	24.8	32.4	32.9
Filter type used	Skinner	Skinner	Skinner	Skinner
Appearance of oil	Clear red	Clear red	Clear red	Clear red

TABLE 8 PISTON-PUMP RUNS ON SILICONE FLUID

	Run F1 (Steel knuckles)	Run VPI (Steel knuckles).	Run VP_2 (Bronze knuckles)
Time in hours	65	80	650
Sump temp. °F	140	100	180
PSI high	1500	1500	1500
PSI low	190	75	64
Flow GPM:			
Start	3.18	2.7	3.4
Finish	0	0	2.99
Cycles approximately	1908	4500	38900
Visc. cs. at 100°F			
Start:	71.11	71.11	71.11
Finish:	73.22	71.28	73.67
Percent change	+ 0.3	nil	+ 0.36
Filter type	Skinner	Purolator aircraft	Purolator aircraft
Appearance after run	Cloudy gray	Cloudy gray	Clear, light amber
Initial Condition of Pump	100 Hrs. on 366B	New	New

The same pump was thoroughly cleaned with solvents and operated on the silicone fluid. After 65 hr of continuous operation at 1500 psi and 140 F sump temperature, the pump failed through seizure. In another trial, using a new pump and a new batch of silicone fluid, it failed through fracture of the universal link after 80 hr of continuous operation at the same pressure and at 100 F sump temperature. In each trial the oil became dark with dispersed particles of steel before failure occurred. The surfaces of the steel pins and knuckles were so worn where they rubbed each other that their diameters changed by apporximately one half of the original values, see Fig. 3. Enough gel and finely divided metal was created to accumulate in the filters and increase the pressure drop across the filter by 87 psi. Pump failure was directly due to the excessive wear in these members, allowing the piston connecting rods to strike the sides of the cylinders, either causing seizure or breaking of the universal link.

The pump failures in these tests had in each instance been caused by the rapid wearing of the steel universal link and steel knuckle. This was not surprising in view of the results of the slider-and-plate experiments previously described. From that study a number of much more suitable metal combinations than steel on steel suggested themselves for materials to be used in making the link and knuckles. As few would have been available soon enough, 80-10-10 bronze was used instead of the steel knuckles.

Accordingly, a set of four such knuckles was made and installed in a new Vickers pump and tested for 50 hr of continuous operation at 180 F sump temperature and 1500 psi. At the end of this time the system was completely taken down and inspected. No appreciable wear being apparent, the test was con-

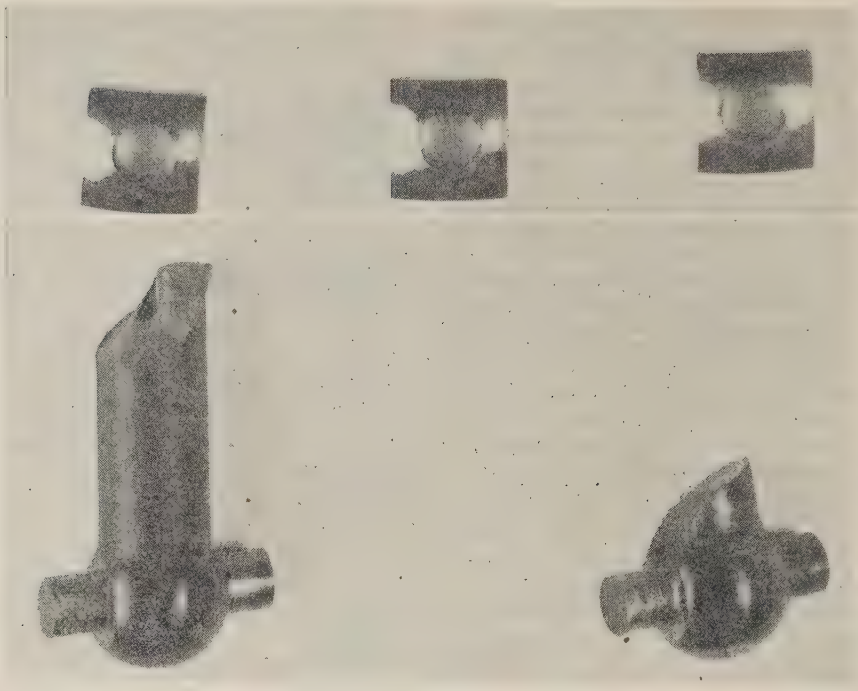


Fig. 3 Universal Link and Steel Knuckles After 80 Hr Operation in Silicone Fluid; Run VPL

tinued for a total of 650 hr of operation, after which it was completely disassembled and inspected. The components of the pump were examined carefully. The universal link pins and knuckles were not unduly worn and the general condition of the pump was considered excellent. The silicone fluid remained clear for the entire test and no significant change was observed in the viscosity. No gel was evident in the fluid and the filter showed freedom from clogging or accumulated slime. However, some gel was found collected around the universal link bearings as it was in all the silicone tests. As the results of the Vickers pump design, these points were exposed to a very small flow of the fluid and any gel accumulated there would have little chance of circulating through the hydraulic system and collecting in the filter.

Hence the use of steel pins and bronze knuckles improved the lubrication characteristics so much as to permit the silicone fluid to be a practicable hydraulic fluid in this type of pump. However, it has not been established here that steel pins and bronze knuckles are the best possible choice of materials. Other combinations of metals deserve consideration by those interested in the necessary changes of design required in pumps to be used with these new fluids.

Conclusions

These experiments make evident the need for discrimination in the use in lubrication and hydraulics of the dimethyl-silicone-polymer fluids. Until a cure is found for the very limited load-carrying capacity of the fluid where both loaded sliding surfaces are ferrous, the application of the fluid to many types of equipment now in use will be limited or impractical. Where one or both of such loaded sliding surfaces are nonferrous or can be replaced by suitable nonferrous metals, much equipment can be adapted to the use of these new fluids. In many cases such changes may require considerable and even undesirable redesign.

The difficulties encountered due to the creeping and leaking tendencies of the silicone fluid will be annoying in some equipment, and the only cure found to date has been in the use of fewer, tighter, and more carefully made mechanical joints. A satisfactory pipe-thread sealant has been found and others probably exist. Its use in tight mechanical joints may be necessary where vibration is severe. The silicone fluids cannot be used where seals are made by metal-to-metal contact without using high sealing pressure or a pipe compound, an example of this difficulty being certain types of aircraft selector valves which require the achievement of a leakproof seal by the use of a steel poppet valve against a steel seat.

The excellent behavior of the silicone fluid in the gear or piston pumps when the proper metal is used for bushings or links and knuckles demonstrates the promise of these new fluids for applications as hydraulic and lubricating fluids. As their peculiarities become better understood, their unusually valuable viscometric, freezing point, and chemical-stability properties will make their adoption increasingly inviting. Therefore, despite the need for redesign, packing changes, tight joints, etc., an increasingly wider use of these remarkable fluids can be expected. The use of the silicone fluid in the composition of the rubber packings instead of the usual plasticizer has shown how readily suitable packings can be prepared.

The protective and durable lacquer formed on the bronze bushings of the gear pump is quite like that found in slowly broken-in journal bearings described elsewhere,[7] and no doubt they are identical in origin and nature. There was no special difficulty with gel formation, for it was significant only after long periods of continuous operation at high sump temperatures. Even then that difficulty appeared adequately cared for by a filter-cartridge change every several hundred hours of continuous operation at high pressure. As the temperature of operation of the pumps rises above 200 F, no doubt the rate of gel formation

will become greater and will necessitate more frequent changes of the filter cartridge.

The results of the simple slider-and-plate tests, although merely qualitative, appear useful. The observed peculiarities of the silicone fluid in lubricating gear and piston pumps, as well as journal bearings,[7] are all in agreement with the predictions made by judicious use of the results of the slider tests, and in fact were used to guide part of the work here described and in the paper on journal bearings.[7] Except for a relatively few combinations, most of the slider-and-plate pairs tried and rated "good" with silicone were also rated "good" with the petroleum oil, and a similar parallelism holds with the combinations rated "bad." Also, many of the metal combinations rated "good" or "bad" for petroleum oil are identical with those experience has shown are useful in bearing systems lubricated with petroleum oils.

The low wear rates found in the gears of the Pesco pumps and the freedom from wear where found in the ball-bearing race of the Vickers pump are considered highly significant with respect to the relation of the behavior of the silicone fluids in rolling and in sliding friction. In both examples of rolling friction encountered here, steel rolled on steel, the unit loads were high, and yet little wear occurred. In contrast, at comparatively very low unit loads in the sliding friction of steel on steel, the failure to lubricate was remarkable. It is well known that sliding friction is less desirable in so far as wear rates are concerned than is rolling friction, but such a contrast as found here requires special explanation. It is suggested that this difference in case of the silicone fluid is due to the fact that, although it has ample chemical stability and adhesiveness to handle high unit loads directed normally to the moving surfaces, it fails under weak loads where sliding occurs as a result of the creation of local hot spots on ferrous and a few other poor-conducting hard metals, causing oxidative breakdown leading to gelling, lacquering, and even local cementing or welding.

ACKNOWLEDGMENTS

It is a pleasure to acknowledge the co-operative efforts of other members of the Lubrication Section, especially Charles M. Murphy, Jr., and Eleanor Bried, Sp(X) 2/c, in making available the necessary viscometric data; Lieut. (jg) G. M. Hain for the careful and useful microscopic examinations; William Kostowicz, MM 2/c, for his skillful machine work; and, in particular, Ensign Harrison Shull, for his experimental and analytical assistance in connection with the slider-and-plate experiments. To Dr. Peter King and Ferdinand Thurman, Sp(X) 3/c of the High Polymer Section, we are indebted for their originality and efforts in the development of the indispensable silicone-oil-resistant packings employed throughout this and related investigations.

Rotary-Pump Theory

By W. E. WILSON,[1] WINNETKA, ILL.

A theory describing the performance of rotary positive-displacement pumps and fluid motors in terms of torque and delivery is presented. From the equations for these two quantities, expressions for power input and power output are developed. The concept of torque efficiency as a complement to the concept of volumetric efficiency is introduced. Equations are developed for the volumetric, torque, and over-all efficiencies. Three operating ranges are identified by outstanding characteristics of the performance in each case. Performance charts in dimensionless form are presented and analyzed. Tests on a conventional gear pump operated as a fluid motor yield data which substantiate the theory in one operating range. Additional tests on a cam-type pump give data confirming the predictions of the theory concerning the performance when pumping oils of high viscosity. Alterations of the physical dimensions of the unit are discussed in terms of effect on the performance characteristics. The possibility of the design of units with optimum dimensions is indicated.

General Theory

The equations which will be developed apply to a rotary positive-displacement unit, acting either as a pump or as a motor. The unit consists of a rotary element, a stationary housing, inlet and outlet passages, and a shaft connected to the rotary element. The definite volume of fluid which is trapped between the rotary element and the housing is positively forced through the unit. Viscous bearing and gear friction as well as other mechanical effects will tend to resist motion of the rotor. These affect the shaft torque by an amount which will be referred to as "torque loss." Flow through the clearances and seals will cause an actual discharge differing from the ideal discharge. These secondary flows are referred to collectively as "slip." A loss in delivery of a pump, not properly designated as slip, is described. This loss is encountered when pumping liquids containing entrained gases, or liquids which vaporize readily.

Three distinct ranges of performance are recognized and defined as follows:

Range 1. All flow of the fluid in the unit in the turbulent range, giving a fluid torque resistance proportional to the square of the angular speed, and slip proportional to the square root of the pressure difference. Slip is independent of the viscosity in this case and torque loss is nearly constant.

Range 2. Laminar flow of the fluid throughout the unit, giving a viscous torque resistance proportional to the first power of the angular speed and first power of the viscosity, and a slip proportional to the first power of the pressure difference and inversely proportional to the viscosity.

Range 3. Laminar flow of the fluid throughout the unit. Slip is negligible due to high viscosity at low speeds. Slip increases as the speed increases due to local heating of the fluid, which in turn is caused by energy dissipated in the shearing action on the viscous liquid. This results in viscous torque loss less than that which would arise from fluid of the original viscosity. Delivery is also reduced by expansion of entrained gases or vapor in the low-pressure region at the pump intake.

In Fig. 1 are shown the characteristics of performance just outlined, in graphical form, in terms of torque and delivery as functions of speed of rotation of rotor shaft at constant pressure differential and constant viscosity at pump intake.

In the development of the theory it is assumed that the rotor shaft is rigid, clearances within the unit remain constant, and the coefficient of friction between unlubricated surfaces is a constant.

Ideal Torque and Delivery

In Fig. 2 is shown a section through a typical rotary pump. Using this sketch for reference, the equations for torque and delivery in general form may be written as follows:

As the rotor of either a pump or a fluid motor rotates, the motion is resisted by a viscous drag which originates in the narrow passages between rotor and housing. In addition there will be resisting torques such as those which originate in the shaft seals which will be nearly constant in magnitude. There are also torques resisting the motion which are proportional to the pressure differential which exists. These torques may originate in the seals if the sealing forces are proportional to the pressure differential, or in the bearings where the resistance is proportional to the bearing load, which in turn depends upon the pressure differential. In the case of the pump the pressure differential causes a torque opposing the motion, and in the case of the fluid motor in the direction of the motion. This torque will be called

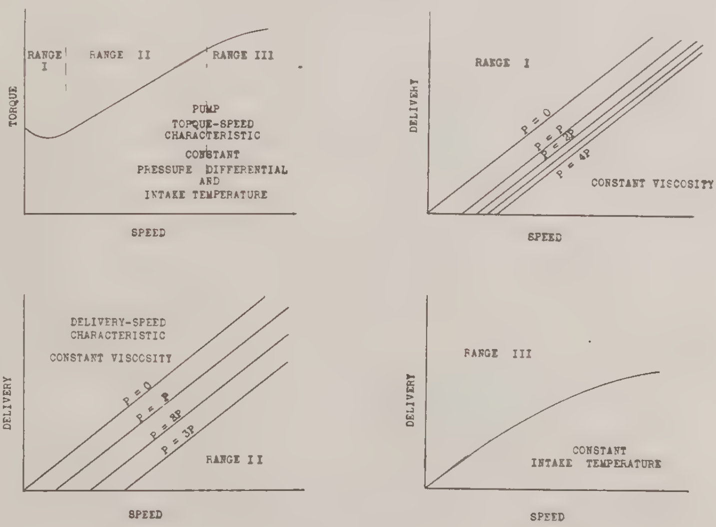
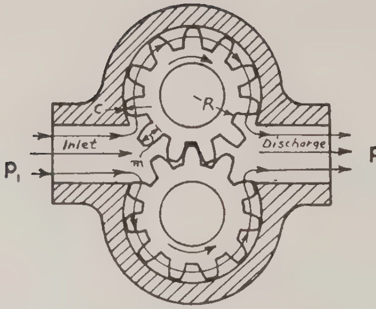


Fig. 1

[1] Consulting Engineer, Winnetka, Ill. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.



Width, perpendicular to sketch, L.

FIG. 2

the ideal torque since it is the torque which would be required to drive the pump or would be developed by the motor if the fluid were frictionless and there were no mechanical friction causing resisting torques.

The torque T required to drive a pump or developed by a fluid motor may be expressed in terms of the torques just discussed as follows

$$\text{Pump:} \quad T = T_i + T_v + T_f + T_c \qquad [1a]$$

$$\text{Motor:} \quad T = T_i - T_v - T_f - T_c \qquad [1b]$$

where

- T_i is ideal torque due to pressure differential and physical dimensions of unit only
- T_v is resisting torque due to viscous shearing of fluid
- T_f is resisting torque, due to mechanical friction, which is directly proportional to pressure differential
- T_c is resisting torque due to mechanical friction which is constant

The delivery of a pump or fluid motor consists of the fluid which is trapped due to the geometrical features of the unit and bodily transported from the inlet to outlet, less that fluid which is returned from the discharge to the intake by the geometrical features of the unit, less that fluid which flows from discharge to intake by reason of the pressure differential (slip) and, in the case of pumped liquids, less the volume of entrained gases or vapor which are expanded at the intake and compressed at the discharge side of the pump. The ideal delivery is that full quantity which is trapped and transported from intake to discharge less that quantity which is returned from discharge to intake by virtue of the geometry of the unit.

The delivery of the pump or fluid motor may therefore be expressed in general terms as follows

$$\text{Pump:} \quad Q = Q_i - Q_s - Q_L \qquad [2a]$$

$$\text{Motor:} \quad Q = Q_i + Q_s \qquad [2b]$$

where

- Q_i is ideal delivery of unit due to its geometrical features only
- Q_s is slip flow caused by pressure differential
- Q_L is delivery loss due to entrained gas or vapor

The ideal torque and ideal delivery of a unit with zero clearance will be distinguished from those of a unit with clearances.

The ideal torque and ideal delivery of a unit with zero clearances may be expressed very simply by considering that the pressure differential acts upon a rotating surface of area A, which by its motion displaces the fluid in a positive manner.

The ideal torque T_i' due to the pressure differential may be expressed in terms of a force $(p_1 - p_2) A$, which acts at the distance r from the axis of rotation thus

$$T_i' = (p_1 - p_2) Ar \qquad [3]$$

where

- p_1 is pressure at intake
- p_2 is pressure at discharge
- A is projected area of moving surface on radial plane
- r is distance from center of pressure to axis of rotation

Similarly, the ideal delivery Q_i' may be expressed as the volume $Ar\omega$ swept out per unit time by the translation of the area A through the distance $r\omega$ thus

$$Q_i' = Ar\omega \qquad [4]$$

where

- ω is angular speed of rotation of rotor shaft
- A is area previously defined
- r is distance from axis of rotation to centroid of area A

It is apparent that the quantity Ar is identical in the two expressions given, when the pressure is constant over the area A. Situations in which the pressure is not constant over the area A will not be considered for the present.

Between the peripheral boundary of the area A and the housing lies a small clearance space through which slip flow occurs. The area of this opening in a radial plane must be considered in some manner in calculating ideal delivery and torque. Definitions to provide for this consideration will be given. Expressions for rate of flow and fluid-resisting torque will be developed on the basis of the assumption that the flow in these small passages is approximated sufficiently closely by the flow between infinite parallel flat plates.

We have defined the ideal delivery as the delivery which would represent the full geometrical delivery of the unit. It is apparent that the fluid which adheres to the moving boundary of the area A in a unit with clearances will result in the transport of fluid through the clearance space. The expression for ideal delivery, Q_i, may then be written

$$Q_i = Ar\omega + ar'\omega$$

or

$$Q_i = \alpha\omega \qquad [5]$$

where

- ar' is volume of fluid carried along by translation of periphery of area A through unit angular distance
- α is equal to $(Ar + ar')$

The quantity α is simply the ideal displacement of the pump or fluid motor per unit rotation of the rotor shaft. This quantity may be most easily determined experimentally by finding the delivery when the pressure differential across the unit is zero. Under these circumstances there can be no slip flow.

One may proceed in a similar manner to write an expression for the ideal torque T_i of a unit with clearance, and for sake of brevity we shall write immediately the expression

$$T_i = (p_1 - p_2)\alpha \qquad [6]$$

where all terms have been defined previously.

The use of the quantity α in the expression for torque as well as delivery may be justified quite rigorously for the case of the ideal pump or fluid motor in a rather simple manner. The mechanical power input to the pump or output of the motor is simply the

product of the torque and angular speed, and the hydraulic power output of the pump or input to the motor is the product of delivery and pressure differential. In an ideal unit the input and output must necessarily be equal. For the case of the ideal unit one may then write

$$T\omega = Q\,\Delta p$$

which in the nomenclature used previously is

$$(p_1 - p_2)\alpha\omega = \alpha\omega(p_1 - p_2)$$

This is obviously an identity as it should be if the quantity α may properly be used in this case. The justification for the use of α in the general case is beyond the scope of the present treatment; however, the case of laminar flow will be considered in detail since the mathematical development is relatively simple.

Slip and Torque Loss

Consider now the flow through a clearance space of the dimensions shown in Fig. 3(*a*). The lower plate moves at the speed $U = r\omega$, where r is the distance from the plate to the axis of rotation. The flow ΔQ_1 through the opening is due to both pressure drop and shear transport of the fluid and is assumed to be in the laminar range.

Referring to Fig. 3(*b*), we consider an element of the fluid at the distance y from the fixed plate, and of the dimensions dx, dy, b. The forces acting on this element are the pressure forces $(p + dp)\,b\,dy$ acting to the right and $p\,b\,dy$ to the left, the shear forces $\tau\,b\,dx$ acting to the left and $(\tau + d\tau)\,b\,dx$ to the right. Assuming steady flow, the acceleration of the element is zero, hence the equation of equilibrium is

$$[(p + dp) - p\,]\,b\,dy + [(\tau + d\tau) - \tau]\,bdx = 0$$

which reduces to

$$\frac{dp}{dx} = -\frac{d\tau}{dy}$$

Since the flow is between parallel plates the pressure gradient dp/dx will be constant. We may therefore integrate the equation just given and obtain

$$\tau = -\frac{dp}{dx}y + \tau_0$$

Newton's viscosity law may now be introduced to relate the shear stress to the velocity gradient

$$\tau = \mu\frac{du}{dy}$$

Substitution of this expression for τ yields the differential equation

$$\mu\frac{du}{dy} = \tau_0 - \frac{dp}{dx}y$$

This may be integrated to yield

$$\mu u = \tau_0 y - \frac{dp}{dx}\frac{y^2}{2} + C_1$$

The boundary conditions are

$$\text{at } y = 0,\quad u = 0$$

$$\text{at } y = c,\quad u = r\omega$$

It follows therefore that

$$C_1 = 0$$

and

$$\tau_0 = \frac{\mu r\omega}{c} + \frac{dp}{dx}\frac{c}{2}$$

The velocity u at any distance y from the fixed plate is then

$$u = \frac{r\omega y}{c} + \frac{1}{\mu}\frac{dp}{dx}\left[\frac{yc}{2} - \frac{y^2}{2}\right]$$

The total quantity of fluid flowing ΔQ_1 is given by

$$\Delta Q_1 = \int_0^c u\cdot b\cdot dy$$

or

$$\Delta Q_1 = \int_0^c \left\{\frac{r\omega y}{c} + \frac{1}{\mu}\frac{dp}{dx}\left[\frac{yc}{2} - \frac{y^2}{2}\right]\right\} bdy$$

and after integration

$$\Delta Q_1 = \left[\frac{r\omega c}{2} + \frac{dp}{dx}\frac{c^3}{12\mu}\right] b$$

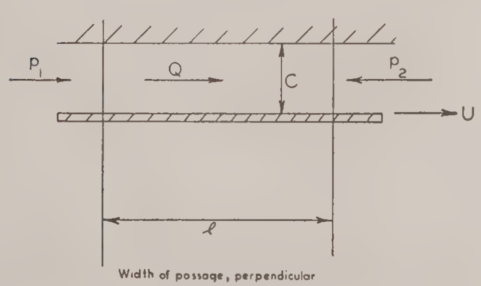
and upon substituting for dp/dx as follows

$$\frac{dp}{dx} = \frac{p_1 - p_2}{l}$$

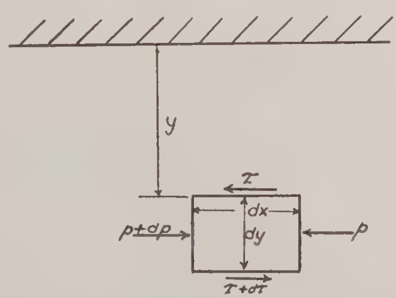
we have finally

$$\Delta Q_1 = \left[\frac{r\omega c}{2} + \frac{(p_1 - p_2)c^3}{12\mu l}\right] b \qquad [7]$$

The unit force τ due to viscous friction which retards the moving plate is given by the value of the shear stress at $y = c$



(*a*)



(*b*)

Fig. 3

$$\tau = -\frac{(p_1 - p_2)}{l}\frac{c}{2} + \frac{\mu r\omega}{c}$$

Since the total force F on the moving plate is

$$F = \tau lb$$

there results a torque ΔT_1 acting on the rotary element of the unit

$$\Delta T_1 = -\left[\frac{(p_1 - p_2)}{l}\frac{c}{2} - \frac{\mu r\omega}{c}\right] lbr \qquad [8]$$

There may be several passages through which the fluid will leak, hence it is necessary to consider the total effect on both torque and delivery. The fact that the torque ΔT_1 and flow ΔQ_1 are linear functions of the pressure difference and angular speed of the shaft permits the equations for net delivery and net torque to be written in the following form

$$Q = Q_i' + \Sigma Q_1$$

$$T = T_i' - \Sigma T_1$$

which upon substitution for ΣQ_1 and ΣT_1 yield

$$Q = Ar\omega + \Sigma\left[\frac{r\omega c}{2} + \frac{(p_1 - p_2)c^3}{12\mu l}\right] b$$

$$T = (p_1 - p_2)Ar + \Sigma\left[\frac{(p_1 - p_2)}{l}\frac{c}{2} - \frac{\mu r\omega}{c}\right] lbr$$

These may be rewritten in the forms

$$Q = \omega\left[Ar + \Sigma\frac{rcb}{2}\right] + \Sigma\frac{(p_1 - p_2)c^3b}{12\mu l} \qquad [9]$$

$$T = (p_1 - p_2)\left(Ar + \Sigma\frac{crb}{2}\right) - \Sigma\frac{\mu\omega lbr^2}{c} \qquad [10]$$

It is apparent that the quantity

$$\left[Ar + \Sigma\frac{rcb}{2}\right]$$

is the factor ω, and $\Sigma\frac{rcb}{2}$ is the quantity ar' previously defined. Also in accordance with previous definitions, the delivery loss due to pressure differential is the slip. It follows therefore that the slip ΔQ_s for a single passage is given by

$$\Delta Q_s = \frac{(p_1 - p_2)c^3b}{12}$$

and the torque loss ΔT_L by

$$\Delta T_L = \frac{\mu\omega lbr^2}{c}$$

Now since in any specific unit (pump or motor) the clearance c, length of path l, width of passage b, and radius to the passage r are different for the various elements of the passages surrounding the rotating element, the expressions for the total torque loss T_L, and total slip Q_s must be written in a more general form. This is done as follows

$$Q_s = \frac{\beta p}{\mu} \qquad [11]$$

$$T_L = k\mu\omega \qquad [12]$$

where

$$p = p_1 - p_2$$

It is readily shown that both k and β have the dimensions length to the third power and may be expressed thus

$$\beta = \Sigma\frac{c^3b}{12l}$$

$$k = \Sigma\frac{lbr^2}{c}$$

In order to simplify the equations and without materially decreasing their significance, the torque loss T_L will be considered to include that loss in torque which is due to entrance and exit losses since in the case of laminar flow, these will be nearly proportional to the product of viscosity and angular speed.

Detailed equations for torque, delivery, power input, power output and efficiencies will now be developed for the three performance ranges defined previously.

Torque and Delivery Range 1

In this range of performance the slip is proportional to the square root of the pressure differential and practically independent of the viscosity. The torque losses are principally mechanical in nature, although the fluid resistance will vary with the square of the speed of rotation and will be practically independent of the viscosity of the fluid. The expressions for torque and delivery may now be written, taking into consideration the nature of the slip and resisting torques, thus

$$\text{Pump:} \quad T = \epsilon p\alpha + k\rho\omega^2R^2 + T_c \qquad [13]$$

$$Q = \alpha\omega - \frac{\beta}{R}\sqrt{\frac{p}{\rho}} - Q_L \qquad [14]$$

$$\text{Motor:} \quad T = \epsilon p\alpha - k\rho\omega^2R^2 - T_c \qquad [15]$$

$$Q = \alpha\omega + \frac{\beta}{R}\sqrt{\frac{p}{\rho}} \qquad [16]$$

where

- T is torque input or output, ft-lb
- p is pressure difference, psf
- T_c is torque due to mechanical friction independent of pressure, ft-lb
- α is ideal displacement, cu ft per radian
- k is a coefficient with dimensions, cu ft
- μ is viscosity of fluid, slugs per ft-sec
- ω is angular speed of shaft, radian per sec
- Q is delivery, cfs
- β is a coefficient with dimensions, cu ft
- ρ is density of fluid, slugs per cu ft
- Q_L is loss in delivery in cu ft per sec due to vaporization of liquid, or liberation of entrained gas at intake
- R is a characteristic radius of the unit, ft
- ϵ is given by the following expression using positive sign for pump and negative sign for motor:

$$\epsilon p\alpha = p\alpha \pm T$$

or

$$\epsilon = 1 \pm \frac{T_f}{\alpha p}$$

T_f is mechanical friction torque which is proportional to pressure differential

Now since T_f is proportional to the pressure differential p, one may write

$$T_f = mp$$

and it follows that

$$\epsilon = 1 \pm \frac{m}{\alpha}$$

which is a constant for any one pump or motor.

Torque and Delivery Ranges 2 and 3

In a similar manner expressions for torque and delivery for both pump and motor in performance ranges 2 and 3 may now be written, using the nomenclature employed previously, as follows

$$\text{Pump:}\quad T = \epsilon p\alpha + k\mu\omega + T_c \quad [17]$$

$$Q = \alpha\omega - \frac{\beta p}{\mu} - Q_L \quad [18]$$

$$\text{Motor:}\quad T = \epsilon p\alpha - k\mu\omega - T_c \quad [19]$$

$$Q = \alpha\omega + \frac{\beta p}{\mu} \quad [20]$$

The distinction between range 2 and range 3 comes in the variation of the viscosity with speed in range 3, and the difference in order of magnitude of the last two terms in the equation for pump delivery. In range 2 the torque shows a straight-line variation with speed, due to the constancy of the viscosity. Also in range 2 the delivery loss Q_L is negligible. In range 3 the torque does not change as rapidly with speed as would be indicated by a constant value of the viscosity, and the slip is either negligible compared with Q_L or shows an increase with the speed corresponding to the decrease in viscosity which is characteristic of this situation.

Power

Equations for power output and power input may be written on the basis of the fundamental expressions

Pump: Power output $= P_0 = Qp$
Power input $= P_i = T\omega$

Motor: Power output $= P_0 = T\omega$
Power input $= P_i = Qp$

Substitution of the previously given expressions for torque and delivery yields the following equations

Range 1

$$\text{Pump:}\quad P_0 = p\alpha\omega\left[1 - \frac{\beta\sqrt{p/\rho}}{R\alpha\omega} - \frac{Q_L}{\alpha\omega}\right] \quad [21]$$

$$P_i = p\alpha\omega\left[\epsilon + \frac{k\omega^2R^2\rho}{p\alpha} + \frac{T_c}{p\alpha}\right] \quad [22]$$

$$\text{Motor:}\quad P_0 = p\alpha\omega\left[\epsilon - \frac{k\omega^2R^2\rho}{p\alpha} - \frac{T_c}{p\alpha}\right] \quad [23]$$

$$P_i = p\alpha\omega\left[1 + \frac{\beta\sqrt{p/\rho}}{\alpha\omega R}\right] \quad [24]$$

Ranges 2 and 3

$$\text{Pump:}\quad P_0 = p\alpha\omega\left[1 - \frac{\beta p}{\mu\alpha\omega} - \frac{Q_L}{\alpha\omega}\right] \quad [25]$$

$$P_i = p\alpha\omega\left[\epsilon + \frac{k\mu\omega}{p\alpha} + \frac{T_c}{p\alpha}\right] \quad [26]$$

$$\text{Motor:}\quad P_0 = p\alpha\omega\left[\epsilon - \frac{k\mu\omega}{p\alpha} - \frac{T_c}{p\alpha}\right] \quad [27]$$

$$P_i = p\alpha\omega\left[1 + \frac{\beta p}{\mu\alpha\omega}\right] \quad [28]$$

Efficiency

In the case of a motor, the ratio of the actual torque to the theoretical torque may be termed the torque efficiency E_T; and the ratio of the theoretical delivery to the actual delivery may be termed the volumetric efficiency E_v. The reciprocals of these quantities are the corresponding efficiencies for the pump. These may be written as follows

Range 1

$$\text{Pump:}\quad E_T = \frac{p\alpha}{T} = \frac{1}{\epsilon + \dfrac{k}{\alpha}\dfrac{\omega^2R^2\rho}{p} + \dfrac{T_c}{p\alpha}} \quad [29]$$

$$E_v = \frac{Q}{\alpha\omega} = 1 - \frac{\beta}{\alpha}\frac{\sqrt{p/\rho}}{\omega R} - \frac{Q_L}{aw} \quad [30]$$

$$\text{Motor:}\quad E_T = \frac{T}{p\alpha} = \epsilon - \frac{k}{\alpha}\frac{\omega^2R^2\rho}{p} - \frac{T_c}{p\alpha} \quad [31]$$

$$E_v = \frac{\alpha\omega}{Q} = \frac{1}{1 + \dfrac{\beta}{\alpha}\dfrac{\sqrt{p/\rho}}{\omega R}} \quad [32]$$

Ranges 2 and 3

$$\text{Pump:}\quad E_T = \frac{p\alpha}{T} = \frac{1}{\epsilon + \dfrac{k}{\alpha}\dfrac{\mu\omega}{p} + \dfrac{T_c}{p\alpha}} \quad [33]$$

$$E_v = \frac{Q}{\alpha\omega} = 1 - \frac{\beta}{\alpha}\frac{p}{\mu\omega} - \frac{Q_L}{aw} \quad [34]$$

$$\text{Motor:}\quad E_T = \frac{T}{p\alpha} = \epsilon - \frac{k}{\alpha}\frac{\mu\omega}{p} - \frac{T_c}{p\alpha} \quad [35]$$

$$E_v = \frac{\alpha\omega}{Q} = \frac{1}{1 + \dfrac{\beta}{\alpha}\dfrac{p}{\mu\omega}} \quad [36]$$

It is convenient to make the following simplification in nomenclature
Let

$$\phi_1 = \frac{\rho\omega^2R^2}{p}$$

in range 1, and

$$\phi = \frac{\mu\omega}{p}$$

in ranges 2 and 3.

The over-all efficiency E is now defined as the power output of the unit divided by the power input; thus

$$E = \frac{P_0}{P_i}$$

The expressions for this over-all efficiency may be written as follows, provided we substitute in each case the appropriate term ϕ or ϕ_1 for the parameters cited previously

RANGE 1

$$\text{Pump:}\quad E = \frac{1 - \dfrac{\beta}{\alpha\sqrt{\phi_1}}}{\epsilon + \dfrac{k\phi_1}{\alpha} + \dfrac{T_c}{p\alpha}} \quad [37]$$

$$\text{Motor:}\quad E = \frac{\epsilon - \dfrac{k\phi_1}{\alpha} - \dfrac{T_c}{p\alpha}}{1 + \dfrac{\beta}{\alpha\sqrt{\phi_1}}} \quad [38]$$

RANGES 2 AND 3

$$\text{Pump:}\quad E = \frac{1 - \dfrac{\beta}{\alpha\phi} - \dfrac{Q_L}{\alpha\omega}}{\epsilon + \dfrac{k\phi}{\alpha} + \dfrac{T_c}{p\alpha}} \quad [39]$$

$$\text{Motor:}\quad E = \frac{\epsilon - \dfrac{k\phi}{\alpha} - \dfrac{T_c}{p\alpha}}{1 + \dfrac{\beta}{\alpha\phi}} \quad [40]$$

It is pertinent to note that significant simplification of the equations for range 2 may be effected by noting the relative order of magnitude of the terms involved in the various equations just given. When operating in this range at fairly high pressures, e.g., over 300 psi, the term $\dfrac{T_c}{p\alpha}$ is frequently found to be negligible in comparison with $k\phi/\alpha$ and ϵ. Also by definition, the term $Q_L/(\alpha\omega)$ is small in this range compared with $\beta/(\alpha\phi)$. The simplified equations in range 2 then reduce to

$$\text{Pump:}\quad E_v = 1 - \frac{\beta}{\alpha\phi} \quad [41]$$

$$E_T = \frac{1}{\epsilon + \dfrac{k\phi}{\alpha}} \quad [42]$$

$$E = \frac{1 - \dfrac{\beta}{\alpha\phi}}{\epsilon + \dfrac{k\phi}{\alpha}} \quad [43]$$

$$\text{Motor:}\quad E_v = \frac{1}{1 + \dfrac{\beta}{\alpha\phi}} \quad [44]$$

$$E_T = \epsilon - \frac{k}{\alpha}\phi \quad [45]$$

$$E = \frac{\epsilon - \dfrac{k\phi}{\alpha}}{1 + \dfrac{\beta}{\alpha\phi}} \quad [46]$$

It is immediately apparent that under these circumstances the performance of the pump or fluid motor may be expressed in terms of dimensionless ratios descriptive of the geometry of the unit, namely, k/α and β/α; and the dimensionless parameter expressed as $\phi = \dfrac{\mu\omega}{p}$, which describes the conditions of operation.

Consideration of the equation for over-all efficiency E in this case leads to the immediate conclusion that there is a particular value of ϕ which corresponds to the maximum efficiency of the unit. The value of ϕ which corresponds to the maximum efficiency may readily be obtained by maximizing the equations for over-all efficiency, considering ϕ as the only variable. The resulting values of ϕ are given by

$$\text{Pump:}\quad \phi_m = \frac{\beta}{\alpha}\left[1 + \sqrt{1 + \frac{\epsilon\alpha^2}{\beta k}}\right] \quad [47]$$

$$\text{Motor:}\quad \phi_m = \frac{\beta}{\alpha}\left[-1 + \sqrt{1 + \frac{\epsilon\alpha^2}{\beta k}}\right] \quad [48]$$

where ϕ_m is the value of ϕ corresponding to E_{max}.

The corresponding values of E_{max} are

$$\text{Pump:}\quad E_m = \frac{1}{\epsilon_1\left[1 + 2\theta + 2\sqrt{\theta(1+\theta)}\right]} \quad [49]$$

$$\text{Motor:}\quad E_m = \epsilon_2\left[1 + 2\theta - 2\sqrt{\theta(1+\theta)}\right] \quad [50]$$

where

$$\theta = \frac{k\beta}{\epsilon\alpha^2}$$

These expressions for efficiency are shown graphically in Fig. 4

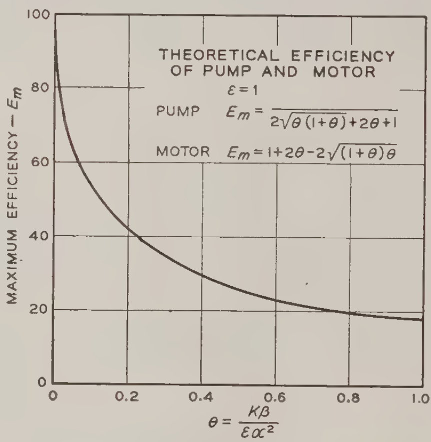


FIG. 4

for the case of $\epsilon_1 = \epsilon_2 = 1$. It is readily shown that motor and pump efficiencies are identical in this case. It is immediately apparent that the maximum efficiency is a function of θ and ϵ only, and that these in turn are determined by the geometry of the unit only.

Since the concept of torque efficiency is somewhat unfamiliar, it is well to point out the relationship which exists between the volumetric efficiency E_v, the torque efficiency E_T, and the over-all efficiency E. By definition the following expressions may be written for a pump

$$E_v = \frac{Q}{Q_i}$$

$$E_T = \frac{T_i}{T}$$

$$E = \frac{Qp}{T\omega}$$

The product of E_v and E_T may then be obtained thus

$$E_vE_T = \frac{QT_i}{Q_iT}$$

Substituting the following quantities for ideal torque and ideal delivery

$$T_i = p\alpha \qquad Q_i = \alpha\omega$$

we obtain

$$E_vE_T = \frac{Qp}{T\omega}$$

from which it may be concluded that the product of the volumetric efficiency and the torque efficiency is equal to the over-all efficiency. A check of the equations given previously for the various cases reveals that they are in agreement with this general conclusion, and a similar demonstration may be made for the case of a motor.

Local Heating of Liquid

It has been pointed out that the reduction in viscosity by the heating developed in the shearing action on the fluid in the narrow passage between rotor and housing is a most important feature of the performance in range 3 (44).[2] An equation describing the important characteristics of this phenomenon will be derived.

The narrow passage, shown schematically in Fig. 3(*a*), will be used as the basis of this development. The work done on the fluid which flows through the passage is equal to the product of the force exerted by the moving plate on the fluid and the speed at which the plate moves. It will be assumed that this energy which is supplied to the fluid is dissipated entirely in raising the temperature of the fluid. The following expressions for work done and heat absorbed may be written

$$\text{Work done on fluid} = \left[\mu\frac{r\omega}{c} + \frac{(p_1 - p_2)c}{2}\right]r\omega bl$$

$$\text{Heat absorbed by fluid} = c_sQw\Delta T$$

where

c_s is heat required to raise temperature of 1 lb of liquid 1 deg F
w is weight of 1 cu ft of liquid
ΔT is rise of temperature in deg F of liquid in distance l
μ is coefficient of viscosity of liquid in slugs per ft-sec
Q is rate of flow of liquid through narrow opening relative to moving plate, cfs

All other terms have been defined previously or are defined in Fig. 3(*a*).

Equating these rates of energy supply and absorption, assuming no heat loss to the surroundings, introducing the mechanical equivalent of heat J and the value of Q from Equation [7], we have

$$J\,c_s\left[\frac{r\omega c}{2} + \frac{(p_1 - p_2)c^3}{12\mu l}\right]bw\Delta T = \left[\frac{\mu r\omega}{c} + \frac{(p_1 - p_2)c}{2l}\right]r\omega bl$$

Suitable algebraic manipulations lead to the following expression for the temperature rise

$$\Delta T = \frac{\mu l r\omega}{c_sc^2wJ}\left[2 + \frac{1}{\frac{1}{4} + \frac{3r\omega\mu l}{2c^2(p_1 - p_2)}}\right] \ldots\ldots [51]$$

It is readily determined that the expression contained within the brackets must have a value between 2 and 6. For purposes of simplification it is assumed that a constant value of 4 for the quantity within the bracket will yield results sufficiently reliable for practical use.

While it is necessary to calculate the temperature rise in the slip passage, it is further required that the change in viscosity be determined. For purposes of calculation the viscosity μ of the liquid is approximated by the expression

$$\mu = \mu_0e^{-\gamma T} \ldots\ldots [52]$$

where

μ_0 is viscosity at $T = 0$
γ is a constant for any particular liquid
e is base of natural logarithms

Considering now the change dT in the temperature along a length dx of the passage, we may rewrite Equation [51] upon using the value 4 for the bracket thus

$$\frac{dT}{\mu_0e^{-\gamma T}} = \frac{4r\omega}{c_swc^2J}dx$$

This may be integrated to yield

$$\mu = \frac{B\mu_1}{\mu_1x + B} \ldots\ldots [53]$$

where

$$B = \frac{c_swc^2J}{4\gamma r\omega}$$

and x is the distance along the passage to the point at which the viscosity is μ. The quantity μ_1 is the viscosity at entrance to the pump or motor.

The average value of the viscosity in the passage is of particular significance. Denoting this average value by μ_{avg}, it may be calculated as follows

Define

$$\mu_{avg} = \frac{\int_0^l \mu dx}{l}$$

then

$$\mu_{avg} = \frac{1}{l}\int_0^l \frac{B\mu_1}{\mu_1x + B}\cdot dx$$

Integration and simplification yield

$$\mu_{avg} = \frac{B}{l}\log_e\left[\frac{\mu_1l}{B} + 1\right] \ldots\ldots [54]$$

This may be put into a more significant form by dividing both sides of the equation by μ_1 the initial viscosity of the liquid

$$\frac{\mu_{avg}}{\mu_1} = \frac{B}{\mu_1l}\log_e\left[\frac{\mu_1l}{B} + 1\right]$$

which may be rewritten in the following form

$$\frac{\mu_{avg}}{\mu_1} = \frac{c_swc^2J}{4\gamma r\omega l\mu_1}\log_e\left[\frac{4\gamma r\omega l\mu_1}{c_swc^2J} + 1\right] \ldots\ldots [55]$$

The variation of μ_{avg} with angular speed ω with constant pressure differential is particularly significant. When the angular

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

speed is zero the ratio $\frac{\mu_{avg}}{\mu_1}$ reduces to the form 0/0 which may be shown mathematically to have the value 1, as would be expected from purely physical considerations. As the angular speed increases the ratio $\frac{\mu_{avg}}{\mu_1}$ approaches zero. The viscous torque may be expressed in terms of the reduced viscosity μ_{avg}

$$T_v = k\mu_{avg}\omega$$

and upon substituting the expression for μ_{avg}

$$T_v = \frac{kc_s wc^2 J}{4\gamma rl} \log_e \left[\frac{4\gamma rl\omega\mu_1}{c_s wc^2 J} + 1\right] \ldots\ldots\ldots [56]$$

which shows a continually increasing value of the torque T_v as the speed increases. However, T_v will, under these circumstances, always be less than $k\mu_1\omega$, the torque which would be encountered if the viscosity remained at its initial value μ_1.

A detailed analysis of the problem of the effect of the reduction in pressure in the neighborhood of a pump inlet on delivery has been given by Pigott (42, 44), for the case of volatile low-viscosity liquids. The effect of viscous resistance in reducing inlet side pressures and thereby causing loss in delivery due to the expansion of entrained air, may be analyzed in the elementary manner to be discussed.

Viscosity and Entrained Gases

When pumping extremely viscous liquids the resistance to flow causes excessive reductions in pressure in the intake passages. Entrained gas will expand in low-pressure regions in the neighborhood of the inlet and fill to a greater or lesser extent the volume intended for the transport of liquid from the intake to the discharge of the pump. When this gas reaches the discharge side of the pump its volume is reduced due to the high pressure encountered (39, 42, 44, 45).

The volume of entrained gas per unit volume of liquid is V_g, at the inlet pressure p_1, and this volume increases to V_g', due to the reduction in pressure as the liquid enters the trapping space in the pump. Then the expanded gas and sufficient oil to fill the displacement volume of the pump are transported to the high-pressure side of the pump where the volume of the gas is reduced to V_g''. If the flow in the intake passages and adjacent to the rotor is considered to be as indicated in Fig. 5, and the flow is treated as

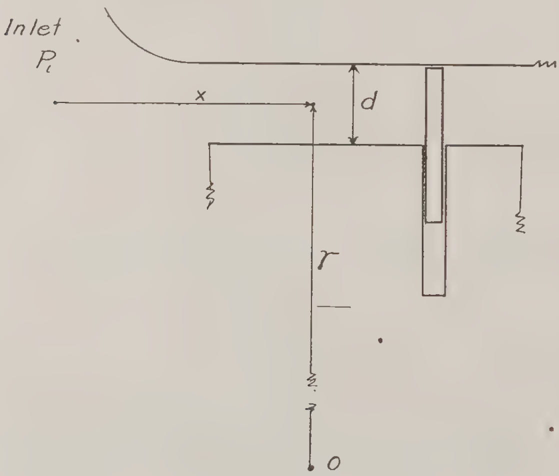


Fig. 5

laminar flow between parallel flat plates, one may state that to fill the space behind the advancing vane a quantity of liquid $bdr\omega$ must be drawn into the space per unit time. Now making use of Equation [7] which describes the flow between parallel flat plates, and setting the rate of flow equal to $bdr\omega$ we have

$$bdr\omega = \left[\frac{r\omega d}{2} + \frac{(p_1 - p)d^3}{12\mu x}\right] b$$

This may be solved for the pressure p and yields

$$p = p_1 - 6\mu\omega \frac{rx}{d^2} \ldots\ldots\ldots\ldots\ldots [57]$$

where

p is pressure at any point within inlet passage, psf
x is distance in feet from intake to point at which p is measured
r is radial distance in feet to point at which p is measured
d is radial distance in feet from rotor to housing

Equation [57] may be put in dimensionless form

$$\frac{p}{p_1} = 1 - \frac{6rx}{d^2}\frac{\mu\omega}{p_1} \ldots\ldots\ldots\ldots\ldots [58]$$

The loss in delivery will be caused by the expansion of the entrained gas of volume V_g. Assuming this expansion takes place isothermally, we have

$$\frac{V_g}{V_g'} = \frac{p}{p_1}$$

where

V_g is volume of entrained gas at intake pressure p_1 per unit volume of liquid
V_g' is expanded volume of entrained air at pressure p per unit volume of liquid

As the liquid and gas progress to the region of high pressure a reduction in volume of the gas to V_g'' is accomplished. Assuming isothermal compression, we may write

$$\frac{V_g''}{V_g} = \frac{p_1}{p_2}$$

where p_2 is the pressure at the discharge of the pump. The loss in volume V_L per unit volume of liquid due to expansion of the gas is simply the difference between V_g' and V_g''. Expressing this as an equation, we have

$$V_L = V_g' - V_g'' = \frac{V_g p_1}{p} - \frac{V_g p_1}{p_2}$$

and upon substituting for p_1/p there results

$$V_L = V_g \left[\frac{1}{1 - \frac{6rx}{d^2}\frac{\mu\omega}{p_1}} - \frac{p_1}{p_2}\right] \ldots\ldots\ldots [59]$$

The quantity V_L represents, therefore, the proportion of the displacement which is lost at a point located x feet from the inlet due to expansion of entrained gas. The quantity V_g is simply the ratio of the volume of entrained gas to the volume of liquid at intake pressure.

It will be noted that the loss in displacement V_L is associated with the distance x. In order to obtain the total loss in displacement it is necessary to sum up the losses V_L, at each point x, over the entire region involved.

Although this equation is based on gross oversimplifications it indicates the nature of the variables which describe the phe-

nomenon, and it is evident that greater refinements will lead to expressions containing the same parameters $\frac{\mu\omega}{p_1}$ and $\frac{p_1}{p_2}$ and differing only in algebraic form. Summation of the losses in displacement will again change only the form of the expression and not the quantities involved. Use will be made of this observation in analyzing experimental data.

Experimental Data for Hydraulic Motor

Data obtained in the test of a conventional gear pump operating as a hydraulic motor are presented in dimensionless form in Figs. 6, 7, 8, and 9. The tests were performed on a pump with a theoretical displacement of 0.283 gal per 100 revolutions. Oils with viscosities ranging from 200 SSU to 6000 SSU were used. Pressures ranged from 180 psi to 600 psi. The speed, torque output, capacity, temperature of oil and pressure differences were observed and the viscosity of the oil determined at various temperatures.

In Fig. 6 the torque efficiency is shown as a function of the dimensionless parameter $\frac{\mu\omega}{p}$. In Fig. 7 the reciprocal of the volumetric efficiency is shown as a function of the dimensionless parameter $\frac{\mu\omega}{p}$. In Fig. 8 the reciprocal of the volumetric efficiency is again shown as a function of $\frac{\mu\omega}{p}$, but the abscissa is in this case $\left(\frac{\mu\omega}{p}\right)^{-1}$. In Fig. 9 the over-all efficiency is shown as a function of $\frac{\mu\omega}{p}$. The quantities β and k are evaluated, assuming the quantity T_f to be negligible, from the data shown in Figs. 6, 7, 8, and 9. The solid line in Fig. 9 is computed on the basis of these values.

In Figs. 11 and 12 are shown the plottings of experimental data on the performance of a cam-type rotary pump operating in range 3. The plotting in Fig. 11 of torque input as a function of speed is with constant pressure differential and approximately constant viscosity at intake. In Fig. 12 is shown a plotting of loss in delivery as a function of the parameter $\frac{\mu\omega}{p_1}$, where ω is the speed of rotation of the shaft, for the same pump operating with a wide range of intake pressures and viscosity.

Evaluation of Theory and Data

The data may be considered to provide a striking confirmation of the predictions of the theory concerning the performance characteristics of a conventional gear pump operating as a motor in performance range 2 and of the effect of high viscosity on a pump operating in range 3. The dimensionless plotting of the torque efficiency as a function of the parameter $\frac{\mu\omega}{p}$, shown in Fig. 6, is characterized by a close grouping of the points about a straight line as predicted by theory. The deviation of a small number of points representing runs at the lowest pressure used, namely, 180 psi, indicating as they do a considerable magnitude of the factor $T_c/(p\alpha)$, might well provide material for an interesting study of the pump mechanism. The volumetric efficiency plotted in reciprocal form in Fig. 8, shows an excellent approach to the straight line predicted by the theory. In Fig. 9 points representing data on the over-all efficiency show excellent agreement with the curve calculated from the values of β, k, and ϵ, which were evaluated by means of the previous plottings.

As predicted by Equation [48], the maximum efficiency occurs very close to $\frac{\mu\omega}{p} = 69 \times 10^{-7}$. The peak efficiency with a few exceptions lies in the range 58 to 67 per cent compared with the

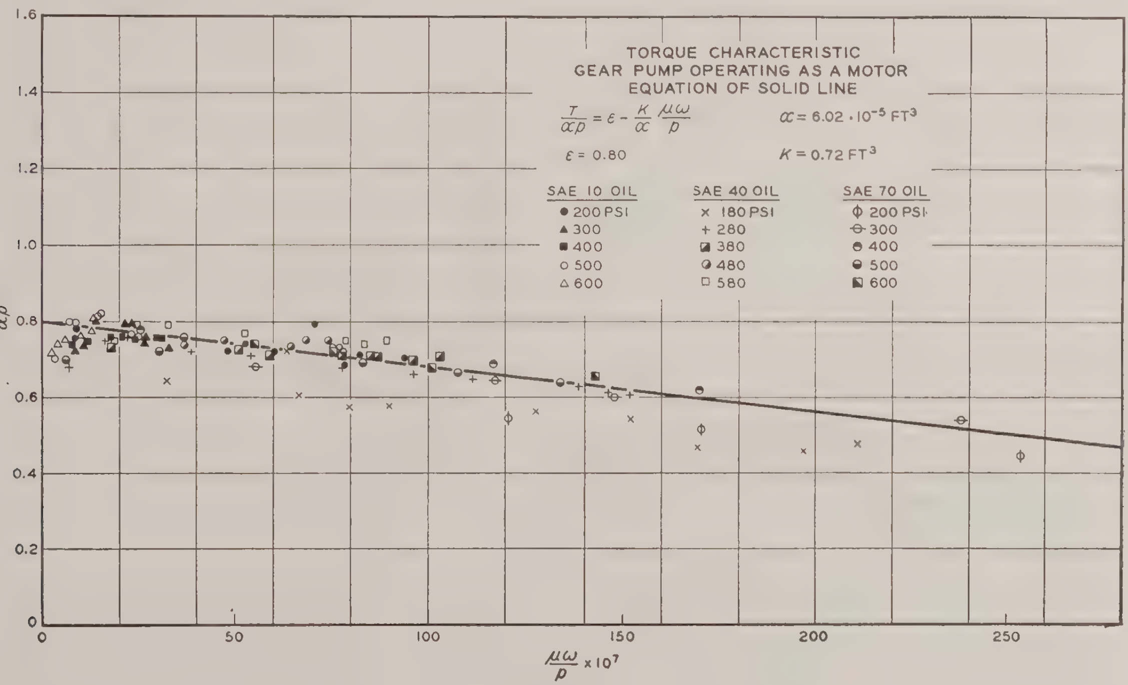


FIG. 6

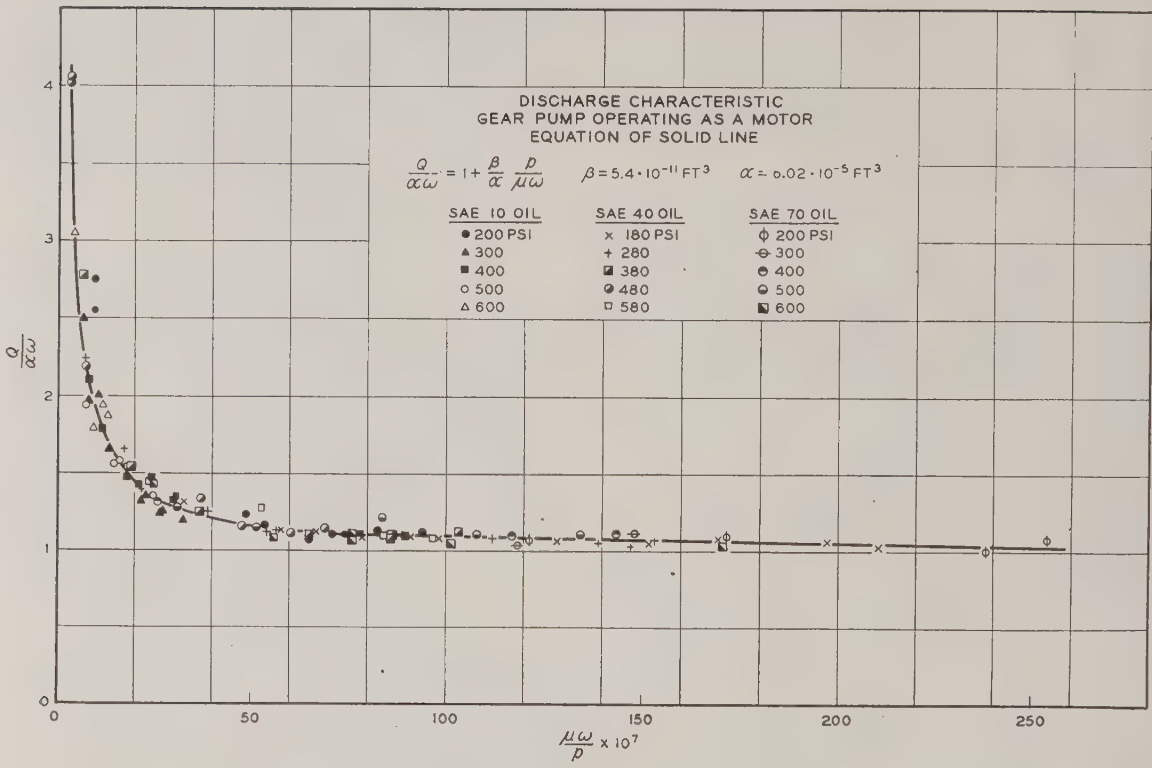

DISCHARGE CHARACTERISTIC
GEAR PUMP OPERATING AS A MOTOR
EQUATION OF SOLID LINE
$\frac{Q}{\alpha\omega} = 1 + \frac{\beta}{\alpha}\frac{p}{\mu\omega}$
$\beta = 5.4 \cdot 10^{-11}$ FT^3
$\alpha = 6.02 \cdot 10^{-5}$ FT^3
SAE 10 OIL
200 PSI
300
400
500
600
SAE 40 OIL
180 PSI
280
380
480
580
SAE 70 OIL
200 PSI
300
400
500
600
$\frac{Q}{\alpha\omega}$
$\frac{\mu\omega}{p} \times 10^7$
0
1
2
3
4
50
100
150
200
250

Fig. 7

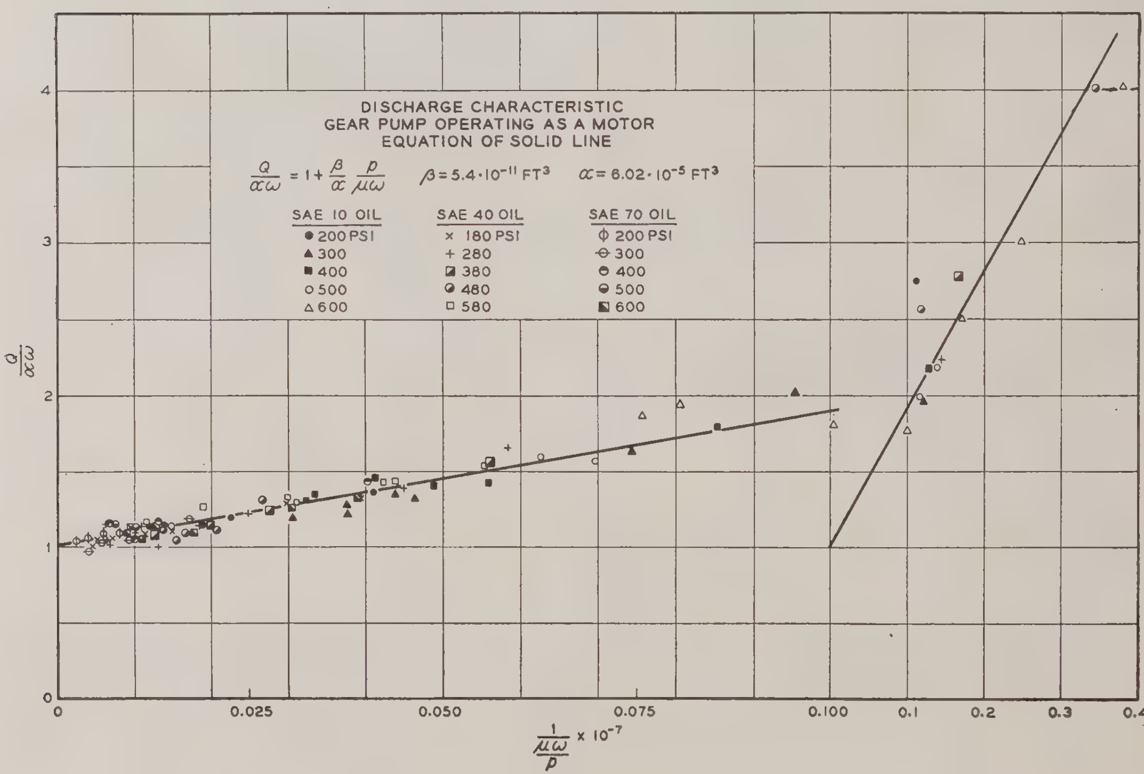

DISCHARGE CHARACTERISTIC
GEAR PUMP OPERATING AS A MOTOR
EQUATION OF SOLID LINE
$\frac{Q}{\alpha\omega} = 1 + \frac{\beta}{\alpha}\frac{p}{\mu\omega}$
$\beta = 5.4 \cdot 10^{-11}$ FT^3
$\alpha = 6.02 \cdot 10^{-5}$ FT^3
SAE 10 OIL
200 PSI
300
400
500
600
SAE 40 OIL
180 PSI
280
380
480
580
SAE 70 OIL
200 PSI
300
400
500
600
$\frac{Q}{\alpha\omega}$
$\frac{1}{\frac{\mu\omega}{p}} \times 10^{-7}$
0
1
2
3
4
0.025
0.050
0.075
0.100
0.1
0.2
0.3
0.4

Fig. 8

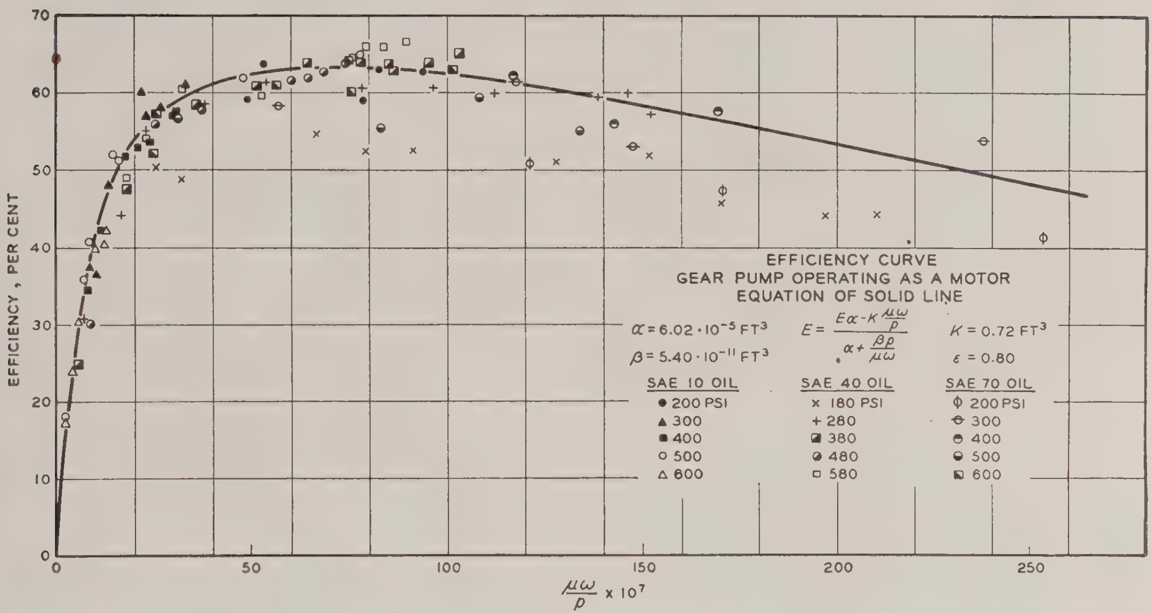


FIG. 9

theoretical maximum of 63.4 per cent, computed by means of Equation [50].

The use of the dimensionless performance curves in evaluating a particular design may now be studied in some detail. In Fig. 10 are illustrated the particular features of these plottings which make them extremely powerful tools in analyzing performance and planning improvements. For simplicity and ease of comparison with the experimental data, the case of a hydraulic motor is considered.

In Fig. 10(a) is shown a graphical representation of the discharge characteristics of the unit on the basis of Equation [36] which may be inverted for this purpose

$$\frac{Q}{\alpha\omega} = 1 + \frac{\beta}{\alpha}\frac{p}{\mu\omega}$$

This plots as a straight line on rectangular co-ordinate paper, with $\dfrac{Q}{\alpha\omega}$ as ordinate and $\dfrac{p}{\mu\omega}$ as abscissa. The intercept on the vertical axis at $\dfrac{p}{\mu\omega} = 0$ is necessarily unity, since $\dfrac{p}{\mu\omega} = 0$ corresponds to either zero pressure difference, infinite speed, or infinite viscosity, each of which obviously necessitates zero slip or $\dfrac{Q}{\alpha\omega} = 1$. The slope of the curve is $\dfrac{\beta}{\alpha}$, a physical characteristic of the unit which is a dimensionless ratio. This ratio would be given, if all slip passages corresponded to the idealized case in Fig. 3(a), by the quantity

$$\frac{\beta}{\alpha} = \Sigma\,\frac{c^3b}{12lcrb} = \Sigma\,\frac{c^2}{12lr}$$

where r is the distance from the moving plate to the axis of rotation. In the case of an actual unit this factor $\dfrac{\beta}{\alpha}$ will depend, as indicated, upon the clearances, lengths of flow paths, and distances to the axis of rotation, for all the passages through which leaks take place.

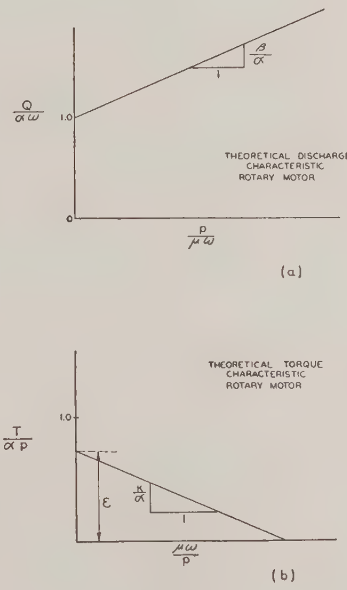


FIG. 10

Similarly in the plotting of torque efficiency $\dfrac{T}{p\alpha}$ versus $\dfrac{\mu\omega}{p}$ a straight line is expected, since the simplified expression obtained from Equation [35] for torque efficiency is

$$\frac{T}{\alpha p} = \epsilon - \frac{k}{\alpha}\frac{\mu\omega}{p}$$

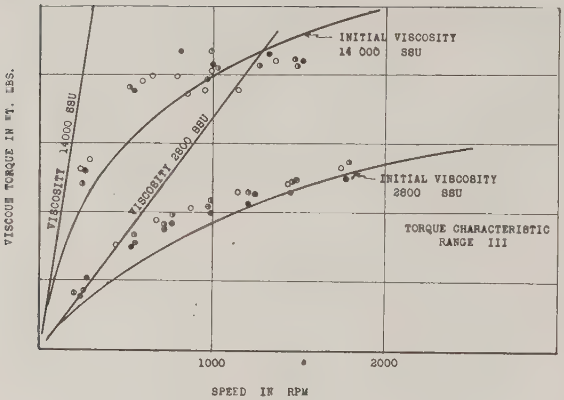


FIG. 11

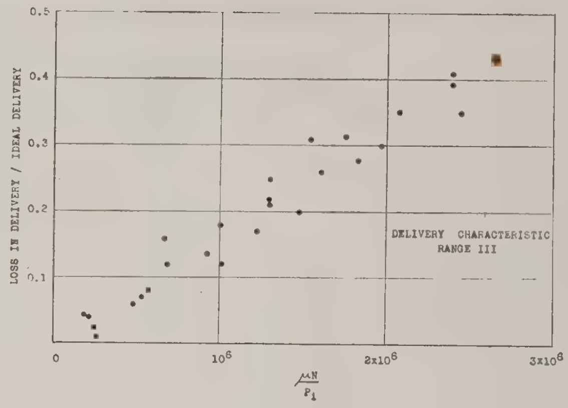


FIG. 12

Deviations of the actual performance from that indicated by the theory will occur if the clearances or alignment of the parts of the unit change, or if the nature of the flow in the unit is different from that assumed, e.g., turbulent rather than laminar. Such changes in clearances and alignment may be caused by pressure increase or decrease or by centrifugal effects due to speed changes.

If in a test the viscosity and pressure difference are held constant at certain values during successive series of runs, one would expect the data to plot as series of points forming straight lines on both types of plottings, shown in Fig. 10. Deviations from the theoretical form of the plots may be of the nature to be indicated and have the significance stated.

If the plotted data give points forming straight lines but the slopes vary from series to series this variation in slope must be caused by changes in the geometrical features of the unit by the pressure, since the viscosity was held constant.

If the curves are not straight lines the reason may be either that the flow is not in the assumed regime, e.g., it is assumed laminar but actually proves to be turbulent, or the geometrical features of the unit change with speed due to centrifugal effects.

If the plottings are parallel straight lines but the intercepts vary from series to series, the cause must be sought in changes in geometry due to pressure resulting in a change in mechanical friction.

The significance of the parameter $\frac{\mu\omega}{p}$ is of note. Since the maximum efficiency occurs at a specific value of the parameter $\frac{\mu\omega}{p}$, it is obvious that this maximum may be attained with various combinations of μ, ω, and p. For example, use of an oil of great viscosity will result in maximum efficiency at very low speed for a given pressure difference. The corresponding speed at a high pressure difference will be proportionately greater. Use of an oil of low viscosity will result in maximum efficiency at higher speeds for the same pressure difference.

A general characteristic of hydraulic units operating with laminar flow is apparent from Equations [49] and [50] in that efficiency of the unit operated as a pump will always be the same as the efficiency of the unit operated as a motor, if $\epsilon = 1$. The difference in efficiency in the two cases will never be very great unless the mechanical friction is greatly different in the two cases.

Specific details of the experimental data will be analyzed, keeping in mind the general principles previously set forth.

The plotting of the data in Fig. 6 makes possible a determination of the coefficient k since $\left(-\frac{k}{\alpha}\right)$ is the slope of the straight line which is taken as a fair representation of the trend of the data. Deviations from the straight line may be due to at least two specific causes, as follows:

1 Local heating of the oil due to energy dissipation causes a local viscosity far different from that measured at the discharge of the unit, thus placing operation in range 3 rather than in range 2 as assumed.

2 Variations in orientation and clearances of the elements of the unit and their lubrication during the runs cause changes in both the mechanical friction and the viscous resistance.

The value of ϵ is given by the value of E_T at $\frac{\mu\omega}{p} = 0$. The plotting of Fig. 7 shows an asymptotic approach of $\frac{Q}{\alpha\omega}$ to the value unity as $\frac{\mu\omega}{p}$ increases as would be expected from the theory. Fig. 8 gives the necessary information to evaluate β since the slope of the solid line is $\frac{\beta}{\alpha}$.

The interpretation of the factor ϵ is important since it represents a major element in the design of hydraulic units. The value of ϵ appears to be essentially a constant for much of the data and its nature as a factor, analogous to a coefficient of kinetic friction describing mechanical friction, appears to be confirmed since the coefficient of kinetic friction is practically independent of speed, pressure, and viscosity. Certain deviations from the expected are evident in the data, especially for runs at 180 psi, which might be interpreted as a change in ϵ but may better be explained on the basis of the neglected term of Equation [35]. The experimentally determined value of 0.80 for ϵ indicates a 20 per cent loss through mechanical friction in this particular pump. A considerable improvement in performance might be effected by a suitable change in design.

It is also pertinent to point out that Equation [50] indicates an efficiency, which might well be termed the hydraulic efficiency, of better than 78 per cent, if $\epsilon = 1$, for all values of θ less than approximately 0.02. It is clear from a consideration of the elements of the pump geometry that a value of θ in excess of 0.02 is a practical impossibility in even a poorly conceived design. The value of θ for the pump studied was approximately 0.013, which corresponds to an efficiency of approximately 79 per cent.

A study of the equations developed reveals the effect of changes in the physical dimensions of the unit which might be made to alter the performance characteristics by changing either clearance or displacement or both.

It is apparent that an increase in clearance results in

1 Decrease in k.
2 Increase in β.

An increase in α results in

1 Increase in torque.
2 Increase in capacity.

However, it is pertinent to point out that a detailed analysis of the performance of a rotary pump as influenced by its geometrical features reveals that for any given value of rotor length and rotor diameter there is an optimum value for the clearance for best performance in range 2. It may be shown also that the performance, solely from a hydraulic standpoint, improves as the ratio of rotor length to rotor diameter is increased. There is of course an obvious limitation on this increase imposed by shaft deflection.

The data for pump performance in range 3 reveals a number of interesting features. The plotting of torque as a function of speed shows the characteristic decrease in slope of the curve as the speed increases. The solid lines indicating the torque which would be expected for the given intake viscosity, if there were no local heating, were obtained from experiments at lower viscosities. The solid curved line for the case of initial viscosity of 2800 SSU was calculated to pass through one of the experimentally determined points thus evaluating the constant $\frac{B}{l}$ of Equation [54] for the pump. The curved solid line for the initial viscosity of 14,000 SSU was then calculated using the same constant. It is apparent that the agreement with the experimental data is as good as could be expected considering the approximate nature of the theory and the oversimplification involved in it.

The relationship between loss in delivery and the parameter $\frac{\mu\omega}{p_1}$ is particularly interesting when it is realized that the plotted points represent a wide variation in viscosity, speed, and intake pressure. The effect of discharge pressure has been neglected in the plotting, hence one may expect that the scattering of points may well be due to some extent to this neglected factor.

A detailed analysis of pump-intake geometry should lead to useful equations predicting the effect on performance in so far as lost delivery with high viscosity is concerned.

Conclusions

On the basis of the experimental data the following conclusions may be derived:

1 Predictions of the theory are borne out.

2 Experimental data may be presented in a compact dimensionless form.

3 Losses in efficiency may be traced readily and the causes determined.

4 Hydraulic and mechanical factors in performance are shown in proper perspective and may be analyzed separately.

5 Performance of units with fluids giving flows in the turbulent range may be expected to be subject to effective study by means of this theory and method of analysis of data.

Acknowledgments

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Air-Gas Ratio Control Applied to Nonatmospheric-Pressure Furnaces

By H. C. McRae,[1] Cleveland, Ohio

For reasons of economy and efficiency, in the burning of fuel gases in industrial furnaces it is essential to maintain the air-gas ratio at approximately the correct point for perfect combustion. Furthermore it is important to control the ratio of air flow to gas flow because of the effect of burned gases or atmosphere upon material being heat-treated. The author discusses in detail the various control processes available, the equipment developed for the purpose, and analyzes the several control circuits, basing the work on the laws of low-pressure gas flow or noncompressible flow.

Introduction

In the process of burning fuel gases in the combustion chamber of an industrial furnace, usually the approximate amount of air required for perfect combustion is introduced along with the gas in some form or another. For the purpose of fuel economy alone, at elevated temperatures, the air-gas ratio must be held close to the proper amount. Aside from this reason, it is important to control the ratio of air flow to gas flow because of the effect of the burned gases or atmosphere upon the material being heat-treated. Sometimes the reaction produced by a small amount of excess air is desirable, while in other applications a deficiency of air may produce the desired result. Of course the air may be introduced as primary air, that is, air which is mixed with the gas before reaching the combustion chamber, or as secondary air which mixes with the gas as it is burning. "Free" air from the atmosphere is often used for a source of secondary air. In this case the air enters directly through ports in the furnace wall because of furnace draft, or this air may be induced by the momentum of the mixture of primary air and gas through a burner port which is discharging into an opening in the furnace wall. The quantity of free secondary air is difficult to control because it is so easily affected by small changes in furnace pressure and therefore that application is excluded from this discussion.

For manual or automatic control of temperature a variable heat input is required. In batch-type furnaces a high heat input is used to heat up a cold furnace or to heat up a cold load. As the desired temperature is reached the heat input is decreased until it just balances the losses of heat through the furnace wall and out of the flue. In continuous furnaces a variable heat input is used to maintain a variable loading at a constant temperature. Variable heat flow means variable gas- and air-flow rates. Maintenance of the correct air-gas ratio with variable demand of heat flow and with variable furnace-pressure conditions, then, is the subject under discussion.

The amount of air required for perfect combustion of natural gases is approximately 10 parts to 1 of gas and for manufactured gases is about 5 parts to 1 of gas. These two gases are the ones most commonly used, although there are others which require more air and some which require less air. Air is therefore considered to be the principal fluid because of its greater quantity. This means that the energy required for the mixing process and delivery of the mixture to the furnace can best be obtained from the air stream rather than from the gas stream. Another reason for considering the air to be the principal fluid is that it may be conveniently compressed to a pound (per square inch) pressure or so by the use of the proper blower while gas is frequently furnished at only a few ounces (per square inch) pressure. Occasionally, high-pressure gas of several pounds pressure is available, in which case the energy from the gas stream may be used to induce free primary air from the atmosphere, but here again as in the use of free secondary air, the air-gas ratio is upset easily by changes in furnace pressure. The use of free air is therefore not considered here. The following discussion assumes that air may be furnished at a pressure of $^1/_4$ to 2 psi, whether it be primary or secondary air.

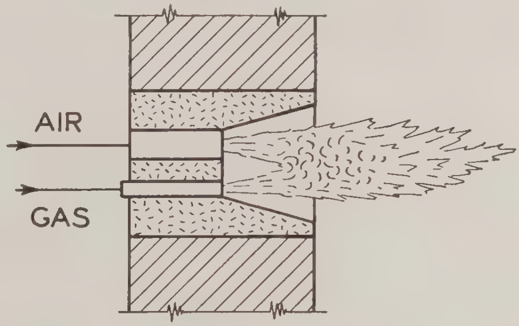

Fig. 1

The gas pressure available will be considered as three distinct cases, low, medium, or high pressure, and may affect the type of control to be used for air-gas ratio control. The low-pressure gas will be reduced to zero pressure or approximately so. Medium-pressure gas will be higher than this, but not as high as the air pressure furnished. High-pressure gas is available at as high a pressure as the air pressure or higher.

The air-and-gas-mixing apparatus or, simply, "mixer" is the unit where the gas and air are brought together and where the mixing process has begun but may not necessarily be completed. In the use of secondary air the mixer is installed in the furnace wall and is usually called the burner, although for purposes of discussion here it shall be considered a mixer. A primary-air mixer discharges the mixture of air and gas to a piping manifold where it is delivered to a single burner or a multiplicity of burners. In either case, there are recognized three distinct classes of mixers. This classification is based upon the effect of the air flow on the gas flow, that is, whether the air tends to help or hinder the flow of gas. This effect may be measured by discharging the mixer into the atmosphere and turning off the gas supply. The resulting pressure obtained at the gas entrance, which is produced by the air flow, is the key to the classification. If the pressure so

[1] Research Engineer, The North American Manufacturing Company.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

obtained is zero or very nearly so, as in Fig. 1, this type is defined as a neutral mixer. On the other hand, if a back pressure is obtained on the gas connection this is called a back-pressure mixer as in Fig. 2. Fig. 3 is an ejector-type mixer which produces suction on the gas and tends to help the flow of gas. When either back pressure or suction is produced, these values are, in general, proportional to the air-pressure supply to the mixer and, indeed, must be so in order to function as a part of the air-gas-ratio control.

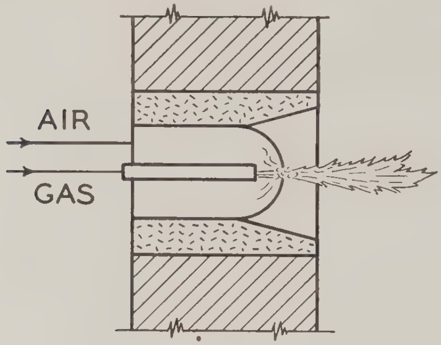

Fig. 2

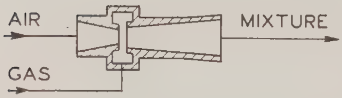

Fig. 3

If, instead of discharging the mixer into atmospheric pressure, it is discharged into a region of back pressure caused by the resistance to flow of a burner or caused by the combustion process itself, the back-pressure effect of a mixer will be increased, or the suction produced by an ejector mixer will be decreased. As long as the back-pressure effect which is downstream from the mixer is constant, it has no detrimental result on the air-gas ratio control, but if this back-pressure effect should become variable, then special allowances must be made in the ratio-control circuit in order to maintain air-gas ratio consistency. In fact, it will be shown how the difference between a constant and a variable back-pressure effect calls for a fundamental difference in the type of air-gas ratio control to be used.

In keeping with the two types of back-pressure effect from the downstream of the mixer, there correspond two classes of furnace-pressure conditions. The first class, a standard-pressure furnace, is defined as one which either has an atmospheric pressure or a back pressure which is in proportion to the mixture pressure being delivered to the furnace by one burner or a group of burners all under the same throttling control. The use of large flue areas and little or no stack effect will maintain zero pressure in the combustion chamber. If a stack is used, an automatic damper control may be installed to maintain zero pressure. Again, in the absence of stack effect, but with limited flue openings, a back pressure is created in the combustion chamber due to the combustion process; however, as long as there is only one throttling control, this back pressure will be in proportion to this throttling-control pressure. With the use of more than one throttling-control on one furnace there is a tendency for interference of one control upon another.

The other type of furnace-pressure condition is called a special pressure furnace and is defined as one which is not standard. This type of pressure is usually variable but may be constant at any value other than zero. The class of special pressures may be produced by the use of a stack, fan, or an extraneous throttling-control system discharging into the combustion chamber of the control group under discussion. Of course the two classes of furnace pressures are relative, not absolute, depending upon the amount of deviation of air-gas ratio that is allowed. In other words, a standard-pressure type of control may be used on an application with a small variation in pressure as long as the corresponding change in air-gas ratio is permissible.

Analysis

At this point it is advisable to review the fundamental gas-flow laws before attempting to analyze the various control circuits. These laws of low-pressure gas flow or noncompressible flow are not exact but are close enough for commercial application. The following terms will be used in the analysis:

Q = volume rate of flow
P = gage pressure
S = specific gravity
R = resistance to flow
A = general bleed constant
K = mixer constant
B, C, D, F, G, H, L, M = general constants

Subscripts:

u refers to upstream
d refers to downstream
1 refers to air stream
2 refers to fuel-gas stream
3 refers to air-gas mixture stream
4 refers to combustion chamber
v refers to bleed valve
o refers to bleed orifice

Any consistent set of units may be used.

The first law of gas flow states that the volume rate of flow of gas (or weight rate of flow if the gas is considered incompressible) is proportional to the square root of the pressure drop or pressure differential across any resistance to flow, such as an orifice plate, Venturi, valve, pipe fitting, etc. This is a common law applied in the metering of gas flow. It is also applied in controlling air-gas ratio. A set of differential pressures indicating air flow and another set of differential pressures indicating gas flow are made to oppose each other through various arrangements of diaphragms.[2] Thus a balance of differential pressures is maintained, which results in a constant air-gas ratio. While it is easily understood how control of the air-gas ratio is obtained using differential pressures, it is much more difficult to understand the control of air-gas ratio by single impulse pressures when used in connection with the various types of mixers and furnace-pressure conditions as heretofore described. If the downstream of an orifice plate is discharged to a known pressure region such as the atmosphere, the upstream pressure tap will indicate flow by itself, since the downstream pressure is known without measurement. In this case the upstream pressure tap becomes the single impulse pressure, by means of which the air-gas ratio must be controlled.

The other gas-flow law is that the volume rate of flow is inversely proportional to the square root of the specific gravity of the gas. Combining the two gas laws

$$Q \sim \sqrt{\frac{P_u - P_d}{S}}$$

[2] "Multiple-Diaphragm Pressure Regulators and Their Application," by H. C. McRae, *Instruments*, vol. 17, September, 1944, pp. 529–531.

It is convenient to express the constant of proportionality as $\sqrt{1/R}$, since the laws have the square-root form and the resistance concept to flow is similar to that used in the electrical, magnetic, thermal, and stress fields. Then, the general law of low-pressure gas flow is

$$Q = \sqrt{\frac{P_u - P_d}{RS}}$$

Any set of units may be used, the resistance unit being defined from the other units. From analogy to the electrical field, resistances may be combined[3] in series and in parallel. Likewise, both of Kirchhoff's laws of electrical networks may be used for gas-flow circuits; namely, (1) the algebraic sum of the gas flows in all the pipes that meet at a point is zero and (2) the total change of pressure around any closed circuit is zero. The resistance form of gas flow and Kirchhoff's first law will be used in the following analysis to prove that the various control circuits do maintain a constant air-gas ratio for variable flow rates. Of course in practice the various control circuits were derived from the analysis.

Although there are several types of control circuits for air-gas ratio control, using single impulse pressures, all of them are derived from the basic form shown in Fig. 4, which is called the

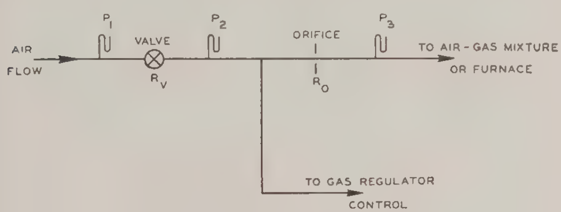

FIG. 4

"general bleed." It must be understood that the bleed mechanism or any of its derivatives is only a small branch of $^1/_4$-in. or $^3/_8$-in. pipe size, going from the main air pipe and usually some point of lesser pressure such as the air-gas mixture pipe, the combustion chamber, or the atmosphere, to the gas-pressure regulator control. Before the complete control circuits can be analyzed it is necessary to develop the following mathematical expressions for the general bleed and other bleed systems.

In Fig. 4 the pipe line going to the gas-regulator control is dead-ended; that is, there is no steady flow of air in this line. Therefore the flow rate through the valve must be equal to the flow rate through the small orifice

$$\sqrt{\frac{P_1 - P_2}{R_v S_1}} = \sqrt{\frac{P_2 - P_3}{R_0 S_1}}$$

Solving this equation

$$P_2 = \left(\frac{R_0}{R_v + R_0}\right) P_1 + \left(\frac{R_v}{R_v + R_0}\right) P_3$$

Let

$$\frac{R_0}{R_v + R_0} = A$$

and, therefore

$$\frac{R_v}{R_v + R_0} = 1 - A$$

[3] "Low Pressure Gas Flow Analyzing by a Resistance Theory," by H. C. McRae, *Industrial Gas*, vol. 19, March, 1941, pp. 17–20.

Rewriting the value of the gas controlling pressure

$$P_2 = AP_1 + (1 - A)P_3 \ldots\ldots\ldots\ldots\ldots [1]$$

where $0 \leq A \leq 1$

Equation [1] is the mathematical expression for the general bleed, stating that P_2 is always a fixed ratio difference between P_1 and P_3, regardless of the rate of flow or how P_1 and P_3 may vary during the control cycle. Since the bleed valve is set once and for all at the beginning of an installation to conform to the particular requirements of the application, R_v and therefore A are constant in the control cycle.

The various types of control are derived from Equation [1] by fixing definite values for A and P_3. For the conditions

$$P_2 = AP_1 + (1 - A)P_3 \ldots\ldots\ldots\ldots\ldots [2]$$

$$0 < A < 1 \quad \text{and } P_3 \neq 0$$

a double connection is required to control the gas (both to P_1 and P_3) and hence the name, "double bleed." If

$$A = 0 \quad \text{and} \quad P_3 \neq 0$$

$$P_2 = P_3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots [3]$$

For this case the bleed valve is closed or $R_v = \infty$. In practice the bleed valve and orifice are not used, since it is necessary to use only a single pipe to connect the air-gas mixture pressure, P_3, to the gas-regulator pressure, P_2. This class of control is called the "reduced cross connection."

The gas-control pressure may also be connected to the combustion chamber, in which case

$$P_2 = P_4 \ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

This type of control is called "cross connection to furnace." In fact, if the mixer is mounted directly on the furnace and the air is used as secondary air, the mixture pressure, P_3, becomes the furnace pressure, P_4. In other words, this case is derived from the previous case.

Again, for the condition

$$P_2 = AP_1, P_3 = 0, 0 < A < 1 \ldots\ldots\ldots\ldots [5]$$

the bleed orifice is discharged directly into the atmosphere. This case is called "single bleed" or abbreviated to simply "bleed."

In addition, if $A = 0$ and $P_3 = 0$

$$P_2 = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots [6]$$

This "zero gas" is the most commonly used type of control; the gas-control regulator need only be vented to the atmosphere and no pipe interconnections between the air and gas are necessary.

The last type of control is derived either from the single or the double bleed, since P_3 may or may not be zero, because $A = 1$

$$P_2 = P_1 \ldots\ldots\ldots\ldots\ldots\ldots\ldots [7]$$

This is called "straight cross connection." For this case the bleed valve is open wide with no resistance; $R_v = 0$. Again, in practice, this is equivalent to a single pipe connection between the air pressure and the gas-control pressure.

The foregoing control classifications give all the necessary combinations in air-gas ratio control by single impulse pressures to cope with the various types of mixers, gas pressures available, and furnace-pressure conditions. There are a few other miscellaneous control types, such as connection of the gas-control pressure to the air and to a gas-pressure point just upstream from the mixer, but these controls only duplicate, in need, what has already been derived. Table 1 is constructed for convenience to summarize the various controls and show their relationship. It will

TABLE 1 SUMMARY OF VARIOUS CONTROLS AND RELATIONSHIP

General bleed
Equation [1]
$P_2 = AP_1 + (1 - A)P_3$
$0 \leqq A \leqq 1$

Double-bleed
Equation [2]
$P_2 = AP_1 + (1 - A)P_3$
$0 < A < 1 \quad P_3 \neq 0$

Straight cross connection
Equation [7]
$P_2 = P_1$
$A = 1$

Single-bleed
Equation [5]
$P_2 = AP_1$
$0 < A < 1 \quad P_3 = 0$

Reduced cross connection
Equation [3]
$P_2 = P_3$
$A = 0 \quad P_3 \neq 0$

Zero gas
Equation [6]
$P_2 = 0$
$A = 0 \quad P_3 = 0$

Cross connection to furnace
Equation [4]
$P_2 = P_4$
$A = 0 \quad P_4 \neq 0$

be shown later how the branch on the right side is used for standard-pressure furnaces, while the left branch is used for special pressure furnaces.

Fig. 5 shows the general arrangement of the air and gas piping with the exception of the bleed control itself. As previously mentioned, the mixer and the burner may be only one unit instead of two separate units as shown. It is important to notice the function of the gas-pressure-regulator control. This device is sometimes called a "zero governor" or "atmospheric regulator." It is nothing more than a diaphragm-operated pressure regulator[4] which controls the gas-discharge pressure to be equal to the impulse pressure imposed from the bleed control, providing of course that the gas-supply pressure is high enough. The control valve is adjusted according to heat-input demand. Thus the air pressure, P_1, and air-flow rate are being adjusted continuously, while likewise the gas-flow rate must remain in step with the air flow through the application of the proper bleed control and the gas regulator.

[4] "Simple Pressure Regulators—Principal Types and Their Characteristics," by H. C. McRae, *Instruments*, vol. 15, February, 1942, pp. 44–45 and 59.

Fig. 6 shows the diagrammatic layout of the circuit in Fig. 5; R_1 represents the resistance to flow of air in the mixer and R_2 is the resistance to gas flow in the mixer, including a gas valve for air-gas ratio adjustment. Term R_3 is the resistance of the burner. The expression, $K_2\ (P_1 - P_3)$, is the suction or back-pressure effect of the air on the gas flow as discussed under the subject of mixer classification. The corresponding effect of the gas on the air flow is $K_1(P_2 - P_3)$. The value of K_2 determines the mixer classification: For $K_2 = 0$, a neutral mixer; for $K_2 < 0$, a back-pressure mixer; and for $K_2 > 0$, the mixer is the ejector type.

The air flow through the resistance, R_1, is

$$Q_1 = \sqrt{\frac{P_1 + K_1(P_2 - P_3) - P_3}{R_1S_1}}$$

The gas flow rate is

$$Q_2 = \sqrt{\frac{P_2 + K_2(P_1 - P_3) - P_3}{R_2S_2}}$$

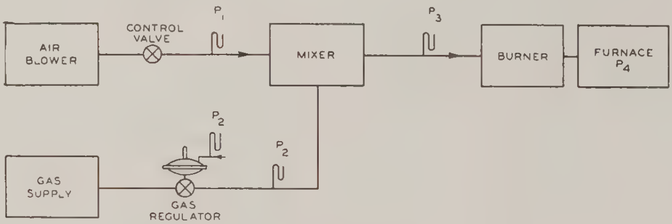

FIG. 5

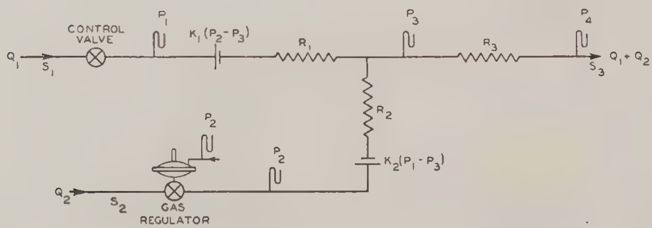

FIG. 6

For the experimental determination of K_2, the gas flow is cut off. Then $Q_2 = 0$ and

$$P_2 + K_2(P_1 - P_3) - P_3 = 0$$

$$K_2 = -\frac{P_2 - P_3}{P_1 - P_3} \text{ (no gas flowing)}$$

In a similar manner, with the air cut off

$$K_1 = -\frac{P_1 - P_3}{P_2 - P_3} \text{ (no air flowing)}$$

In the case that the mixer and the burner become one unit, combustion may affect the K values. For this case the K values will have to be determined under actual combustion and by metering the air- and gas-flow rates; however, P_3 is replaced by P_4.

From the foregoing air- and gas-flow rates the air-gas ratio is

$$\frac{Q_1}{Q_2} = \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left[\frac{P_1 + K_1(P_2 - P_3) - P_3}{P_2 + K_2(P_1 - P_3) - P_3}\right]} \quad \ldots\ldots [8]$$

Since R_1, R_2, S_1, S_2, K_1, and K_2 are constants, it is only necessary to eliminate the pressure terms, which vary in the control cycle, to prove that the air-gas ratio is constant.

Taking a simple case to illustrate the method of proving air-gas-ratio constant, consider a neutral-type mixer discharging into a zero-pressure furnace. In this case, $K_1 = K_2 = P_3 = 0$. Equation [8] becomes

$$\frac{Q_1}{Q_2} = \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left(\frac{P_1}{P_2}\right)}$$

Let $P_2 = AP_1$ be the control chosen, which is a single bleed or straight cross-connection for $A = 1$. After substituting this value for P_2

$$\frac{Q_1}{Q_2} = \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left(\frac{1}{A}\right)} \text{ const}$$

Consider a standard-pressure furnace for which $P_4 = 0$ or in the case that P_4 is proportional to P_3, the furnace resistance to flow may be included in the burner resistance and the flue gases are discharged from the furnace to the atmosphere or zero pressure anyway. In other words, for standard-pressure furnaces the downstream resistance from the mixer is constant. Then, for this case: $P_4 = 0$ and R_1, R_2, R_3, S_1, S_2, S_3 are constant. The control chosen is $P_2 = AP_1$ which is the single-bleed, straight cross connection for $A = 1$, or zero gas for $A = 0$. The pressure, P_3, must be proved to be proportional to P_1, before Equation [8] may be used. Equating the sum of the air and gas flow to be equal to the mixture flow

$$\sqrt{\frac{P_1 + K_1(P_2 - P_3) - P_3}{R_1S_1}} + \sqrt{\frac{P_2 + K_2(P_1 - P_3) - P_3}{R_2S_2}} = \sqrt{\frac{P_3 - P_4}{R_3S_3}}$$

Using the values just given for P_2 and P_4

$$\sqrt{\frac{P_1 - P_3 + AK_1P_1 - K_1P_3}{R_1S_1}} + \sqrt{\frac{AP_1 - P_3 + K_2P_1 - K_2P_3}{R_2S_2}} = \sqrt{\frac{P_3}{R_3S_3}}$$

This equation may be written in more simple form by the use of general constants, since the actual value of P_3 is not needed

$$\sqrt{FP_1 + GP_3} + \sqrt{BP_1 + DP_3} = \sqrt{HP_3}$$

After squaring this equation twice and combining the proper terms in order to eliminate the radicals

$$[(F + B)P_1 + (G + D - H)P_3]^2 = 4[BFP_1^2 + (DF + BG)P_1P_3 + DGP_3^2]$$

Again, simplifying by using general constants

$$P_3^2 + LP_1P_3 + MP_1^2 = 0$$

or

$$P_3 = \frac{-LP_1 \pm \sqrt{L^2P_1^2 - 4MP_1^2}}{2} = \left[-\frac{L}{2} \pm \frac{\sqrt{L^2 - 4M}}{2}\right]P_1$$

and finally

$$P_3 = CP_1$$

Now Equation [8] becomes

$$\frac{Q_1}{Q_2} = \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left[\frac{1 + AK_1 - CK_1 - C}{A + K_2 - CK_2 - C}\right]}$$

which is constant and independent of the control pressure P_1.

Since it is proved that zero gas produces constant air-gas ratio for standard-pressure furnaces or zero-pressure furnaces, this case may be easily extended to prove that cross connection to furnace holds constant air-gas ratio for special pressure furnaces. These two cases become equivalent when all the pressures in the latter case are referred to P_4, the furnace pressure, as a base instead of the atmosphere or gage pressure. In other words, since $P_2 = P_4$, P_2 will always be zero if referred to its value exceeding P_4. Likewise, P_1 and P_3 take on new values that differ from their old gage values by the same amount that P_4 differs from zero gage pressure.

Now consider the general case of special-pressure furnaces. Both P_4 and R_3 become variable for this case, although R_3, the burner resistance, is not usually varied purposely but may vary slightly with temperature change. P_3 is no longer proportional to P_1 and therefore the single-bleed will not hold constant air-gas ratio. Choose

$$P_2 = AP_1 + (1 - A)P_3$$

the double bleed control, the reduced cross connection for $A = 0$, or the straight cross connection for $A = 1$. Equation [8] becomes

TABLE 2 TYPE OF CONTROL FOR GIVEN APPLICATION

Mixer classification	Gas pressure available	Standard pressure furnaces	Special pressure furnaces
Ejector type	Low	Zero gas	Cross connection to furnace
	Medium	Zero gas	Cross connection to furnace
		Single bleed	Reduced cross connection
			Double bleed
	High	Zero gas	Straight cross connection
		Straight cross connection	
Neutral or back-pressure type	Low	No method available	No method available
	Medium	Single bleed	Double bleed
	High	Straight cross connection	Straight cross connection

$$\frac{Q_1}{Q_2} = \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left[\frac{(P_1 - P_3) + K_1(AP_1 + P_3 - AP_3 - P_3)}{AP_1 + P_3 - AP_3 - P_3 + K_2(P_1 - P_3)}\right]}$$

$$= \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left[\frac{(P_1 - P_3) + AK_1(P_1 - P_3)}{A(P_1 - P_3) + K_2(P_1 - P_3)}\right]}$$

$$= \sqrt{\left(\frac{R_2S_2}{R_1S_1}\right)\left[\frac{1 + AK_1}{A + K_2}\right]} \text{ const}$$

Table 2 shows the type of control to use on any given application. The use of the single bleed and the double bleed have been avoided when possible, although these types of control sometimes have to be used to obtain a sufficient quantity of gas. The reason for avoiding these controls is because of the necessity of adjusting the bleed valve properly to get just the sufficient amount of gas pressure without exceeding the gas-supply pressure minus the pressure drop through the gas regulator. All of the other controls require only a single pipe connection to the gas regulator without any adjustments. The zero gas system is the most popular, because in this case no connection need be made to the gas regulator, and there is always a sufficient supply pressure for this control.

Figs. 7 to 11, inclusive, show the complete circuit for some of the most frequent applications.

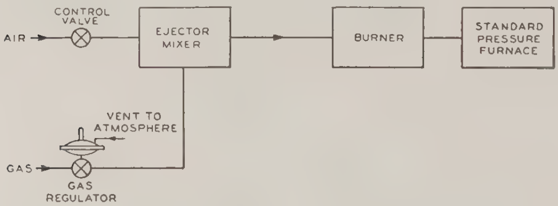

Fig. 7 Zero Gas

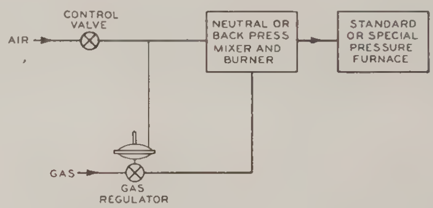

Fig. 9 Straight Cross Connection

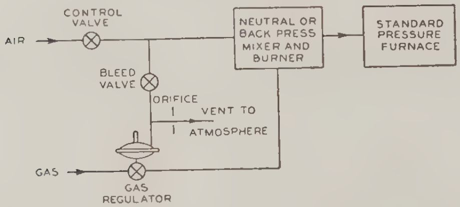

Fig. 8 Single Bleed

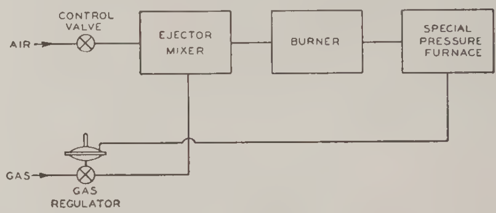

Fig. 10 Cross Connection to Furnace

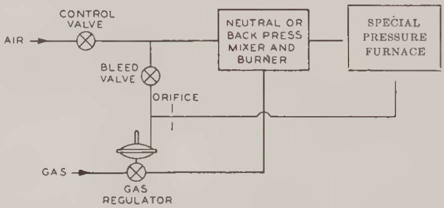

Fig. 11 Double Bleed

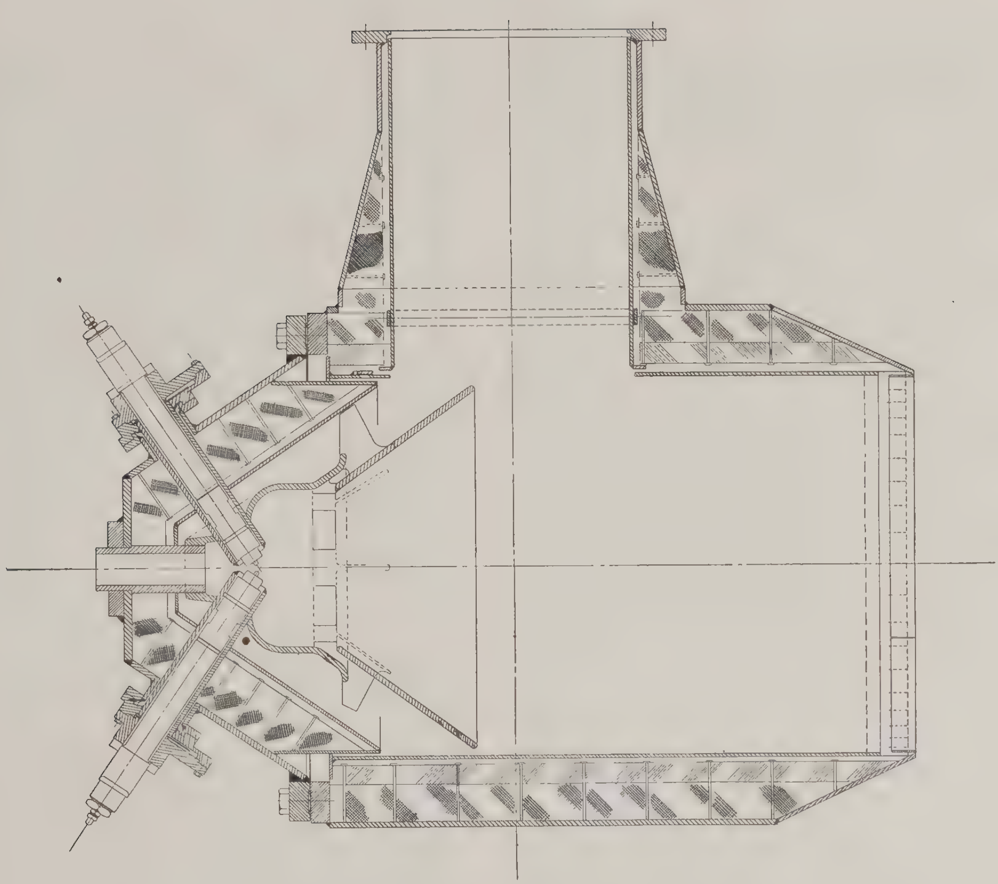

Fig. 1 Elbow Combustion Chamber

The Elbow Combustion Chamber

By M. A. Mayers[1] and W. W. Carter,[2] Jeannette, Pa.

A description is given of the "elbow" combustion chamber, developed for use in the gas-turbine power plant designed and built for the United States Navy by the Elliott Company. This type provides extremely high heat-release rates which is an essential characteristic for gas-turbine applications. The fuel-oil-burning system which utilizes standard Diesel fuel-injection equipment has been developed to a high point of efficiency, and may be varied over a range from full load to 1/10 full load without material change in quality of atomization. The paper discusses in detail the air-flow characteristics, fuel supply, ignition and control devices, and analyzes the combustion-chamber performance-test data.

THE "elbow" combustion chamber was developed for use in the gas-turbine power plant designed and built for the U. S. Navy by the Elliott Company during the period 1942–1944. It has turned out to be exceptionally adapted to gas-turbine use in that it permits heat-release rates of 2.5×10^6 Btu per cu ft per hr, with air supplied at 95 psia, and 650 F, when it requires a pressure drop of only 16 in. water column; it permits a "turndown" to $^1/_{20}$ of full load; and is extremely insensitive to fuel-air ratio. As shown in Fig. 1, it consists of a right-angle elbow in a duct, of which the upstream arm is smaller in diameter than the downstream one. The upstream end of the large arm which forms the combustion chamber proper, is closed by a head which carries an "ignition cone," the fuel-oil burners, an igniter burner, and the "flame-rod" or contact member of the flame-sensitive controller. The chamber contains no inner shell separating combustion air from tempering air, and all the air in the gas-turbine circuit is supplied to the chamber without the

[1] Formerly Development Engineer, Elliott Company. Mem. A.S.M.E.

[2] Engineering Research and Development Department, Elliott Company. Jun. A.S.M.E.

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necessity for separate control of combustion air. Fuel is supplied to the chamber through standard Diesel fuel-injection equipment, producing a pulsating fuel flow which on a single burner may be varied over the range from full load down to $^1/_{10}$ full load without material change in the quality of the atomization. The rate of fuel flow is controlled by adjusting the position of the control rod of the Diesel injection pump.

Gas-Turbine Combustion-Chamber Requirements

The Elliott gas turbine was designed from the basic premise that such an engine, to be a successful prime mover, must have an efficiency comparable with those of others with which it would compete. To accomplish this aim, it was necessary to provide heating of the working medium in two stages, one before the first stage of expansion and a reheat after this; and to reduce the parasitic losses associated with the combustion chambers to a minimum. The thermodynamic advantages to be gained by reheating have been treated elsewhere (1)[3] and need not be repeated; the process does, however, entail the necessity of supplying heat to a flue gas, resulting from the combustion in the initial, or high-pressure combustion chamber, which may contain 2.7 per cent carbon dioxide, 2.6 per cent water vapor, and whose oxygen concentration may be reduced to about 16.5 per cent.

It may not at first be apparent why the pressure drop in the combustion chambers, as well as in the ducts and heat exchangers, must be so severely limited. The loss in gross turbine work is proportional to the ratio of the pressure drop to the absolute pressure at which it occurs and thus does not appear to be serious. Only a portion of the gross turbine work appears as useful output, this fraction being represented by the work ratio, α (1). The proportional loss of net turbine output due to parasitic losses is inversely proportional to α and may be therefore from 2.5 to 4 times as great as the loss in gross turbine work.

Since this machine was designed for naval ship propulsion, it was necessary for it to operate over a wide range of loads. The cubic relation between speed and power in ship propulsion, coupled with the fact that the gas turbine is governed directly by the rate of heat input, demands a high degree of flexibility in the fuel-burning system. Fig. 2 shows how the fuel-burning rate, and the power output depend upon the propeller-shaft speed. For maneuvering at speeds down to 45 per cent of full-power speed the high-pressure chamber must operate stably over a range of burning rates of 10 to 1; while for the design conditions of speeds down to $^1/_3$ of full the turndown must approach 20:1. Conventional marine practice of changing the tips or spray plates of fuel burners is not acceptable for this application, since burner removal from chambers operating under pressures greatly exceeding atmospheric could hardly be made rapid enough.

Associated with the need for wide-range characteristics of the fuel-burning equipment is the necessity that it be insensitive to over-all excess air. It is evident from Fig. 2 that the air-to-fuel ratio covers an extreme range from 600:1 in the low-pressure chamber to about 120:1 in the high-pressure chamber. In addition, variations in operating conditions such as weather, draft of vessel, etc., might impose variations about the curve of air-to-fuel ratio, at any value of speed, of 10–25 per cent. Previous chambers (2) had used separate control of combustion air to meet this type of difficulty. In the present instance this solution was unsuitable because air was supplied to the chambers at such high temperatures that valves or dampers could hardly be expected to operate reliably, and because of the stringency of pressure-drop restrictions.

It is desirable that the combustion chambers for gas-turbine plants permit very high heat-release rates; in fact, the possibility that this may be secured is one of the main attractions of the internal-combustion gas turbine. It was, moreover, believed to be essential that these combustion chambers operate with a perfectly clean discharge, as the effect of soot, smoke, or gaseous products of incomplete combustion on the turbines and heat-transfer surfaces could not be predicted. The requirement of clean discharge also made it undesirable to leave exposed surfaces of nonmetallic refractories in the combustion chamber, since spalling or dusting of such material would foul the gas stream. Of course, safety, reliability, and convenience in operation, as well as ease and accessibility for necessary repairs are also desirable characteristics.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

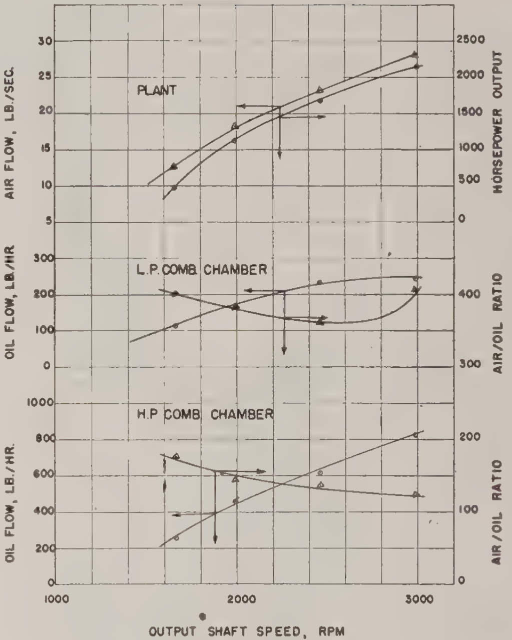

Fig. 2 Dependence of Combustion-Chamber Performance on Output Shaft Speed

Characteristics of Elbow-Type Combustion Chamber

Air Flow. The air, on entering the main combustion chamber through the side inlet, follows the flow pattern characteristic of flow through an elbow in a pipe line (3), producing a double-vortex system. The two vortexes lie on either side of the center line of the inlet duct and have opposite senses of rotation. When the chamber is in operation the centers of these vortices appear as two bright spots to an observer looking upstream from the outlet of the chamber.

The strength of the vortex system is enhanced by making the diameter of the inlet duct less than that of the main chamber. While exact limits of this ratio have not yet been established, it has been found by model tests that the vortexes become less stable and show more tendency to break down into a single vortex having relatively undisturbed laminar flow, when the ratio of the diameters becomes greater than 0.6. The same kind of breakdown is produced by eccentricity of the inlet, or departure of the inlet center line from an extension of a radius of the combustion chamber proper.

A portion of the air entering the chamber is trapped behind the edge of the ignition cone which projects across the inlet air stream. Some of this air flows into the cylindrical part of the ignition cone through the slots shown in Fig. 1, producing a ring

vortex which has been shown to stabilize the ignition of fuel sprays (4), by some such mechanism as the following: Air from the slots in the ignition cone, on meeting the back wall of the ignition cone, turns radially outward, causing the formation of a vortex ring whose sense of rotation is such that the flow through the center of the ring is upstream. This causes the ring to take up a fixed position in relation to the back wall into and out of which air passes without affecting the stability of the vortex system as a whole (5). Thus the vortex ring is similar to a smoke ring, except that it is fixed in position and is continuously replenished. The entering air which has swept over the fuel spray injected into the chamber, carries with it a portion of finely atomized oil and oil vapor which serves as fuel for a continuously regenerated pilot flame established in the vortex ring.

The pilot flame formed by the vortex ring probably serves as a source of heat and of "active centers" to initiate the combustion of the main oil spray with the remainder of the air flowing through the ignition cone. Under conditions of very low oil-air ratio this small amount of air may be sufficient to burn the oil completely within the ignition cone, itself; but at higher loads, this protected region serves only as a space within which the whole mass of fuel may be vaporized and ignited. It should be noted that no rotation about the axis of the chamber is imparted to the flow through this cone. In every instance in which such rotation was produced, drops of burning oil were thrown against the walls of the cone, which eventually produced coke deposits.

The mass of ignited fuel issuing from the ignition cone appears to flow into the centers of the two vortexes formed by the main air flow, in which their combustion is completed. Observation of the bright spots formed by the double-vortex system shows that their size depends on the load, or rate of burning oil. At medium loads, large enough so combustion is not complete within the ignition cone, the spots are quite small and are clearly separated. As the load increases they increase in size until the two spots nearly touch. One might imagine that the flow out of the cone is directed into the centers of the vortices and that the burning fluid diffuses radially out from these centers to an extent determined by the amount of air required to complete its combustion. It has been suggested (6) that the reduction in static pressure at the center of a vortex (7) may be the agency which produces such a directed flow.

That some mechanism, such as that just suggested, governs the operation of the chamber can scarcely be doubted. At no time has there been evidence of incomplete combustion such as would result from the quenching of the burning by too great an amount of excess air, even though the chambers have been operated at air-to-oil ratios as great as 600:1. Moreover, the temperature of the flame, estimated from visual observation of its color, must be of the order of 3000 F, a value associated with combustion with only a small excess of air. Another observation, which helps to show that only the portion of the air that penetrates nearly to the center of the vortexes takes part in the combustion of the fuel, is that the air in the outer filaments of the vortex flow, which sweeps the walls of the combustion chambers, remains cool in the sections of the chamber in which rapid combustion occurs and serves to cool the metallic walls. This portion of the air rises in temperature some distance downstream and is eventually mixed by the residual vortex motion with the combustion air proper.

This characteristic of the elbow combustion chamber makes it almost unique among high-duty chambers; other types, which depend upon partial admission of air into the combustion space vented through liner walls, thereby make the liner wall behave like a ported gas burner except that in this case flames of air in a fuel-rich atmosphere are set up. The proximity of these flames to the chamber wall leads, of course, to overheating of the liner. In the elbow chamber no such overheating or warping of the chamber liner has ever been observed; tests, described below, indicate that no point on the inner shell rises more than 300 deg F above the outlet gas temperature even though it is exposed to direct radiation from a dense flame.

Fuel Supply. Fuel is supplied to these combustion chambers by means of commercial types of Diesel injection equipment. Number 2 Diesel oil is pumped through wound cartridge filters made by Commercial Filter Corporation under the trade name "Fulflo," to the inlet of 8-cylinder pumps made by the American Bosch Corporation. The discharges of all eight cylinders of each pump are led through $^1/_8$-in. $\times$ $^1/_4$-in. hydraulic tubing to a high-pressure manifold in which the lines are brought together. From here they pass through $^1/_4$-in. $\times$ $^1/_2$-in. hydraulic tubing, to a single water-cooled fuel injector, of a standard design produced by Aircraft and Diesel Equipment Corporation. Thus each pump, header, and injector combination forms a single burner unit.

The injection pumps have cylinder bores of 10 mm and actual plunger strokes of 10 mm. The plungers are cam-actuated so as to have essentially constant velocity over the effective or delivery part of the stroke, which can be varied by changing the point at which a relief port is uncovered by the plunger. Thus fuel is delivered at a constant rate during a short variable time interval. The volume of fuel per injection is directly proportional to the length of the time interval during which injection of fuel occurs and is controlled by the position of a pushrod which simultaneously fixes the effective stroke of each plunger. Since the maximum duration of injection from each pump cylinder is 26 deg of camshaft rotation, while the interval between the start of each succeeding injection is 45 deg, it is evident that the discharges do not overlap. The pumps are driven at a constant speed of 1200 rpm, producing a frequency of injection through each burner of 9600 per min, or 160 per sec. Thus the time interval between the start of each injection is 6 milliseconds, which is short enough to assure stable flame, without pulsations, when the fuel is discharged into the ignition cone previously described.

The Adeco injector mentioned previously, contains a spring-loaded differential-area valve which opens when the fuel-oil line pressure reaches 3300 psi. This valve insures clean sharp dis-

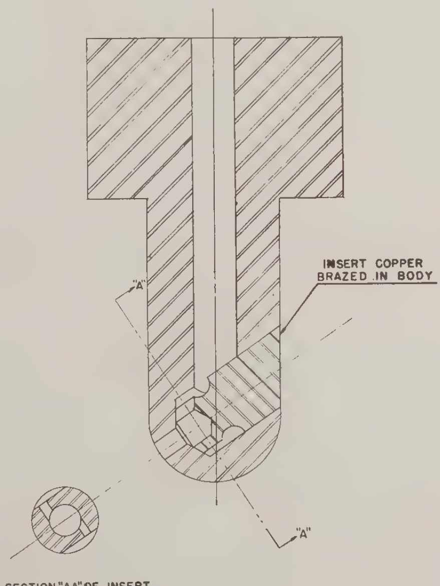

Fig. 3 Diesel Injector-Nozzle Tip

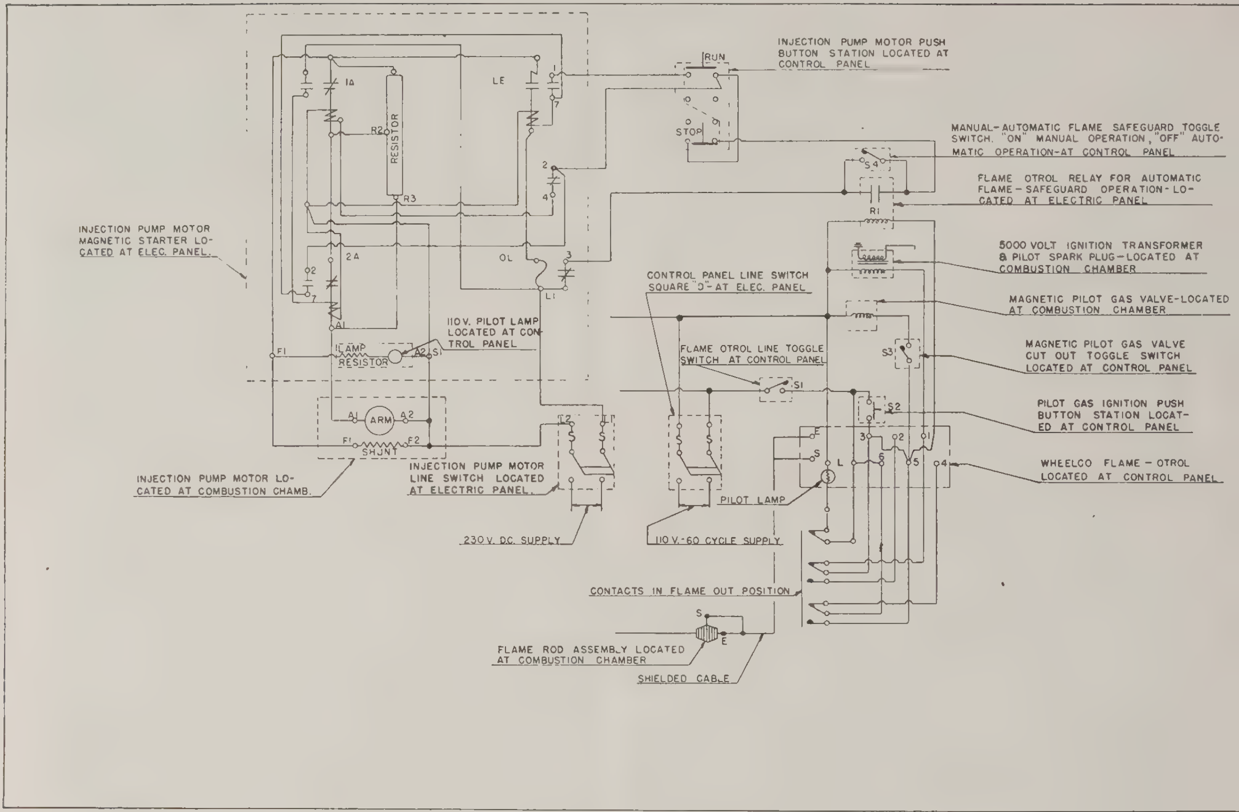

Fig. 4 Flame-Control Circuit

charges and prevents "dribbling" between pulses. The injector is equipped with nozzle tips whose construction is shown in Fig. 3. By this means the advantages of the fine atomization and uniform distribution produced by a whirling cone spray, used conventionally in large mechanical-atomizing oil-burner nozzles, were secured for the spray from a Diesel injector. The discharge orifice is set at an angle to the axis of the tip to permit inserting two injectors in the same ignition cone with their tips not more than 1 in. apart, and having them discharge axially down through the same cone. The tips are installed in the injector by means of a jig which insures the correctness of their orientation, while the injectors are held in the chamber by a fixture built into the burner support. Since fuel is delivered to the injector at a constant instantaneous rate during the injection interval, the atomizing energy is constant from maximum to minimum capacity and good atomization is obtained over the full range.

Ignition and Control Devices. The head of the chamber also carries a spark-ignited gas-fired igniter burner, and the support for the "Flame-rod" or sensing element of the controller, made by Wheelco Instrument Company, which insures that fuel will not be supplied to the chamber in the absence of a source of ignition. This device uses the electrical conductivity of the flame to balance a bridge circuit, allowing a relay to close. This relay may be connected, as shown in Fig. 4, to hold open a valve for gas supply or to close contactors supplying power to the injection pumps. If the conductivity of the circuit including the Flame-rod is either above or below that of the flame the relay opens, cutting off power from these operations, and lighting a signal lamp. The Flame-o-trol, or measuring and control element of this device, which is mounted on the main operating panel board of the gas-turbine unit, is by-passed at starting by depressing a push button which simultaneously opens the gas-supply valve to the igniter burner and applies a sparking voltage to the electrodes of the igniter.

The igniter burners are supplied with a propane-and-air mixture formed in a Venturi mixer; propane is supplied from commercial tanks. Air to the mixer, and cooling air to the Flame-rod holders is supplied by the high-pressure compressor of the gas-turbine set. A small portion of the discharge of this compressor is drawn off from the main discharge duct ahead of the regenerator, passed through a small aftercooler, and is supplied to these auxiliaries. The air- and gas-piping diagram of these devices is shown in Fig. 5.

Sight glasses are mounted on the combustion chambers at convenient points to permit observation of the flame. Those installed in these chambers do not, unfortunately, permit removal of the glass while the chamber is under pressure, but the sight glasses designed for the future will. To date, no glass has ever broken in service.

Materials and Construction. The inner shell of the chamber, which acts merely as a shield to prevent abrasion of the internal insulation by the gas stream and takes no pressure load, is of 25 per cent Cr, 20 per cent Ni, type No. 310 steel. The inner shell is supported centrally in the chamber by means of two "spoked" supports which permit radial expansion of the shell without altering the center-line position. The shells are fixed axially near the air-inlet position and are free to expand in both directions from this point. Flexible seals to prevent pieces of insu-

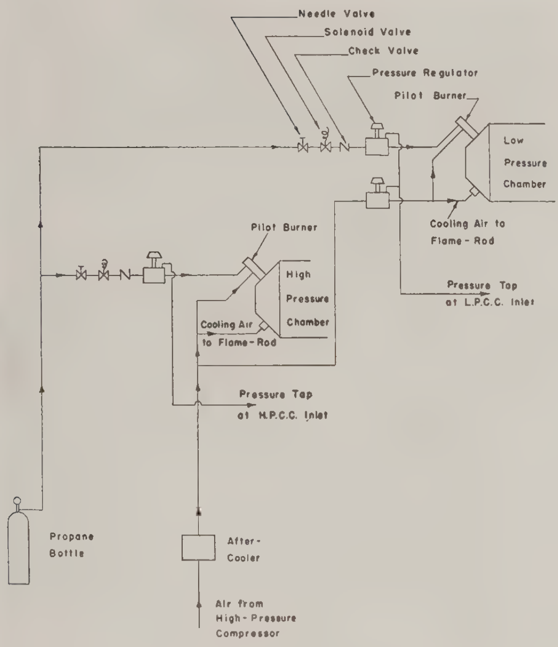

FIG. 5 AIR AND GAS PIPING TO IGNITERS AND FLAME RODS

lation getting into the gas stream are installed at either end of the inner shell.

The ignition cone is cast of chrome iron containing a minimum of 28 per cent Cr. The burner holders are of No. 302 stainless steel. Except for the insulation, which occupies the space between the inner and outer shells, these are the only parts exposed to extreme temperatures.

The internal insulation consists of two layers; the inside, which is exposed to the highest temperature, being of a lightweight brick $1^1/_2$ in. thick. The outer layer is a slag-wool block, wired and cemented to the outer shell. In the high-pressure chamber it was desired to minimize the thermal expansion of the structural member of the combustion chamber, so the block insulation is 2 in. thick and the outer pressure-retaining shell is of mild steel. In the low-pressure chamber some thermal expansion was desired, intermediate between the amount that would be caused by exposure of the shell to gas temperature and that of a "cold" shell. One inch of block insulation was therefore placed inside the shell and $1^1/_2$ in. outside, and the outer shell was made of 18 Cr, 8 Ni, type No. 302 stainless steel.

The heads for both chambers have outer shells of mild steel, inner shells of type No. 310 and are insulated between the shells. They are bolted to the combustion chamber proper by studs tapped into the combustion-chamber flange. The joint is sealed by a double-ring copper-wire gasket. Since the head of the chamber carries all the internal elements, these are immediately accessible for repair or replacement on its removal.

COMBUSTION-CHAMBER PERFORMANCE

An extensive series of tests was made on an experimental combustion chamber, a duplicate in its principal dimensions to the high-pressure combustion chamber. The chamber could be supplied at a maximum pressure of 8 psig, with 5600 cfm of air which could be heated to temperatures up to 1000 F, by burning natural gas in it in a special furnace. The maximum weight flow of air was only about one fifth of that in the gas-turbine set, and only one burner-injection pump unit was installed. It was anticipated, however, that these tests would permit determining if the performance of the combustion chambers would limit the plant operation, as a wider range of air-oil ratios could be tested than would be used in the plant, and shorter burning times could be obtained, since larger volume flows and higher velocities could be produced by lowering the pressure. It was expected that the fourfold increase in pressure in the prototype would permit increasing the heat-release rate by more than the factor of 3 between that of the prototype and that reached in the tests.

These tests were aimed at the determination of the operating efficiency of the combustion chamber. For a system in which the flue-gas discharge temperature is far above the air inlet temperature, it is not immediately obvious how the thermal efficiency should be defined. One possibility is to compare the higher heating value of the fuel with the difference between the enthalpy of all of the products of combustion and that of the air and fuel supplied. The first term in this difference is subject to some error because of uncertainty in the actual amount of moisture in the flue gas, but the results calculated in this manner have been presented in Table 1, under the column heading "Efficiency by Heat-Balance Method." Another method of expressing the efficiency, derivable from the same heat-balance equation, results in a comparison of the change in enthalpy of the air supplied to the chamber with a quantity defined by chemists (8) as the "heating value at the temperature" of the combustion-chamber discharge. The combustion-chamber efficiency thus defined may be expressed

$$\eta = \frac{A}{F}\,\frac{i_2 - i_1}{H_2} \qquad [1]$$

TABLE 1 TEST RESULTS FROM EXPERIMENTAL COMBUSTION CHAMBER

Run Number	Inlet Temp. (°F)	Outlet Temp. (°F.)	Inlet Pres. (Psia.)	Air Rate (#/Sec.)	Oil Rate (#/Hr.)	Heat Input in Million Btu./Hr.	Air/Oil Ratio	Static Pres. (Psi.)	Eff.-Bu. Ships Method	Eff.-Ht. Bal.
29	498	1050	20.1	6.04	180.5	3.55	120.4	0.63	96.0	97.6
30	497	1258	20.3	6.00	245.0	4.81	88.2	0.71	99.3	99.9
31	130	860	20.0	5.97	217.5	4.28	98.7	0.62	97.1	99.2
32	130	890	18.8	5.95	226,5	4.45	94.6	0.64	99.6	99.2
33	498	1049	19.7	6.12	185.5	3.65	118.8	0.70	94.8	96.0
34	499	1063	19.7	6.14	191.5	3.76	115.3	0.66	94.3	94.7
35	497	1250	19.8	5.99	250.5	4.92	86.1	0.73	95.8	95.6
36a	117	259	16.8	6.19	51.0	1.00	437.2	0.42	83.2	85.6
36b	103	288	15.2	4.00	43.5	.86	331.2	0.21	81.0	80.9
37	1003	1254	20.7	6.18	100.5	1.97	221.4	0.61	86.0	87.3
38	505	1035	19.6	6.13	174.5	3.43	126.5	0.66	96.9	96.3
39	498	1039	19.6	6.15	175.5	3.50	126.2	0.66	98.7	98.3
40	116	290	16.6	5.91	58.5	1.15	363.9	0.42	83.4	----
41	122	1035	18.5	5.64	262.5	5.16	76.9	0.68	97.9	----
42	475	996	18.9	5.85	166.5	3.27	126.5	0.69	93.2	----
43	121	393	17.0	5.92	80.5	1.58	264.7	0.53	95.2	----
44	487	1041	19.1	5.86	178.5	3.51	118.2	0.75	93.0	----
45	465	989	18.9	5.87	167.5	3.29	126.2	0.72	94.3	95.4
46	480	995	19.1	5.92	166.5	3.27	128.0	0.77	94.2	95.9
47	134	1365	18.8	5.23	330.5	6.50	57.0	0.75	101.1	102.7

where η is combustion-chamber efficiency

$\frac{A}{F}$ is air-fuel ratio at which chamber is operated

i_2, i_1 are enthalpies of air at discharge and inlet conditions, respectively

H_2 is heating value of fuel at temperature of combustion-chamber discharge and is defined by

$$H_2 = H_0 - [w_c(i_{C_2} - i_{C_0}) + w_H(i_{H_2} - i_{H_0}) - w_0(i_{0_2} - i_{0_0})] \quad [2]$$

where H_0 is heating value of fuel determined under standard conditions

w_C, w_H, w_0 are weights of carbon dioxide, water, and oxygen entering the reaction per unit weight of fuel

i_{C_2}, i_{C_0}, etc., are enthalpies of constituents indicated by letter subscripts at the condition of discharge from the chamber and standard condition, respectively.

This efficiency is entered in Table 1, under the column heading "Efficiency by Bureau of Ships Method" (9), and is plotted in Fig. 6.

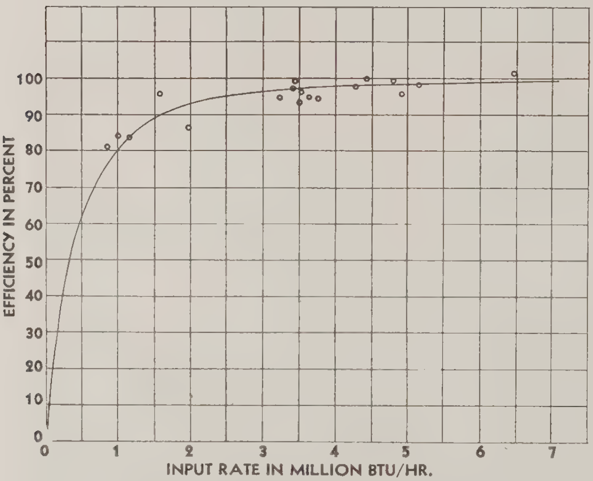

Fig. 6 Combustion-Chamber Efficiencies at Various Loads

Instrumentation. Both methods of determining efficiency require accurate measurement of the temperature of the discharge from the combustion chamber. Two methods of measuring this temperature were used. One was similar to that used by King (10) in determining a reference temperature for the calibration of shielded thermocouples, while the other, which also permitted estimating the degree of stratification in the discharge temperature, was based on the average of the indications of three shielded thermocouples (11) placed in the combustion-chamber discharge. The average of the three thermocouples was always higher than the indication of the single reference couple, which could be accounted for by heat loss between the two temperature-measuring stations. Because stratification in the outlet duct cast some doubt on the validity of the average, and since there was a possibility that the temperatures of the shielded couples were influenced by catalytic combustion on the metallic surfaces of the shields, the reference couple was always used for efficiency estimates.

The pressure measurements used for estimation of the pressure drop across the chamber were the subject of rather intensive study. This characteristic of the chamber was of special importance, but was very small in magnitude so that great care was needed in its measurement. The static-pressure measurements in both the inlet and outlet ducts were checked, during cold-air runs, against the indications of a Fecheimer tube (12) to assure freedom from errors due to impact pressures.

Gas samples were drawn from both inlet and outlet ducts and were analyzed on an Ellison three-chamber gas analyzer of the Orsat type. This instrument could not be expected to indicate small concentrations of the products of incomplete combustion, which might, however, have a relatively large effect on the chamber efficiency. In order to permit the diagnosis of losses from this source, a combustible-gas analyzer, having a full-scale range equivalent to 1 per cent of carbon monoxide, was kindly lent us by the Mine Safety Appliances Company, Pittsburgh, Pa., and a series of tests (runs Nos. 40–44 inclusive) were run, covering the range of air-oil ratios, in which gas samples from the regular taps were conducted to the analyzer. As shown in Table 2, the maximum indication of combustible gas during the run was about 0.02 per cent, equivalent to about 0.6 per cent heat loss to incomplete combustion. For the first 10 to 15 min of operation of the chamber after lighting off, a decreasing indication, starting at a peak of about 0.06 per cent equivalent carbon monoxide, was observed. As this analyzer, by producing catalytic combustion of combustible components of the gas stream on the surface of the platinum wire whose temperature affects the balance of a resistance bridge, indicates the presence of all products of incomplete combustion, these tests were considered conclusive proof that the fuel was completely burned out at the discharge of the chamber.

TABLE 2 GAS ANALYSES FROM TESTS ON EXPERIMENTAL COMBUSTION CHAMBER

Run Number	Combustible Gas Content of Outlet Gas by MSA Analyser	Gas Analysis by Orsat Gas Analyzer: Inlet Gas CO_2	O_2	CO	Outlet Gas CO_2	O_2	CO
29	-	0.8	19.2	0.0	2.8	16.5	0.0
30	-	0.8	19.2	0.2	3.4	15.8	0.0
31	-	0.0	20.9	0.0	2.0	17.8	0.0
32	-	0.1	20.2	0.0	2.2	17.6	0.0
33	-	0.8	19.3	0.0	2.4	17.2	0.0
34	-	0.8	19.4	0.0	2.6	16.8	0.0
35	-	0.8	19.4	0.0	3.2	15.8	0.0
36A	-	0.0	20.9	0.0	0.5	20.4	0.0
36B	-	0.0	20.9	0.0	0.5	20.5	0.0
37	-	2.3	16.3	0.0	3.2	15.4	0.0
38	-	0.8	19.3	0.0	2.3	17.1	0.0
39	-	0.7	19.5	0.1	2.3	17.2	0.0
40	0.02	-	-	-	-	-	-
41	0.	-	-	-	-	-	-
42	0.	-	-	-	-	-	-
43	0.	-	-	-	-	-	-
44	0.01	-	-	-	-	-	-
45	-	0.5	19.8	0.0	2.4	17.2	0.0
46	-	0.5	19.5	0.0	2.4	17.0	0.0
47	-	-	-	-	4.1	14.9	0.0

Test Results. The detailed results of the combustion-chamber tests are presented in Tables 1 and 2, while the efficiency measurements are summarized in Fig. 6. Above fuel-supply rates of 2.5 $\times$ 10^6 Btu per hr, the combustion-chamber efficiency remains above 99 per cent, up to the maximum load carried in these tests, 6.5 $\times$ 10^6 Btu per hr. The limitation on the maximum load was imposed, not by oil-supply capacity or combustion conditions, but by the necessity of limiting the temperature of the discharge, which was vented to the atmosphere through mild-steel ducts.

This maximum represents about 35 per cent of the full-load heat input of the high-pressure chamber in the gas-turbine set. Tests on the chambers installed in the gas-turbine set, in which rates up to the design full-load condition have been carried, indicate no decrease in combustion efficiency as the load is further increased. Of course, these tests are not of as high precision as those on the experimental combustion chamber because of the impracticability of accurate measurement of air flow, and the lack of checks on temperature measurements. A significant result is that neither the stability of operation, nor the combustion efficiency is affected by the presence of products of combustion in

the inlet air, or by the reduction of the oxygen content of the inlet air to 16.3 per cent on the dry basis, corresponding to about 15.6 per cent before condensation of moisture.

As the combustion rate is decreased the efficiency of the chamber appears to fall off to about 81 per cent at a heat-liberation rate of 0.7×10^4 Btu per hr, the lowest stable combustion rate that could be maintained. At the lowest rate the radiation from the outside shell of the chamber accounts for approximately 5 per cent of the heat supplied by fuel, while incomplete combustion accounts for only 1 per cent or so more. The remainder of the apparent loss, amounting to about 14 per cent of the heat in the fuel at the lowest rate, is unaccounted for and must be ascribed to inaccuracies in the measurements.

Preliminary tests had shown that the pressure loss of the air passing through the chamber amounted to approximately one velocity head of the fluid at the entrance to the chamber. These tests were made in a chamber lacking the transition and elbow at the discharge of both the experimental chamber and those in the gas-turbine plant and were not of a high order of precision. The tests reported in Table 1 confirmed this result, however; they also permitted the formulation of a relatively precise expression for the loss in pressure. Study of the data by the methods of correlation analysis (13) showed that the pressure drop was dependent upon the velocity head of the gas leaving the combustion chamber, as well as that at the entrance. Since the quantity of fluid is essentially the same at entrance and exit, these velocity heads are simply related through the ratio of the absolute temperatures at the two sections, designated here as β. Using this convention, the results of these tests may be expressed

$$p = (0.360\beta - 0.077)h_1 \quad \text{[3]}$$

where

p is static pressure drop in any convenient units

h_1 is velocity head at combustion-chamber inlet in same units

$\beta = \dfrac{T_2}{T_1}$ is ratio of absolute temperature of air leaving to that entering the chamber

This expression represents the data of Table 1 to within about 9 per cent and was used to predict the performance of the chambers for the gas-turbine plant.

Stratification of the gas stream with respect to temperature has been the most troublesome characteristic of the combustion chamber. The difference in temperature across the gas stream may amount to 50 per cent or more of the average temperature rise. Temperature traverses have shown that the hot spot is always displaced from the center line toward the inlet of the chamber. It is curious that the chamber may operate consistently with low stratification at one speed of the turbine and just as consistently with high stratification just a few hundred revolutions higher or lower. Further investigation of this characteristic is being made.

The highest metal temperatures occurring in the chamber are those of the ignition cone. Temperatures of the inner shell were also measured during the early runs on the experimental chamber, but the couples at these positions were broken, before the tests reported here were run, by the axial movement of the inner shell relative to the outer shell accompanying heating up and cooling down. They remained in place long enough, however, to show that the shell at no point exceeded the temperature of the ignition cone. Immediately in front of the cone, at the inlet side, the shell temperature usually approached the maximum cone temperature. The variation of the average ignition-cone temperature with the load on the chamber is quite regular, as shown in

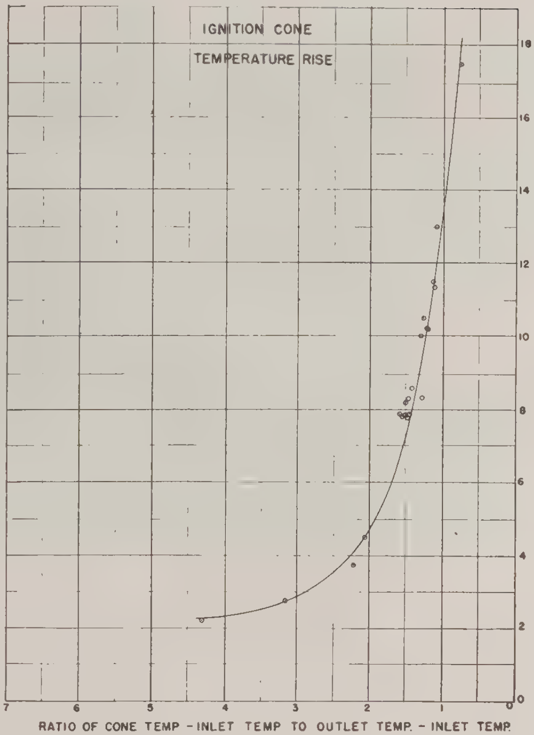

Fig. 7 Ignition-Cone Temperature Rise

Fig. 7. The results shown here may be generalized by the conclusion that at no point in the chamber does the metal temperature exceed the chamber discharge temperature by more than 300 deg F, under any condition of operation.

Notes on the Development

Work on this project started late in 1942, with an attempt to design a combustion chamber that would fit the space requirements of the proposed plant layout. Several designs following more or less conventional practice were completed and chambers of mild steel were built for testing in an outdoor test stand.

In these conventional chambers combustion air was separated from tempering air by a metal shell (2) since it was thought that the large amount of excess air would quench the burning of the fuel. The largest part of the air passed around this shell while combustion air was admitted to the burner end of the shell through a variety of registers, most of which were equipped with adjustable vanes, as in boiler practice. A "spinless" register was also built, in which the primary air was admitted between two coaxial cones, which produced much more stable ignition in a smaller volume of the chamber than did any of the others. It was also found that such stable ignition was essential, with either continuous or intermittent injection of the fuel, to reduction of the flame length. This register was the elementary form of the ignition cone now in use.

Of the chambers proper, those having circumferential slots which admitted air axially along the walls of the inner shell were found to produce very long flames, while those having round holes producing radial jets gave short flames but required relatively larger pressure drop. The "elbow" form of chamber was produced fortuitously by the removal of all inner shells in an attempt to reduce pressure drop.

Acknowledgments

The authors wish to acknowledge the assistance and encouragement supplied throughout this development by Commodore R. V. Kleinschmidt and his staff in the Research Section of the Bureau of Ships; and to thank the Chief of the Bureau of Ships for permission to publish this material. We wish also to thank Mr. R. B. Smith, vice-president Engineering, and the rest of the management of the Elliott Company for their support of this work; and to express our gratitude to William Schoenfelt, formerly a member of the Engineering Research and Development Department of the company, and especially to George Schertzinger, for their devoted work on this problem.

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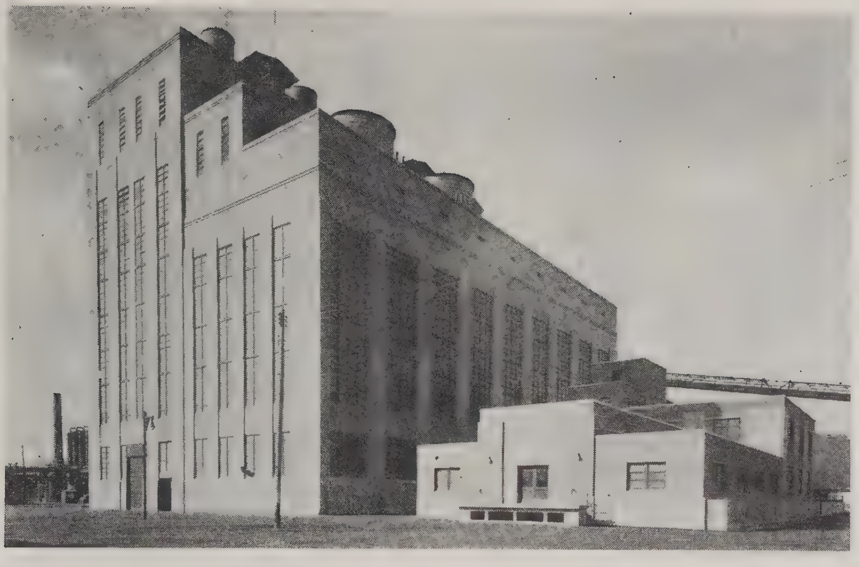

FIG. 1 GENERAL VIEW OF POWER PLANT

A Comparison of Operation of Forced- and Natural-Circulation Boilers

BY G. F. ROSS[1] AND LEONARD WILKINS,[2] KOBUTA, PA.

Steam-generating facilities at the Koppers Company, Incorporated, Butadiene Division, Butadiene-Styrene Chemical Plant, at Kobuta, Pa., include two types of units; natural-circulation boilers and the first forced-circulation boiler to be installed in the United States for industrial-plant processes. Since plant operation began in the summer of 1943, this installation has afforded boiler operators an opportunity to compare the operating characteristics of the two types. This paper describes briefly the design and operating features of these steam generators, together with some of the difficulties encountered and how they were overcome.

THE installation of a controlled forced-circulation boiler unit adjacent to three natural-circulation boiler units of identical rating at the Koppers Company, Incorporated, Butadiene Division, Butadiene-Styrene Chemical Plant[3] has afforded boiler operators an opportunity to compare the operating characteristics of these two types of steam-generating units. Boiler designers and operators are naturally interested in the first application in this country of a forced-circulation boiler unit to supply steam for industrial-plant processes. The steam-generating facilities at Kobuta were built at a time when the steel demand for other urgent war needs was at its peak; the Government directed that a forced-circulation boiler unit should be installed. Plant operations began in June, 1943, with one natural-circulation boiler; in August, 1943, the forced-circulation unit was placed in service.

It will be the endeavor of this paper to describe briefly the design features and compare the major operating differences between the two types of steam-generating units and to outline a few of the operational difficulties that accompany boiler generators of this type. Problems attendant to starting up the units for the first time are intentionally omitted, as boiler designers and operators are well aware that initial operating difficulties rarely recur after the difficulty is once remedied. All boilers installed at this plant met the manufacturers' expected-performance specifications and this paper will deal only with problems arising after acceptance.

It is well to point out that operating personnel was obtained several months prior to operations, in order that start-up of the boiler units could proceed without delay. Operators observed the complete erection of each boiler unit with its auxiliaries, and supervisors conducted short classes with operators to familiarize them with the operating characteristics of each boiler. The wisdom of such a procedure was illustrated by the ability of the operators to start up each unit as it was completed and maintain it on the line as a reliable steam source for plant process. The start-up of the forced-circulation unit was no more of an event than that of the natural-circulation units, as operators had become accustomed to its unconventional design.

[1] Assistant Superintendent of Utilities, Koppers Company, Inc., Butadiene Division.

[2] Maintenance Engineer, Koppers Company, Inc., Butadiene Division.

[3] This plant is operated by Koppers Co., Inc., Butadiene Division, an agent for the Reconstruction Finance Corporation, Office of Rubber Reserve, Washington, D. C.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of authors and not of the Society.

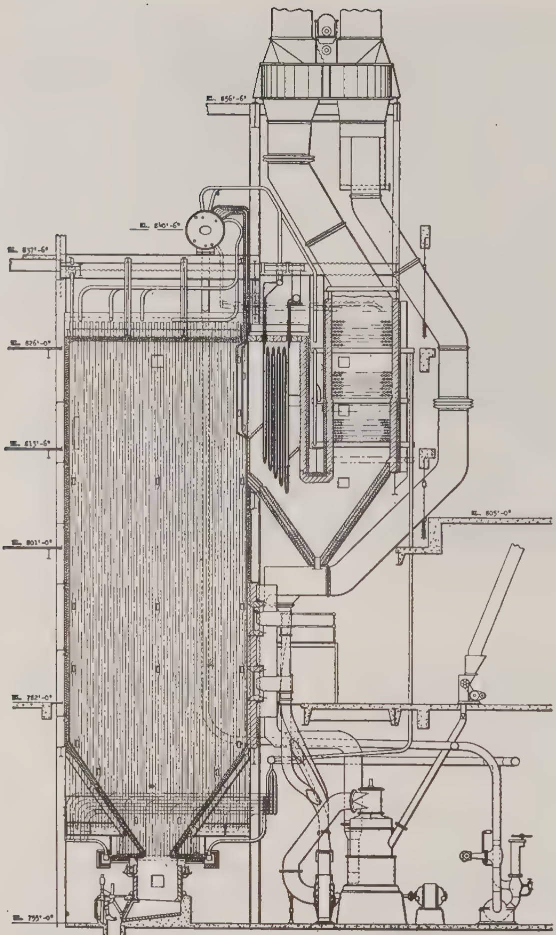


FIG. 2 CROSS-SECTIONAL VIEW, FORCED-CIRCULATION BOILER

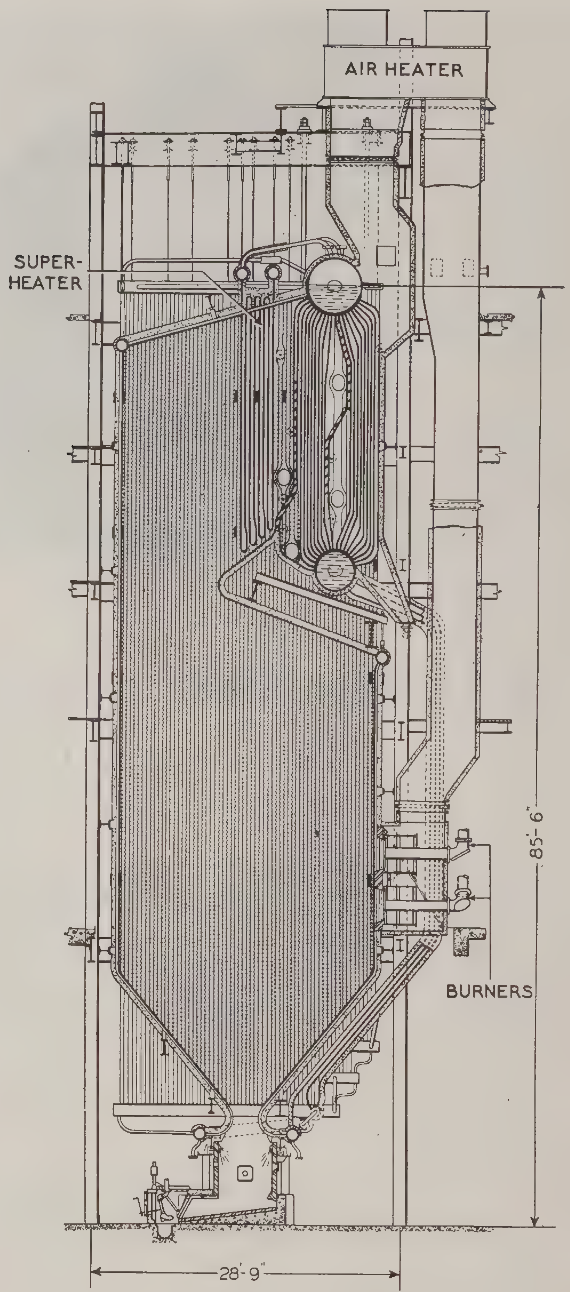


FIG. 3 CROSS-SECTIONAL VIEW, NATURAL-CIRCULATION BOILER

It is beyond the scope of this paper to present in detail the design data of these boiler units. Aside from a secondary interest in these data, boiler operators are primarily interested in the operating characteristics, availability, water control, and maintenance of their steam-generating units. A description of the physical layout of the steam plant is included to better acquaint the reader with its size, design features, and operating conditions. It must be remembered that war regulations prohibited extensive refinements in this steam plant, and these limitations were met remarkably well.

The time limit of construction and operating date prohibited exhaustive studies in layout and operating equipment. That the plant has produced steam consistent with high demand attests to the ability of the designers and operators to meet the wartime limitations. Most boiler operators will agree that boiler auxiliaries present more operating difficulties than the actual furnace, and it is well to note that in this case the boiler auxiliaries are not as a whole duplicated in each of the types of boilers. No attempt will be made to compare these auxiliaries, but the inclusion of the operation of the forced-circulation pumps is made, since it is considered an integral part of the forced-circulation boiler unit.

It is not the intention here to present data to influence the selection of any particular type of boiler unit. Boiler operators pride themselves on being able to fire any type of steam-generating equipment and to secure from it the maximum possible efficiency and reliability. After two years of experience with each of these types of boilers, it is possible to make a comparison of their operation and to present to those interested a summary of our experience at Kobuta.

DESCRIPTION OF PLANT AND BOILER AUXILIARIES

The original steam plant consists of four 350,000-lb per hr

boilers, designed for superheater-outlet conditions of 725 psi pressure and 750 F temperature. The first is a Combustion Engineering Company single-drum controlled forced-circulation boiler, Fig. 2, and the other three are Babcock & Wilcox Company two-drum, radiant-type, natural-circulation units, Fig. 3. Operating experience indicated that the capacity of three boilers was not sufficient to supply all the steam demanded by the chemical-process units with one boiler off the line for inspection and maintenance. Accordingly, a fifth boiler of 180,000 lb per hr capacity at 450 psi pressure without superheat was installed. This latest addition is a Combustion Engineering Company four-drum natural-circulation unit. Total steam-generating capacity available from the five boilers is 1,580,000 lb per hr. The four high-pressure units are housed in a reinforced-concrete boilerhouse, approximately 130 ft high, while the low-pressure unit is installed in a separate annex at the south end of the original concrete boiler room. No further reference will be made to the smaller natural-circulation unit and its auxiliaries.

Draft equipment for each of the high-pressure units is identical, consisting of one induced- and one forced-draft fan per boiler, with turbine drives. An important factor in selecting boiler capacity was the limitation of draft equipment to single fans per boiler so as to require a minimum number of reduction-geared-turbine drives, an extremely critical item during the construction period. Fly-ash collection equipment is within the induced-fan housings, thus minimizing the use of steel. Each boiler is equipped with a Ljungström air preheater. A combination of two coal-and-gas, and two coal-and-oil burners fire the forced-circulation boiler. Natural-circulation-boiler furnaces are equipped with six combination coal-and-oil, and two separate gas burners. All furnaces are dry-bottom, with intermittent ash-sluicing facilities, and continuous ash sprays. The forced-circulation unit is equipped with two Raymond bowl mills, with integral exhausters, and the natural-circulation units are fired with two Hardinge-type ball mills, with remote exhausters located on the operating floor. Suitable feed pumps are installed in the pump bay on the ground-floor elevation. These consist of five units, i.e., three turbine-driven, one motor-driven, and one combination turbine-and-motor dual-drive unit.

Steam at 700 psi and 750 F goes to a 35,000-kw turbogenerator which exhausts at 165 psi back pressure for chemical-plant-process requirements. Total plant electrical load approximates 14,000 kw, the excess electrical energy going into the local utility system. Exhaust steam leaves the turbine discharge and passes into two 24-in. lines at 165 psi and 460 F, which maintains slight superheat to the farthest end of the process-plant distribution lines. Part of this steam is reduced to 75 psi for process, as needed. In order to furnish 165-psi steam with a turbine or utility tie-line outage, desuperheating and pressure-reducing stations are installed between the boiler headers and exhaust lines. Any sudden interruption to exhaust-steam supply operates these stations through master controllers. Desuperheating water is supplied from the condensate system through special pumps installed at the feed-pump bay.

Main high-pressure boiler-plant piping is so arranged that boilers may be isolated into two separate generating systems, each

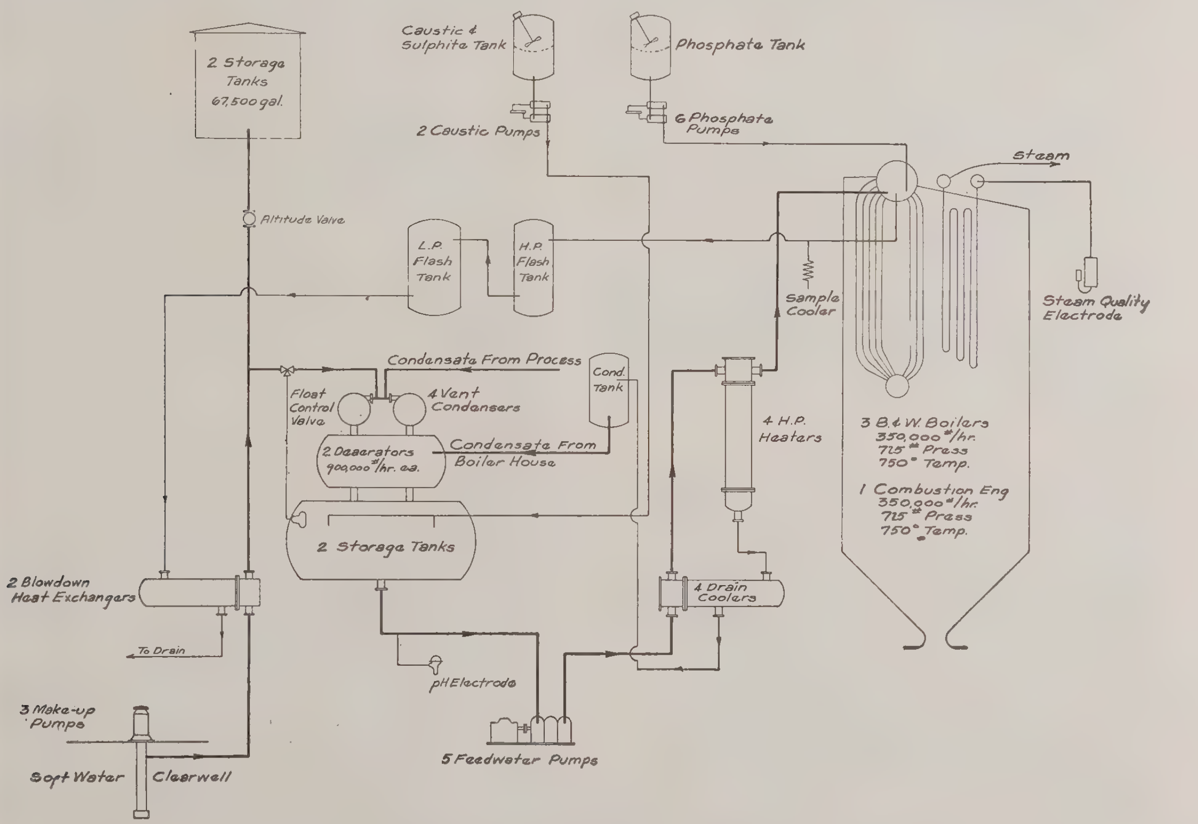


FIG. 4 BOILER-FEEDWATER FLOW DIAGRAM

containing its own feedwater, steam-generating, and blowdown facilities. The forced-circulation main steam lead and one of the natural-circulation units are tied to a common header with the remaining two high-pressure units discharging into a similar header. The turbogenerator is fitted with double inlet valves, one from each main boiler header.

The blowdown system consists of two separate series of high- and low-pressure flash tanks, from which 165-psi and 10-psi steam is supplied to the high-pressure heaters and to the deaerators, the effluent being sewered through the blowdown heat exchangers. It has been noted that during peak loads and high rate of blowdown, approximately 50 per cent of the 165-psi steam requirements for the high-pressure heaters is met by the high-pressure flash steam. The feedwater flow is illustrated in Fig. 4.

Coal comes to the plant by rail, truck, or barge. Belt conveyers transport it from unloading points to storage or directly into the crushers where it is sized to $^3/_4$ in. or less, before entering the bunker house. A feeder tripper distributes coal into eight 250-ton hoppers from which it drops by gravity, through automatic scales to mill feeders and pulverizers at each boiler. Furnace volumes permit carrying full boiler load on coal alone, consuming 1500 to 1700 tons per 24 hr for the four large boilers. All-electric combustion control is duplicated on all boiler units, regulating air and coal feed according to steam demands. Hand preadjustment of the control mechanism assures sufficient combustion air for oil and gas.

Feedwater-pump- and draft-fan-turbine exhaust goes to two deaerating feedwater heaters that handle both return condensate and raw make-up. A reducing valve from the main-turbine exhaust supplies any deficiency. The high-pressure closed heaters receive steam directly from the main-turbine exhaust, supplemented by flash from the blowdown system.

Water-Treating Facilities

Treating-plant equipment and basins are located adjacent to the powerhouse in a separate building and connected to the station by a pipe tunnel. Ohio River water is fed into three Spaulding precipitators in the water-treating plant through a 20-in. line. The raw-water flowmeter controls the rate of chemical input. Lime, soda-ash, and copperas solutions are fed by individual pumps into the inlet flume of the precipitators. At the maximum rate of flow through the precipitators, the detention time is 90 min. The water, having been clarified and partially softened in its passage through the precipitators, flows by gravity to six Anthrafilt filters. The filtering capacity is 3900 gpm and

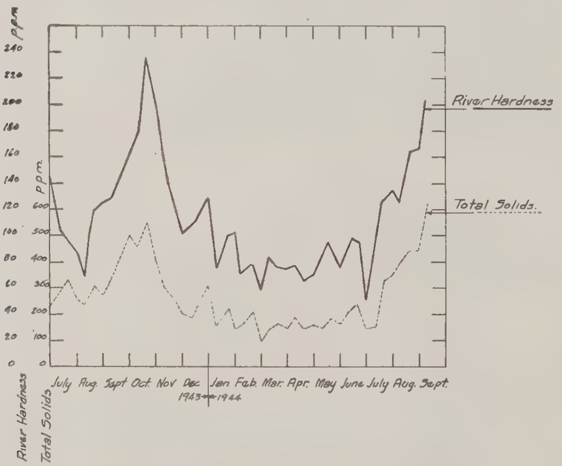


Fig. 5 Ohio River Water Characteristics

the flow of water through the filters is regulated by rate-of-flow controllers. In leaving the filters, the water is discharged into an acid-mixing chamber before passing into a filter clearwell, and a pH value of 8.3 is maintained to prevent deterioration of the zeolite in the final softeners.

After the water is clarified, partially softened, filtered, and acidified to the correct pH, it is pumped to six gravity-flow zeolite softeners. The design of the equipment was based on a hardness of 55 ppm from the filter clearwell. With this capacity setting, each unit is capable of softening about 770,000 gal of water in each regeneration. The zeolite operates in the sodium cycle with rock salt for preparation of the solution. The treating process is capable of delivering the required boiler make-up water to the boilerhouse with a residual hardness rarely exceeding 2 ppm even during periods of peak hardness of the raw water.

Make-up water is pumped from the soft clearwell to two deaerators located in the boiler plant. Each deaerator is designed to operate at a pressure of 10 psi with a guarantee to remove the oxygen from 900,000 lb of feedwater per hr. To protect the equipment from corrosion attack, sodium sulphite is fed into the deaerator storage tank. Caustic soda is also added at this point to maintain a safe pH value in the feedwater, as it passes through the feed pumps and high-pressure heaters. Caustic-soda feed is also used to maintain the specified alkalinity range in the boiler water. Sodium-metaphosphate solution is fed directly into the upper drums of the four boilers to satisfy demands of residual hardness of the boiler feedwater and acts as a sludge-forming accelerator. Each boiler is equipped with a boiler-water-sample cooling coil located at the upper drum, and two Straub condensers with conductivity cells and recorders are provided to obtain a continuous check on the quality of steam leaving the boiler drum.

Controlled Forced-Circulation Unit

The sectional view of the forced-circulation unit shows no lower or mud drum. Forced circulation eliminates the need for the lower drum with its interconnecting generating surface necessary to set up circulation between the lower and upper drums. Boiler feedwater enters the single drum after passing through the economizer which is located in the upper section of the convection zone, immediately below the air preheater. The feedwater is introduced into the drum through a submerged pipe which extends the entire length of the drum, discharging the water through perforations. Two 10-in. downcomer pipes, located about 6 ft from each end of the drum, carry the water down to the circulating pumps. These downcomer openings are protected by wire screens as mechanical protection for the circulation-pump impellers and are also fitted with downtake spiders to eliminate vortex formation.

The circulating pumps are 10 × 10 Allis-Chalmers horizontal single-stage type, Fig. 6, rated at 4500 gpm, 151 ft head, 517 F, and suction pressure of 800 psi. They have an end suction and a side discharge and are driven by horizontal steam turbines through a Fast "spacer" coupling. Ball-thrust bearings absorb all the thrust load of the pump. Through close co-operation between the representatives of the boiler and pump manufacturers, the pumps are so designed to maintain a constant and unfailing flow of water through the boiler and to operate continuously and without interruption, in order to protect the boiler tubes. To facilitate removal of the rotating element, there has been incorporated in the design of the pump a casing cover, or end head, through which all the internal parts of the pump can be removed. A close-clearance bushing directly behind the pump impeller is used with "bleed-in" and "bleed-off" connections to reduce the temperature and pressure on the stuffing box. The bleed-in connection permits a water flow of 20 gpm at 240 F and 900 psi from the boiler feed pump discharge main to enter the inner end

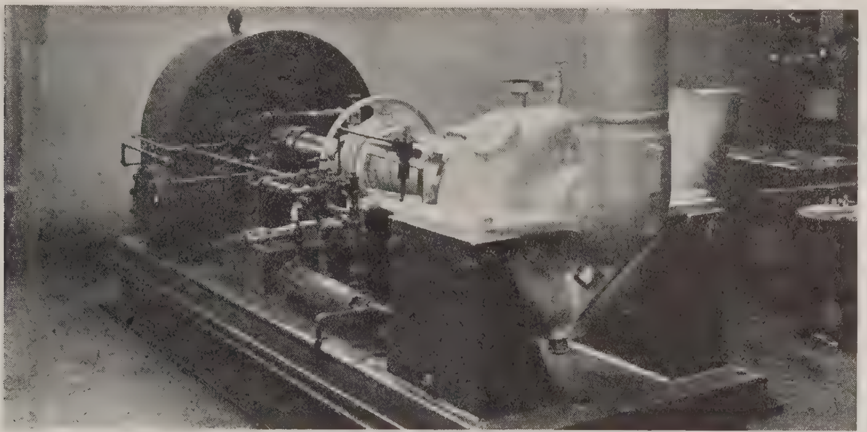


FIG. 6 FORCED-CIRCULATION-BOILER PUMP
(Rated at 4500 gpm, 151-ft head, 1500 rpm, 517 deg F; suction pressure 800 psi.)

of the reducing bushing. This flow of water at a pressure higher than pressure in the pump maintains a water flow at 240 F out of the bushing into the pump, instead of the hot boiler water in the pump flowing back toward the packing. About 2 gpm of 240 F water flows into the pump, and the remaining 18 gpm is bled off through the bleed-off connection. By this arrangement it is possible to pack against a low pressure and low temperature, rather than against the normal boiler pressure and temperature. Further reference is made to this feature later in the text.

The circulating pumps located on the ground floor withdraw water from the drum and discharge into a loop which supplies water to wall-distributing headers. Two of these headers supply the burner and rear-wall tubes, two supply the central section of the side walls, four others supply the outside sections of the side walls, and the remaining one supplies water to the secondary generating section.

Flow of water from the headers to the wall and secondary-generator tubes is controlled by a series of combination orifice strainers, consisting of a nipple approximately 6 in. long perforated by $^3/_{16}$-in. holes and drilled with a control orifice. The arrangement is such as to permit removal of the orifice strainers from the headers for inspection and cleaning. One orifice strainer serves a tube nozzle which is trifurcated for the three wall tubes which arise from it. A total of 276 strainer orifices are provided to supply 183 tubes on each of the burner and rear walls, 198 tubes on each of the side walls, and 32 tubes in each section of the secondary generator located immediately below the economizer. Front- and rear-wall tubes lead directly into the top of the drum above water level, while the side-wall and secondary-generator elements collect into headers and pass into the drum through 4-in. risers entering the drum above the water level.

The 54-in-drum internals are so arranged that the water-steam mixture is first directed downward by a deflector baffle at the tube entrances. The flow is then reversed in direction permitting the water to separate from the steam. This reversal permits the steam part of the mixture to continue directly to the screen driers, while the remainder of the mixture bubbles through a submerged screen below the water level. Steam at relatively low velocity passes through three sets of screen driers for the complete removal of any entrapped moisture. In leaving the drum, the steam passes around deflector plates located directly under the tubes leading to the superheater.

The superheater element hangs directly in the gas stream at the entrance to the convection section and is protected by a slag screen of staggered front-wall tubes, three rows deep. Combustion gases pass out of the furnace through the front-wall screen, across the superheater bank, through the convection bank containing two sections of generating tubes, and through the economizer. The last banks are supported on water-cooled beams which are part of the secondary-generator circuit. Soot is removed below the superheater, and by the dust collector in the induced-draft-fan housing.

It will be noted that in the forced-circulation unit there is no need for generating tubes to be suspended in the gas passes and baffles for routing gas flow are not required. The forced-circulation unit thus approaches a favorable design of gas passes and tube arrangement to overcome any fly-ash erosion difficulties.

Boiler dimensions of the forced-circulation unit are given in Table 1.

TABLE 1 FORCED-CIRCULATION-UNIT DATA

Furnace volume, cu ft gross	25,400
Boiler heating surface, sq ft	4,510
Waterwall surface, sq ft	8,470
Superheater surface, sq ft	4,430
Economizer surface, sq ft	9,830
Air preheater surface, sq ft	38,900
Operating water volume, lb	36,643
Steam drum	54″ diam × 28′–9½″ long
Upper wall tubes	$1^1/_4$″ OD × 0.120″ wall thickness
Lower wall tubes	$1^1/_4$″ OD × 0.150″ wall thickness

NATURAL-CIRCULATION UNITS

As noted by the sectional side view, Fig. 3, the natural-circulation boilers have two drums with pendant superheater. Feedwater enters the upper drum through a perforated pipe below the water level and passes down to the mud drum through interconnecting generating and heating tubes. Water is then distributed to the waterwalls through eight headers located at the bottom of the furnace. Two of these supply front and rear walls, two supply the center and rear section of the side walls, and the other four supply the front sections of the side walls. Rear- and front-wall tubes terminate directly in the steam drum entering below water level. The side-wall tubes terminate in headers at the top of the furnace and continue to the drum by risers entering the drum above water level, but inside the internal baffle compartment.

The 60-in-drum internals cause the water and steam mixture in the wall and generator tubes to enter a baffled compartment from which the mixture passes tangentially into 44 cyclone steam separators. As a result of the whirling action of the cyclones, the

water is directed downward while the steam passes upward through the corrugated scrubber elements for further removal of moisture. In leaving the drum the steam passes through a twin-type steam scrubber where the last trace of moisture is removed. The natural-circulation boilers are not equipped with economizers. The convection bank and superheater elements are protected from direct radiation by a water-cooled arch extending approximately halfway into the furnace, immediately below the mud drum.

Attention is called to the baffle arrangement as shown in the sectional view, Fig. 3. The baffle between the first and second pass is of tile construction. The upper section of the baffle separating the second and last pass was originally as shown in Fig. 3 and is also of tile construction, the sloping portion being of the poured type, and the lower section again of tile construction. In recent months two generating-tube failures have occurred on two of these natural-circulation units due to fly-ash erosion of the tube metal. In the first case, leakage of dust-laden gas through the baffle, separating the second and last pass, seriously eroded the generating tubes directly behind this baffle, a short distance below the top drum. In the other case, concentration of fly ash sweeping down the slope of this same baffle eroded tubes in the row just ahead of this baffle at a point just below the baffle slope. Baffle changes have been completed on all three of the natural-circulation units by making the entire baffle vertical from the drum downward. This arrangement should overcome both difficulties as well as afford convenient access to the furnace for inspection of the baffle for leakage. Soot is removed at the second pass of the convection bank at mud-drum level, as well as by collection in the induced-draft fan similar to the forced-circulation unit.

Boiler dimensions of the natural-circulation units are given in Table 2.

TABLE 2 NATURAL-CIRCULATION-UNIT DATA

Boiler heating surface, sq ft	20,640
Waterwall surface, sq ft	6,367
Superheater surface, sq ft	4,030
Total steam-generating surface, sq ft	31,007
Air-heater surface, sq ft	38,600
Volume of combustion space, cu ft	29,900
Operating water volume, lb	150,680
Front-wall tubes	3¼″ OD × 0.203″ wall thickness
Steaming drum	60″ diam. × 30′ long
Mud drum	42″ diam. × 29′ long
Rear-wall tubes	2½″ OD × 0.170″ wall thickness
Side-wall tubes	2½″ OD × 0.170″ wall thickness
Roof tubes	4 ″ OD × 0.213″ wall thickness
Generating tubes	2½″ OD × 0.148″ wall thickness

Operating Conditions

During the latter part of 1943, the entire year of 1944, and the first quarter of 1945, the directed rate of chemical production demanded a peak steam load of 1,300,000 lb per hr, with an installed capacity of 1,400,000 lb per hr at that time. It was the responsibility of the boiler operators to maintain maximum steam generation under extreme varying conditions. Boiler outages were purposely kept to an absolute minimum by maintenance forces on a 24-hr schedule to keep the boiler units in operating service. Initial planning called for scheduled outages of boiler units each 6 months, but revisions were made on first inspections and subsequent tube failures to remove the forced-circulation boiler from the line each 60 days for inspection, and the natural-circulation units each 180 days. This schedule was adhered to during the entire period of peak steam demands.

The complexity of the butadiene-styrene chemical process, with its large number of stills, compressors, and heat exchangers, imposed on the boilers the requirement of maintaining correct pressure and temperature at all times. The chemical process is extremely sensitive to variations in steam conditions and slight steam upsets curtail chemical production. During boiler difficulties, when generating capacity could not maintain proper steam conditions, operators were reluctant to drop steam delivery to process because subsequent readjustment to normal production rate would require 24 to 48 hr, in addition to entailing large losses of raw materials. The boilers were called upon to recover from adverse conditions promptly and in most cases, boiler operators were able to maintain a continuity of steam flow to the process units.

In contrast to the operation and performance of a boiler under central-power-station conditions where make-up water rarely exceeds 2 per cent, and where even this amount is supplied by evaporation, boiler make-up here at Kobuta has been of the order of 60 to 70 per cent due to process-steam requirements. These steam requirements necessitate discarding two thirds of the steam condensate in process wastes.

Ohio River water, which is the raw-water source, contains principally sulphate salts and is generally slightly acid in nature, with a pH of 6.5. Total hardness for the major portion of the year ranges between 70 and 90 ppm expressed as calcium carbonate.

In Table 3 the analysis given for Ohio River water may be considered as being the average.

TABLE 3 TYPICAL ANALYSIS OF OHIO RIVER WATER

	Ppm
Bicarbonate (HCO_3)	7
Sulphate (SO_4)	96
Chloride (Cl)	15
Silica (SiO_2)	5
Iron (Fe)	4
Calcium (Ca)	23
Magnesium (Mg)	5
Sodium (calc.) (Na)	23
Suspended solids	17
Total dissolved solids	195
Soap hardness as $CaCO_3$	78

Composition of the water is subject to wide variations in hardness and dissolved solids, as shown in Fig. 4. During the 2 years that the plant has been in operation, control of water conditioning and continuity of operation have been maintained under conditions where hardness of the water has fluctuated from a low of 60 ppm during the winter and early spring seasons, to a high exceeding 250 ppm during periods of low river flow. Similarly, dissolved solids fluctuated from a low of 100 ppm to a high exceeding 600 ppm. Obviously with boiler make-up exceeding 60 per cent, and with solids concentrating in the boilers in a very short time during periods of peak dissolved solids, conditions exist which tax a conventional natural-circulation boiler to the limit and impose on the operation of a forced-circulation unit the additional measures necessary to prevent an accumulation of suspended matter at its control orifice strainers great enough to cause tube failures. Blowdown equipment is adjusted to maintain a dissolved-solids content of 2000 ppm in the boiler water.

On the basis of the average analysis of raw water in which dissolved solids were 178 ppm a blowdown of 10 per cent was expected. As most of the heat-recovery equipment was designed to handle this quantity, the blowdown facilities were vastly overloaded when it became necessary to blow down 30 per cent of the feedwater in order to maintain the desirable boiler-water conditions when the dissolved solids in the raw water exceeded 600 ppm. After installing larger blowdown valves and pipes, and with the continuous-blowdown equipment operating at maximum, it is still necessary to supplement this with continuous blowing of the mud drums. With the installation of the fifth boiler, blowdown-equipment arrangements can be made to modify the blowdown lines in such a manner as to recover the heat normally lost in the use of the mud-drum blows.

Residue oil and gas furnished from the process units is delivered to the boilerhouse for burning in combination with the

coal obtained from the local bituminous mines. Normally, moisture content of the coal does not exceed 4 per cent, but there have been periods during transportation difficulties, when 10 to 15 per cent moisture coal has been delivered to the pulverized-fuel equipment; such periods occurring when steaming capacity was at a premium. While the ability to satisfy steam demand under these moisture conditions is a function of the fuel-burning and handling auxiliaries, boiler operators were called upon to adjust boiler ratings continuously to maintain proper pressure and loading. These periods made it possible to observe the individual boiler response to the demands for steam to process. That no particular difference between the boiler units was found might be surprising to those who believe that the forced-circulation unit responds and recovers load in less time than the natural-circulation units of the same capacity. With the units equipped with different types of mills, any indication of such conditions might be attributed to the mill performance rather than to that of the boiler.

Local strip-mined coal, ranging in quantities of 25 to 50 per cent of the total fuel, has been fired in all boilers. Two generating-tube failures and severe erosion of generating tubes in the natural-circulation boilers have led several consultants to consider these conditions to be the result of the high silica content of the ash of this strip-mined coal.

Operating Characteristics

The forced-circulation boiler, with $1^1/_4$-in-OD wall tubes and only one drum, has considerably less water in it than the natural-circulation unit has, with two or more drums and larger-diameter wall and generating tubes. The forced-circulation boiler holds approximately 37,000 lb of water at steaming level, compared to the natural-circulation boiler which holds 155,000 lb, or nearly $4^1/_4$ times as much. This causes a greater sensitivity in the forced-circulation boiler to changes in feedwater conditions or changes in controls.

With the natural-circulation boiler operated at 350,000 lb of steam per hr for a period of 2 hr with the blowdown completely shut off and with 400 ppm dissolved solids in the feedwater, an increase of 1800 ppm in the solids concentration of the boiler water would result. This added to the normal 2000 ppm carried would mean a total concentration of the boiler water of 3800 ppm. Actual operating data indicate that these boilers can be operated at 400,000 lb per hr with 4000 ppm dissolved solids. Should the continuous blowdown be completely shut off on these natural-circulation boilers, no serious results would be experienced before the boiler operators could remedy the condition.

In the case of the forced-circulation unit under the same conditions, the concentrating effect of evaporation would result in a total dissolved solids content of 9600 ppm. At a steam rate of 350,000 lb per hr, solids carry-over would probably begin with a concentration of 5000 ppm and would doubtless be experienced if the blowdown were stopped for only $^1/_2$ hr.

To be reasonably free from upsets of the nature described, analyses of the boiler waters are made every 2 hr, total solids being checked by the conductivity method. In so far as the natural-circulation boiler is concerned, this 2-hr frequency is adequate to detect any major change in conditions before the results become serious. Such is not the case in the forced-circulation unit. Carry-over has been experienced on several occasions with the forced-circulation unit as a result of rapid increase in solids concentration, the carry-over being detected by means of the steam-temperature recorder.

Arrangements are being made to install a conductivity recorder on all the boilers, recording total solids in the continuous blowdown. Especially in the case of the forced-circulation boiler, where the quantity of water at steaming level is relatively small and where the percentage make-up is relatively high, a conductivity apparatus recording total solids should be installed as standard equipment with the boiler.

Natural-circulation units depend on differences in specific gravity to set up circulation; and as a result, circulation rates vary with changes in rating or changes in heat input. In the forced-circulation boiler the rate of circulation is fixed and controlled by means of centrifugal pumps which take the water from the main drum and discharge it to distribution headers where it enters the tubes through control orifices. These orifices are so sized that a reasonable pressure drop is experienced across them and good control of distribution is the result.

It is natural to expect that a centrifugal pump installed in the boiler-water circuit would be regarded as an outstanding weak point in the entire system. In actual practice, this has not proved to be the case at Kobuta. To date there has been no instance of circulation failure due to the pumps. The forced-circulation unit is equipped with two pumps, one operating at rated speed and load, the other idling at approximately 400 rpm. The original installation required manual operation to place the idling pump in service should the loaded pump fail. This arrangement made it mandatory to station an operator at the pumps at all times. Physical layout has placed the pumps remote from the boiler feed pump bay, and in the future, boiler-installation designers might consider locating a forced-circulation boiler in such a manner that the circulation pumps can be attended by the normal boiler feed pump operator or water attendant. The usual interlocks for tripping the coal-pulverizing mills when the pump differential reaches the low limit were provided, but this feature was removed shortly after the unit went into service, because of the feeling that the pump attendant was sufficient protection.

The stationing of the pump attendant was regarded as a stopgap measure for emergency protection of the boiler, and at present quick-opening steam valves have been installed to bring the idling pump to full speed and rating automatically. The control for these valves is taken directly from a differential-pressure controller. On occasions when upsets resulted in low steam pressure to the pumps, and pump differential reached the low safe limit, the idling pump was automatically put in service in a very satisfactory manner. It is now believed that with this feature added, and based on the performance of the pumps to date, the reliability of the circulation system is assured.

Efficiency and performance tests have indicated no outstand-

TABLE 4 PERFORMANCE DATA OF FORCED-CIRCULATION AND NATURAL-CIRCULATION STEAM GENERATORS

	Forced-circulation unit	Natural-circulation unit
Steam output, lb per hr	352500	345000
Temperature of air for combustion, deg F	90	106.3
Temperature of feedwater to boiler, deg F	472	364
Exit gas—CO_2 content, per cent	14.9	15.2
Heat content of coal (as fired), Btu per lb	13695	13095
Steam pressure, psi	720	722
Air pressure at burners, in. water	1.92	2.8
Draft in furnace, in. water	0.139	0.22
Draft at boiler outlet, in. water	1.9	2.98
Draft at economizer outlet, in. water	4.31	....
Draft at air-heater outlet, in. water	6.81	10.65
Steam temperature, deg F	726	690
Temperature of air leaving air heater, deg F	574	554
Temperature of gases leaving boiler, deg F	1038	658
Temperature of gases leaving economizer, deg F	697	....
Temperature of gases leaving air heater, deg F	379	376
Temperature of feedwater entering economizer, deg F	350	....
Combustion space per lb of coal per hr, cu ft	0.78	0.92
Heat absorbed by water in economizer, Btu per lb	133	....
Heat absorbed by water and steam in boiler, Btu per lb	742.3	864
Refuse, per cent of fuel (as fired), per cent	11.82	15
Rate of heat absorption per lb of fuel, KB	11.54	10.95
Rate of heat absorption per sq ft of steam-generating-unit surface per hour, KB	13.75	11.47
Efficiency of steam-generating unit, per cent	84.3	84.0
Average draft loss, in. water	6.67	10.43
Average air-pressure loss, in. water	5.93	9.65
Steam purity, ppm	0.6	0.7

ing differences in the two types of boiler units. Table 4 gives comparative test data taken from each of the boilers. These data are presented for general information and do not indicate performance under stable conditions. The forced-circulation and the natural-circulation units have been and are consistently operated at 85 per cent efficiency. Steam requirements for normal operation of the pumps of the forced-circulation unit amount to 10,000 lb per hr, or 3 per cent of the boiler capacity. This power requirement imposes an additional loss of over-all steam output, resulting in a lower inherent over-all efficiency of the forced-circulation type of boiler unit.

With the high make-up previously mentioned, and with the boilers being of necessity maintained on the line for considerable periods, it has been observed that the natural-circulation units form considerable sludge in the vicinity of the downcomer tubes at the top drum. In the forced-circulation unit, this sludge with additional scale particles collects on the orifice strainers tending to plug the control orifices. On two occasions this plugging of orifice strainers has resulted in tube starvation and failure. It has been particularly observed that at no time have scale particles been encountered on the natural-circulation units receiving the same feedwater and treatment. The exact location of the origin of these scale particles has not yet been determined, but they appear to form in the region of the distributing headers ahead of the orifice strainers. Chemical composition of these scale particles is identical with that of the soft sludge which is formed in the natural-circulation boilers.

Removal of the sludge is accomplished in the natural-circulation boiler by the conventional tube turbining of the downcomers with final removal at the mud drum. Build-up of this sludge indicates that a scheduled outage once each 6 months is sufficient for its removal with little danger of serious difficulty for longer periods. In the forced-circulation unit, sludge and scale removal is accomplished by removing and cleaning all the orifice strainers. Past operating data indicate that sludge and scale particles in the forced-circulation unit require a boiler outage each 60 days for inspection, thus reducing its availability, as compared to the natural-circulation boilers.

During internal inspections of the circulating pumps for the forced-circulation unit, it has been revealed that at times considerable build-up of scale takes place on the pump impellers with indications of scale breaking off and entering the orifice headers. As previously described, the circulating pumps have an arrangement of bleed-in and bleed-out connections which permit a small amount of boiler feedwater to enter the pump. Water consultants agree that under pH conditions existent in the feedwater, a reaction between the soluble phosphate present in the boiler water and the hardness present in the incoming feedwater results in a precipitation of calcium phosphate which forms a scale rather than the normal soft sludge precipitated in conditioned boiler water. It has been deemed advisable to maintain this "in" leakage to the pump at a minimum, and differential-pressure regulating valves have been installed to maintain a constant differential pressure automatically in the bleed-in line to the pump over the circulating pump discharge pressure. Prior to this installation, manual adjustments were made with probable erratic results.

In an effort to improve the availability of the forced-circulation unit, and to minimize the work required for sludge and scale-particle removal, two external strainer drums have been installed in each of the pump discharge lines as near as possible to the orifice headers. These strainers have openings of exactly the same size as those in the header orifice strainers, and each strainer has a free area more than twice that of the total for all the orifice strainers. These strainers were installed at a time when it was believed the scale particles were being formed between the steam drum and the distribution headers, including the pump impellers. It was hoped these strainers would thus effectively remove all of the scale particles and cause scheduled boiler outages only to clean the external strainers. Results observed on two occasions, since installation, reveal conditions in general to be no better than prior to their installation. It is apparent that scale is formed on the distribution headers ahead of the orifice strainers, a region of non-heat-absorbing surface, a condition which is contrary to that found in the conventional boilers. No definite conclusion concerning scale formation at this location has been advanced, although suspicion has been pointed to velocities, to mechanical agitation due to the pumps, and to minimum chemical-reaction time in the drum.

The apparent failure to remove the scale particles by mechanical means in the forced-circulation unit and the desirability of preventing sludge accumulation in both types of boilers has led all concerned to an attitude that the problem is one of chemical treatment and elimination of the formation of the sludge and scale rather than removal of scale particles after they have formed. A modified chemical treatment is now being used to overcome this condition and so far the results appear favorable.

The forced-circulation type of boiler has proved to be much more stable than the natural-circulation boilers from the standpoint of water level with varying operating conditions. The fact that the forced-circulation boiler is equipped with an economizer which furnishes the feedwater to the drum at 480 F, as compared with 340 F feedwater entering the natural-circulation drum, is given as one factor for such stability. Another factor is that the rate of circulation is relatively constant for forced circulation at either maximum or minimum rating. This pumping rate is 5100 gpm, or 1,900,000 lb per hr which gives a ratio of circulated water to steam of $5^1/_2$ to 1 at rated load.

In a natural-circulation boiler the rate of water circulation increases with rating, probably resulting in considerable turbulence in the steam drum at higher load conditions. With a slight lowering in drum pressure due to suddenly increased steam demand, the volume of steam bubbles in the generating tubes will increase sufficiently to cause a rise in water level in the drum, since the generating tubes enter the drum below water level. Such a rise, often called "swell," tends to upset feedwater regulation. This condition can only occur in the natural-circulation type of boiler unit. Consequently, the forced-circulation boiler-water level will be upset to a lesser degree by rapid and large load changes regardless of feedwater control than with the natural-circulation boiler. However, the natural-circulation units have, on a number of occasions, experienced load changes from 200,000 to 400,000 lb per hr without seriously upsetting water level and with no sign of carry-over. The ability of the operators and of the feedwater control to handle these conditions indicates that natural-circulation units can take rapid load swings without serious water-level upsets.

The expectation that the forced-circulation boiler with its closely spaced $1^1/_4$-in. tubes would present a nearly solid water-cooled metal surface to the furnace and that slag would not so readily adhere to this surface, has not been realized. Wall soot blowers were installed slightly above burner level to aid in removing this slag deposit. It appears that the greater number of spaces between the tubes for a given width of wall in the forced-circulation unit offers a larger number of points for slag to adhere, thus making it possible for the slag to bridge over more readily in this type of boiler than in the natural-circulation unit with larger wall tubes and greater spacing. With coals of the quality now burned, with an ash-fusion temperature between 2300 F and 2400 F, wall blowers or lancing will not be required for the natural-circulation units.

Tube ruptures at different times on both types of units afforded

the opportunity for boiler operators to compare urgency of removing the boiler from service. With chemical operating units unable to reduce steam demand safely at a moment's notice, the forced-circulation boiler, in one case, was kept on the line at full rating in excess of 3 hr, despite a complete wall-tube rupture. This type of boiler, with its relatively smaller-diameter tubes and small control orifice in the water inlet, in addition to the fact that the generating tubes terminate above the water level in the drum, can suffer a complete tube rupture without the necessity of immediate removal from the line. This is an extremely vital factor in chemical processes where shutdowns must be made in an orderly and safe manner. In the two instances when rupture occurred in a wall tube or generating tube on the natural-circulation units, it was necessary to take the boiler off the line immediately, resulting in pressure upsets to chemical process. A natural-circulation boiler, upon loss of any tube, immediately becomes inoperable.

Conclusion

In summarizing and outlining the characteristics found in these two types of boiler units at the Koppers plant, it is not the intent to suggest advantages or disadvantages of any particular type of unit. That the forced- and natural-circulation principles are radically different would of necessity present different operating characteristics.

The forced-circulation unit has demonstrated particular operating assets as follows:

1 The ability to suffer loss of a tube and continue to operate until other apparatus is brought on the line to take its place, or until load can be sufficiently reduced so that boiler capacity is not required.

2 Water-level stability under conditions of rapid and large load changes.

The natural-circulation units have particular operating assets as follows:

1 Greater availability in comparison with forced-circulation type.

2 Less sensitivity to changes in feedwater conditions, rendering it more suitable for high make-up with high solids in the feedwater.

3 Lower susceptibility to scale formation under equivalent feedwater conditions.

The operating record of both of these types of steam generators reveals their ability to supply process-steam requirements under many adverse conditions. Chemical production of butadiene and styrene has been maintained at high rates throughout the operating period to date, and over-all operation should compare favorably with that of any other steam plant of its size and operating conditions.

Discussion

J. M. Harvey.[4] The character of the water in the Ohio River and its tributaries is such that many problems must be met which do not exist where water of more stable quality, such as water from the Great Lakes, is utilized for boiler-feed purposes. Acid mine drainage and extensive use of these streams for disposal of industrial waste and sewage from the heavily populated surroundings make for extreme variation in the character of the river water.

Characteristic is the trend from a bicarbonate condition, with low dissolved solids and increased suspended material during the heavy run-off in the spring and early summer to a highly acid condition, with high dissolved solids and low suspended material during the low flow in late Summer and Fall. Localized pollution and the presence of river pools may further influence the character of the river water at a particular plant intake.

Generally, the periodic variations in dissolved solids will follow a somewhat fixed pattern with minimum and maximum concentrations reaching about the same value each year, and with the average concentration varying not more than a few ppm over a period of several years. However, combination of conditions may produce considerable deviation from the normal pattern as shown by Fig. 5 in the paper, charting variations during a period when industrial activity was at an unprecedented high.

The wide flexibility of the water-conditioning facilities installed at Kobuta made it possible to cope with the variations imposed and maintain boiler-water conditions within closely defined limits. Even so, characteristics of the river water exert their influence and have been productive of adherent sludge deposits in the boilers. Specific measures were adopted several months ago which we believe, and which are proved in part, will solve this problem.

In January, 1945, silicate treatment was introduced to one of the natural-circulation boilers. At this time the routine water-testing program was expanded to include determination of the silica concentration in the boiler water and saturated steam. These tests soon established the fact that there was no significant difference between the silica present in the steam generated by the boiler receiving silica treatment and the boilers not receiving this treatment.

Since all the natural-circulation boilers had been mechanically cleaned just prior to the introduction of silicate treatment, inspection of these boilers in June, 1945, after a 6-months' run, afforded the opportunity to compare the appearance of the internal surfaces of the boilers. The boiler subjected to silicate treatment was decidedly cleaner than the other natural-circulation boilers.

On the basis of these observed conditions, silicate treatment was extended to all the natural-circulation boilers as well as the forced-circulation boiler. Inspection of the forced-circulation boiler in October, 1945, after a run of approximately 60 days on silicate treatment, revealed accumulation of scale particles in the orifice headers to about the same extent as noted during previous inspections. However, finely divided sludge, which had been found intermingled with the scale particles on previous occasions, was almost entirely absent. The scale particles are identical in chemical composition to the finely divided sludge formerly found along with the scale particles. This is indicative possibly that the scale particles originate at some point in the circulating system where the finely divided sludge adhered and attained more coherent and compact form under conditions existing at that point. Thus we believe there is some possibility that the scale particles will be eliminated with continued operation on the silicate treatment.

In connection with the silicate treatment, it is also worthy of mention that inspection of the 35,000-kw back-pressure turbogenerator in October, 1945, showed the complete absence of any deposits. This inspection was made after approximately 2 years of operation, and silicate treatment was in effect during the last 9 months of operation.

H. J. Klotz.[5] The authors of this paper are to be commended for the unprejudiced presentation of the comparison of the operating characteristics of forced- and natural-circulation boilers of the same capacity installed side by side in an industrial plant and performing under identical conditions as regards load and

[4] Engineering Service, Hall Laboratories, Inc. Pittsburgh, Pa.

[5] Chief Power Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Mem. A.S.M.E.

feedwater. The writer's connection, as consultant to the Office of Rubber Reserve, with the operation of the Koppers plant, has afforded an opportunity to observe that the plant-operating organization has been equally unprejudiced in its handling of the boilers. There has been not the slightest tendency to favor either type of boiler, but a conscientious effort has always been evident to meet any special requirements imposed by the two types of boilers. The fairness of their comparison has thus been assured.

It is, of course, too much to expect that a comparison in one plant only could form the basis to permit final and definite conclusions as to whether one or the other type of boiler should be installed. The principal offsetting features would appear to be the ability of the forced-circulation unit to continue in operation following the loss of a tube, an important item in many processes, compared with the greater availability of the natural-circulation boilers. Corrective measures adopted at the Koppers plant will probably obviate the importance of this comparison as it is believed the causes of the tube ruptures in the natural-circulation boilers have been greatly minimized, and the lower availability of the forced-circulation boiler should eventually be overcome by further progress in chemical treatment of the feedwater plus possible additional mechanical features to prevent sludge accumulation on the orifice strainers.

The extremely variable nature of the make-up water at this plant makes the latter an important consideration and one which, until it is overcome, points to the desirability of the natural-circulation boiler for comparable water conditions.

It is important to note that the pumps required for circulating water with the forced-circulation boiler have imposed no operating problem.

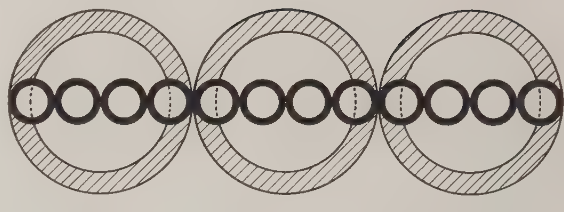
Experience at the Koppers plant indicates that the practicability of the forced-circulation boilers for industrial plants has been sufficiently proved to warrant additional installations, and that any plant considering the choice between the two types would be safe in basing its decision on the relative investments and space requirements after assuring itself that any possible feedwater limitations would be satisfactorily met.

W. S. Patterson.[6] The authors have refrained from comparing differences in design in the two boiler types unless these differences have, up to the present time, resulted in operating assets in favor of one type or the other. However, since forced circulation, applied to large steam generators, is new in this country, it will be of interest to point out some of the reasons for the difference which will be noted from a study of the paper, and some of the potential assets.

Small-diameter tubes are used in forced-circulation boilers because a high flow resistance and velocity in the tubes are desirable, and because a high fluid velocity in the tubes can be realized with a minimum quantity of circulating water handled by the pumps. Small-diameter tubes can have very thin walls, resulting in low hot face skin temperatures and low temperature stress in the tube metal even with very high rates of heat absorption. There is also a saving in weight of tube metal and weight of water in the tubes; but the advantages of small tube diameter are most apparent in high-pressure boilers as will be seen from Fig. 7 of this discussion.

Although bifurcated tubes have been used in natural-circulation-boiler furnaces, this is the first application of trifurcated furnace tubes. Such an arrangement decreases the number of header connections, orifices, and access openings. Before this boiler was placed in operation Pitot tubes were installed in each tube of several of the trifurcated furnace elements to check relative water distribution. The results are presented in Table 5 of this discussion.

[6] Assistant Manager, Forced Circulation Boilers, Combustion Engineering Co., Inc., New York, N. Y. Mem. A.S.M.E.



Tube Diameter	4 in.	1 in.
Wall thickness	.48 in.	.15 in.
Velocity ratio	1	4.7
Tube weight ratio	1	.335
Water weight ratio	1	.213
Temp. gradient ratio	1	.326

Fig. 7 Furnace-Wall-Tube Comparison for Tangent Spacing; Small-Diameter Versus Large Diameter

(Velocity ratio is based upon same circulation ratio for both arrangements.)

TABLE 5 DISTRIBUTION OF FLOW IN TRIFURCATED FURNACE CIRCUITS

Date of tests	Dec. 12 to 21, 1943
No. of tests on each circuit	6
Boiler output	250,000 to 300,000 lb per hr
Boiler pressure	730 to 740 psig

	Front furnace wall								
Trifurcate no.	1			13			22		
Tube no.	1[a]	2	3	37	38	39	64	65	66
Flow through tube, per cent	33.6	34.8	31.6	35.0	34.8	30.2	31.8	37.8	30.4

	Rear furnace wall			Side furnace wall		
Trifurcate no.	6-1			1		
Tube no.	181	182	183[a]	1[a]	2	3
Flow through tube, per cent	27.4	35.8	36.8	26.8	37.8	35.4

[a] Denotes tube in corner of furnace.

The convection boiler, called the "secondary generator," also employs $1^1/_4$-in-OD tubes, but they are fitted with $1^3/_4$-in-high fins and disposed horizontally in a bank below the economizer. These tubes are bifurcated at the end adjacent to the control orifice so that each orifice serves two circuits which are intermeshed, with forged return bends and long-radius bends alternating through the length of each circuit. The tubes are arranged in staggered relationship with $2^1/_2$-in. horizontal and $3^1/_2$-in. vertical spacing, and because of their small diameter and transverse flow of the gases, a very high heat-transfer rate is obtained without the use of baffles and without resorting to a high gas velocity. The surface installed per cubic foot of space in the secondary generator is about twice that of a conventional boiler and each element is independently removable through a large door at one end.

The "water-cooled beams" mentioned in the paper, support the secondary generator and both economizer sections. They are shown at three elevations in Fig. 2 of the paper, and consist of two parallel sets of three headers each, with series flow through each set. The tubes supplying the outer end of the uppermost headers are shown coming off the main distributing header. The water passes through the uppermost headers, thence by external connections to the inner end of the lowest headers, through the latter to the outer end, and thence through external connections to the single secondary-generator inlet header containing the control orifices. The two intermeshed circuits of each secondary-generator element terminate in separate discharge headers which support the lower economizer section. From these headers the steam-water mixture is discharged to the boiler drum.

Since the steam drum is not required to support a bank of steam-generating tubes, it is located entirely outside the setting as shown in Fig. 2 of the paper, which is sometimes possible but

rarely convenient in designing a natural-circulation boiler. In the case of thick drum shells exposed to hot gases, drum protection must be provided, which introduces a maintenance item for the operator; also, where a boiler baffle abuts the drum shell, a positive and tight seal must be maintained to prevent fly-ash erosion of the drum due to baffle leakage. These problems are absent when the drum is outside the setting and therefore represent potential operating assets. It will also be noted that all the drum-tube connections in Fig. 2 of the paper are in one quadrant of the shell, thus reducing to a minimum the amount of thick shell plate required.

The authors point out an operating disadvantage of a high-capacity boiler containing a very small quantity of water, when the rate of blowdown is high due to high make-up of water having high total solids. However, the low weight of water is also associated with low weight of tube metal and this combination results in a lower fuel requirement to place a forced-circulation boiler in service and less heat loss on cooling down. This would be a distinct asset in the case of a boiler subject to week-end shutdowns in many industrial plants. In combination with forced circulation, it also permits entry to the furnace for maintenance within 8 hr of the time the boiler is taken off the line, even in the case of a large high-pressure unit with a wet-bottom furnace. Also, with the increasing popularity of acid cleaning of boilers it is an operating asset to require only one quarter of the amount of acid, flushing water, and neutralizing solution, to say nothing of the time saved in filling draining, etc.; and 2 min operation of the pumps at hourly or half-hourly intervals will completely mix the solvent, re-establishing a uniform concentration and uniform temperature, both of which are important.

Fig. 2 of the paper shows that each pump is equipped with a shutoff valve at suction and discharge. It was originally intended that only one pump would be in operation and the other held as a spare with suction and discharge valves closed. With the method of operation used at this plant the spare pump idles at 400 rpm and with all suction and discharge valves open. The check valve located at each pump discharge has a bypass which permits operating the spare pump in this manner but limits recirculation through the idle pump. The shutoff valves have never been used during operation because the availability of both pumps has been 100 per cent.

The authors give a figure of 10,000 lb per hr steam consumption for circulating-pump drives but the heat extracted in the turbine is only about 200 Btu per lb or 2,000,000 Btu per hr. The fuel equivalent of this is about 0.50 per cent of the fuel fired at maximum rated capacity and part of this energy is returned to the boiler water in passing through the pump.

The occurrence of scale chips in the orifice headers has not been uniform in all headers, and this was originally explained by the fact that the first supply tubes coming off the main distributing header go to the front or burner wall; the next group at each end go to the side-wall headers and the center group go to the rear-wall header. When the sludge and/or scale chips have been present in serious quantities, the front-wall header has generally contained the most and the rear-wall header the least. In fact, the strainers were not removed from the rear wall during one entire year of operation, and beginning in April, 1944, and ending July, 1944, there was one operating period of 30 days, followed by another of 60 days, during which no significant deposits occurred in any of the orifice headers.

The differential-pressure regulating valves on the bleed-in lines to the pumps were recommended by both the pump and the boiler manufacturer during the design stages but were not installed until the spring of 1945.

The authors mention "minimum chemical-reaction time in the drum," as a possible reason for scale in the orifice headers of the forced-circulation boiler. However, it has been noted that a ring of sludge forms even in the perforations of the submerged feed pipe, which is an indication of very quick reaction.

The furnace-wall soot blowers on the forced-circulation boiler were furnished by the boiler manufacturer in 1943 in lieu of removing superheater elements, because during early operation the steam temperature exceeded the guarantee and it was found that removing ash from a band of the furnace walls just above the burners would give 50 deg F steam-temperature control.

One reason for keeping the steam temperature down on the forced-circulation boiler has been because the natural-circulation boilers do not deliver steam at the predicted temperature (see authors' Table 4), and this results in different steam temperatures in the two turbine leads, which is objectionable. Predicted performance of the forced-circulation boiler was deliberately based upon moderately dirty furnace conditions. At maximum load the contract average gas temperature leaving the furnace was 2110 F. A test in January, 1945, at maximum load using an 18-point traverse with a high-velocity shielded platinum thermocouple at three elevations at the superheater entrance gave the following averages:

Upper 1870 F; middle 2015 F; lower 2025 F; during the test the CO_2 entering superheater was 16.3 per cent, the average steam temperature 760 F; the furnace-wall blowers had not yet been placed in service and the walls were moderately dirty. In reference to the authors' conclusion on availability, it might be added that the boiler manufacturer definitely recommended that the forced-circulation boiler be taken out of service every 60 days for inspection, because this boiler was a new development and frequent inspections would avoid the possibility of any major damage to the unit, as well as decrease the possibility of a forced outage. Nevertheless, the availability of the forced-circulation boiler averaged close to 95 per cent from December 1, 1943, to April 1, 1945, and it would be of interest to learn how much greater availability was realized on each of the natural-circulation boilers to justify the conclusion drawn.

S. T. Powell,[7] H. E. Bacon,[8] and L. G. von Lossberg.[9] Personal inspections of these boilers and studies of the feedwater treatment by the writers confirm the authors' observations. In particular, chemical and x-ray analyses of deposits removed from the forced-circulation boiler at the last inspection, representing the conditions during the feeding of sodium silicate to control magnesium in the sludge, show that the desired precipitate, magnesium silicate, is now being produced. However, our recent inspection indicates that troublesome deposits are still occurring in the orifice strainers.

Our experience with water conditioning for the forced-circulation type of unit has also shown that feedwater of high quality is required to prevent deposits in the control-orifice strainers. Also, the relatively small volume of water contained in this type of boiler limits the concentration of constituents that may be tolerated in the feedwater.

The favorable comparison with regard to the operating characteristics of the forced- and natural-circulation boilers for the production of process steam is noted with interest.

It is our opinion that the operators of the plant are to be complimented upon maintaining continuous operation of the steam-generating facilities and related equipment under the rigorous conditions imposed throughout the 2-year war-production period.

[7] Consulting Chemical Engineer, Baltimore, Md. Mem. A.S.M.E.

[8] Engineering Assistant to Sheppard T. Powell, Consulting Chemical Engineer, Baltimore, Md. Mem. A.S.M.E.

[9] Engineering Assistant to Sheppard T. Powell.

W. F. Ryan.[10] The authors are to be congratulated on the impartiality with which they have presented the facts in regard to these two types of steam-generating equipment. If they have any preference it cannot even be inferred from their paper, and the reader is left to draw his own conclusions. This form of presentation is as commendable as it is unusual in engineering literature.

Steam consumed by the circulating pumps means little unless translated into energy; the latter is obviously a charge against the efficiency of the forced-circulation unit. No discussion is offered of the greater space and building costs required for a natural-circulation design. At Kobuta, it appears that the building design was determined by the space required for the larger boiler and hence no appreciable savings accrued from the installation of a single forced-circulation unit. These savings might have been considerable, however, had the building been designed for four forced-circulation units.

It is unfortunate that Figs. 2 and 3 of the paper, showing cross sections of the two boilers, are not shown to the same scale. The tabulations, however, show that the natural-circulation boiler has 4500 cu ft greater furnace volume. For four boilers, this amounts to 18,000 cu ft which might have been saved had the building been designed for the forced-circulation type of boiler only. Weights are not given, but from the data on heating surface, tube sizes, and water volumes, it is clear that the weight of the natural-circulation boiler, filled with water for test, is more than twice the weight of the forced-circulation unit. As shown in the tables, the operating weight of water in the natural-circulation boiler is more than 4 times that in the other type. These factors materially affect the cost of building steel and foundations and should ultimately have an appreciable effect on the cost of equipment as well.

These are the reasons, of course, that manufacturers and operators are willing to spend time and money on such developments of the art. What we all want are better boilers at less cost than we now have. It is apparent from this paper that forced circulation is a constructive step in this direction.

[10] Assistant Chief Mechanical Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Fellow A.S.M.E.

Authors' Closure

The silicate treatment as mentioned in Mr. Harvey's discussion definitely showed promising results as to the elimination of sludge in the natural circulation boilers. Duplicate treatment in the forced-circulation boiler has not produced hoped for results and all inspections of this boiler unit since the treatment was initiated revealed scale at the orifice strainers in amounts equal or greater than on previous inspections. Sludge accumulations were, however, somewhat reduced. Operations at this boiler plant are now such that continued observation of this method of feedwater treatment is not possible as the forced circulation boiler has been out of service since early in November due to low steam demand to process. The elimination of the troublesome scale at the orifice strainers remains as an important factor in securing greater availability of the forced circulation unit, and the authors believe that feedwater consultants will overcome this difficulty by introduction of other suggested treatments, should the silicate not produce the desired results. The divergence in results of this duplicate treatment of the feedwater for both types of boilers further demonstrates that the natural circulation boilers are less susceptible to scale formation under equivalent feedwater conditions than the forced circulation unit.

Mr. Patterson's discussion of certain design features and potential operating assets of the forced circulation boiler enhances the value of the paper for design engineers and operators of boiler plants. The authors, as operators, are naturally inclined to overlook design features unless they materially affect the boiler operation or are modified to improve the operation. The references to savings in weight, "secondary generator," "water-cooled beams," and drum location will no doubt be of great interest to all readers, and the authors concur in his discussion on certain minor advantages gained by low weight of water and lower requirements for acid cleaning. The further explanations of Mr. Patterson in regard to pump operation, soot blowers, and steam consumption of the pumps contributes much to enlighten the reader on points which the authors unintentionally might have slighted.

The conclusion that the natural circulation boiler has greater availability has been demonstrated by the operating records of the two types of units. While operating at near peak demands, the outage schedule required that each natural circulation boiler be taken off the line every six months with the outage time limited to three days. The outage schedule of the forced circulation unit required an outage each two months with outage time limited also to three days. The amount of maintenance work required on both types was relatively equal. Thus it can be seen that on a percentage of availability basis the natural circulation boilers would reach 98.36 per cent or 6 days outage per operating year while the forced circulation boiler would reach 95.08 per cent or 18 days outage per operating year. While the percentage availability difference does not seem too great, it is significant that the scheduled outage time required by the forced circulation unit amounts to three times that of the natural circulation boilers.

The authors appreciate the discussions of all these engineers and the favorable comments contained in these discussions. That the Kobuta Plant has had such an outstanding operating and performance record is partially due to the suggestions and recommendations of these outstanding men of the steam-generation field.

It is well to point out that since preparation of this paper all the 3 in. tubes between the pump discharge header and the orifice headers were mechanically turbined. The amount of scale removed during this operation has led us to believe that these tubes are the source of scale particles, and their scheduled turbining at times of boiler outages would do much to increase the operating time between outages of the forced circulation boiler.

It will be noted that the operating assets of the forced circulation unit are inherent for this type of boiler unit, while those of the natural circulation boiler are not limited by its type. It would thus seem that further developments to overcome any deficiency of operating assets for forced circulation will enable us to attain better boilers at less cost as asked by Mr. Ryan.

Operating History and Performance of 2000-Psi Forced-Circulation Boiler at Somerset Station of Montaup Electric Company

By G. U. PARKS,[1] W. S. PATTERSON,[2] and W. F. RYAN[3]

In this paper the operating experiences with the high-pressure forced-circulation boiler at Somerset station of the Montaup Electric Company are given in considerable detail. Special test and research activities on the installation are reported in this and companion papers. The conclusion is reached that the selection of a bare-tube, slagging-bottom, radiant, forced-circulation boiler for this high-pressure application has been verified by its operating record.

The forced-circulation boiler at Somerset Station was installed during 1940–1942, as part of the topping installation. Initial pressure and temperature were dictated by the necessity of obtaining 25,000 kw by topping a 375-psi 750 F plant. The size and shape of the boiler were determined by the floor space and headroom available in a vacant space in the existing boiler room which had been allocated in 1925 to a future boiler of 120,000 lb per hr capacity.

The subject boiler has a guaranteed maximum output of 650,000 lb per hr at 960 F at the superheater outlet. Drum safety valves are set to blow at 2000 psi, but average drum pressure over an extended period of time has not exceeded 1925 psi. Pressure at the superheater outlet averages approximately 1850 psi, which gives name-plate pressure at the topping-turbine throttle. The reheater takes exhaust steam from the topping turbine at 400 psi 603 F, and exhausts to the station's low-pressure header at 380 psi 765 F. The boiler and related equipment have been described in detail in references (1)[4] and (2).

Normally the station burns West Virginia semibituminous coal or bunker "C" fuel oil, and the boiler was therefore designed to use either fuel or any combination of the two. Since the station's pulverized-coal system is of the storage-bin type, and it is necessary to supply this new slagging-bottom furnace and the older dry-bottom furnaces from the same supply, the furnace and combustion equipment were so designed that coals with ash-softening temperatures between 2200 F and 2600 F could be used satisfactorily.

It is the purpose of this paper to discuss the operation problems encountered and their solution and present the results of research and tests on circulation, tube temperatures, and performance.

Although all equipment manufacturers have been very co-operative, wartime conditions have made it impossible to effect some changes when they were desired. It is hoped that the complete topping installation can be discussed at some later date when changes and modifications in equipment in no way related to the boiler have been completed.

After 3 months' preliminary operation, delivering steam through the by-pass line, the boiler was placed in commercial operation with the turbine on October 26, 1942, and has been taken out of service 54 times from that date to the present time, November 1, 1945.

The longest outages, which are identified by numbers in Fig. 1, were chargeable to equipment as listed below the chart. Table 1 supplements this list and shows that there were in the first three years of commercial operation, 8 forced outages and 46 scheduled outages, about half of which were directly chargeable to research and tests. Most of the outages charged to research merely consisted of taking the unit off the line for a few hours for checking dissolved and suspended solids in the boiler water. The scheduled outages charged to other equipment were sometimes also used for scheduled boiler maintenance if time permitted. On the other hand, the forced outage in 1944, charged to slag in the superheater, was caused by malfunctioning of the air-flow recorder and operation with low excess air.

Fig. 1 also shows average output calculated from steam-flow-integrator readings for the duration of each operating period, without correction for pressure or steam temperature. The meter is calibrated for 1825 psi and 960 F.

Although the boiler has been operated in excess of 650,000 lb per hr with both coal and oil firing, output during the past two years was normally limited to approximately 585,000 lb per hr, and the average monthly load is under 500,000 lb per hr. This has been for the following reasons, rather than desire or necessity due to boiler limitations:

(*a*) The necessity for keeping spinning reserve on the system at other points.

(*b*) The large amounts of hydro-generated power which tie lines make available during evening and night hours.

(*c*) The rebuilding of the 10-in. pressure-reducing and desuperheating station which leaves only the 4-in. station with 200,000 lb per hr capacity in case of turbine failure.

(*d*) Vibration and shaft whip in the topping turbine under certain load conditions.

(*e*) Inability of the low-pressure plant to absorb more than 500,000 lb per hr when either low-pressure unit is not on the line.

Since October, 1942, it has been impossible for the station to receive coal from the normal suppliers in the Pocahontas No. 3 seam or the Sewall seam. Based on monthly composite samples, the heat content of coal as received has varied from 13,000 Btu per lb to 14,000 Btu per lb, moisture from 4 to 8 per cent, ash from 6 to 13 per cent, and grindability from 70 to 100 per cent on the Hardgrove scale. Ash-softening temperature has varied between 2100 and 2800 F. A substantial part of the coal burned has been from either West Virginia or Pennsylvania strip operations.

[1] General Manager, Montaup Electric Company, Fall River, Mass. Mem. A.S.M.E.

[2] Assistant Manager, Forced-Circulation Boilers, Combustion Engineering Company, Inc., New York, N. Y. Mem. A.S.M.E.

[3] Engineering Manager, Stone & Webster Engineering Corporation, Boston, Mass. Fellow A.S.M.E.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Power, Industrial Instruments and Regulators Divisions, and the Joint Research Committee on Boiler Feedwater Studies, and presented at the Annual Meeting, New York, N. Y., November 26–29, 1945, of The American Society of Mechanical Engineers.

TABLE 1 OUTAGE TABULATION

Reason for Outage	Annual Period Nov. 1 to Nov. 1			3-Yr Period
A - Forced Outages	1942-43	1943-44	1944-45	1942-45
Furnace Tube Failure	2	1	1	4
Reheater Tube Failure	0	0	2	2
Slag in Superheater	1	1	0	2
Total Forced Outages	3	2	3	8
B-Scheduled Outages				
Research and Tests	7	15	1	23
Piping External to Boiler	2	2	2	6
Low Pressure Station Repairs	0	2	1	3
Pressure Reducing Station	1	0	1	2
Miscellaneous Boiler Items	1	1	0	2
Safety valves	0	0	2	2
High Pressure Turbine	0	1	1	2
Acid Cleaning Boiler	1	0	0	1
Soot Blowers	1	0	0	1
Reheater Repairs	0	0	1	1
Boiler Vent Valve	0	0	1	1
Electrical Trouble	1	0	0	1
Drum Internal Changes	0	1	0	1
Total Scheduled Outages	14	22	10	46
Total Outages	17	24	13	54

Due to overworked mills, the amount of pulverized coal through a 200-mesh screen has rarely been as high as 80 per cent, and averages 70 per cent, with 3 to 4 per cent retained on 60 mesh.

The normal operating crew has been the boiler operator, stationed at the control panel, and his assistant who keeps the log sheet and also operates blowdown, soot blowers, and attends to changing and cleaning oil burners. They are assisted part of the time by an oiler on the feeder and fan floor, who also works on the fans and feeders of five low-pressure boilers, and by a pump tender who operates the high-pressure heaters and drip pumps, as well as the circulating pumps. When coal from normal suppliers is available, the latter also acts as ashman. Feedwater chemistry is directly controlled by the station's own laboratory on a single-shift basis. The high-pressure-turbine operator acts as chemist on the other two shifts.

The arrangement of the boiler unit as originally installed with changes and modifications from 1942 to 1945, inclusive, indicated in notes, is shown in Fig. 2. Following is a summarized description of operating experiences with the boiler unit including a discussion of changes made and the reasons for them.

Operating History and Changes

Evaporating Surface. A furnace-tube failure occurred within the first 2 months of preliminary operation. It was in the front wall a short distance below the roof and in the second tube from the right side wall. A short section of tube was welded in for repair. Microscopic examination of the failed section indicated overheating but the reason for overheating was not immediately determined.

A second tube failure occurred in January, 1943, in the furnace roof in the same tube that had failed previously. This failure was occasioned by pitting on the inside surface of the tube, one of the pits having extended entirely through the tube. A large amount of solid deposit found on the inside surface of the tube was considered to be a contributing factor to the corrosion. Repair was made by welding-in a replacement section, and an attempt was made to locate any source of free oxygen which might have contributed to this failure. Consultants were also engaged to study the conditions which caused the deposition of foreign matter on the inside tube surface.

The third tube failure occurred in February, 1943, while this investigation was in progress. This failure was in the left side wall, third tube from the rear wall. Microscopic examination of the failed portion indicated overheating and the cause was thought to be plugging of one of the orifice strainers.

Each time the boiler was drained substantial amounts of foreign matter were found in the orifice headers, particularly near

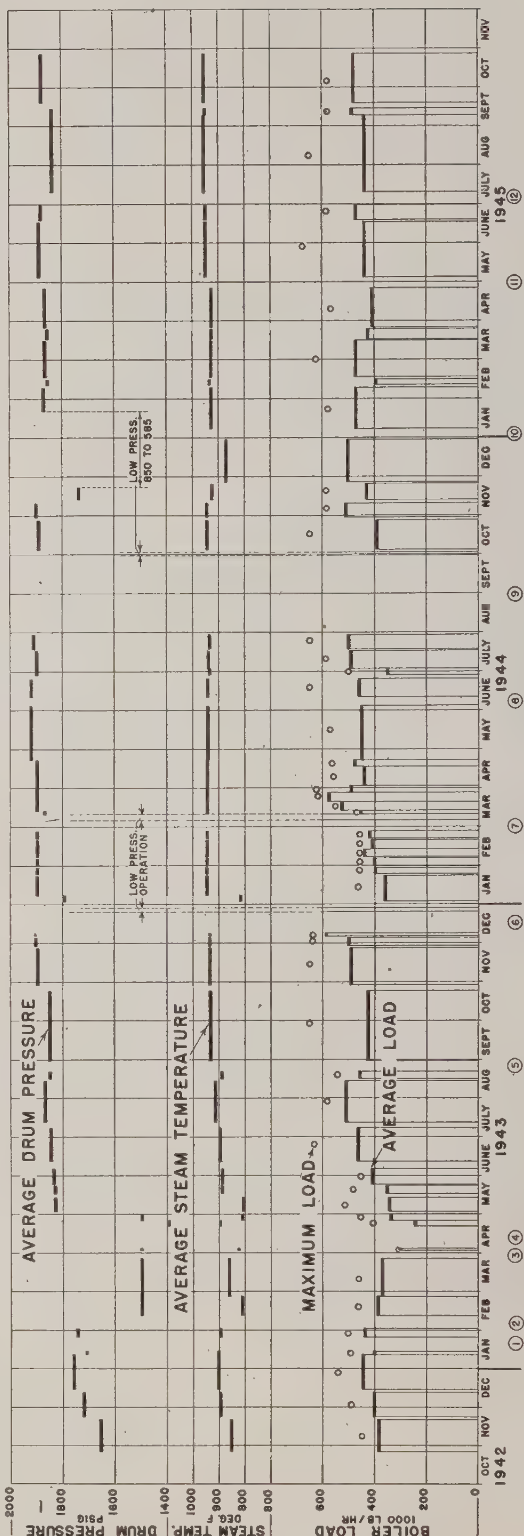


Fig. 1 Load, Temperature, Pressure, and Outage Chart

The numbered outages were charged as follows:

1 Furnace tube failure
2 Furnace tube failure
4 Acid-cleaning boiler
5 Miscellaneous boiler changes
7 Miscellaneous boiler changes
8 Drum internal changes
10 Pressure-reducing station
11 Reheater repairs

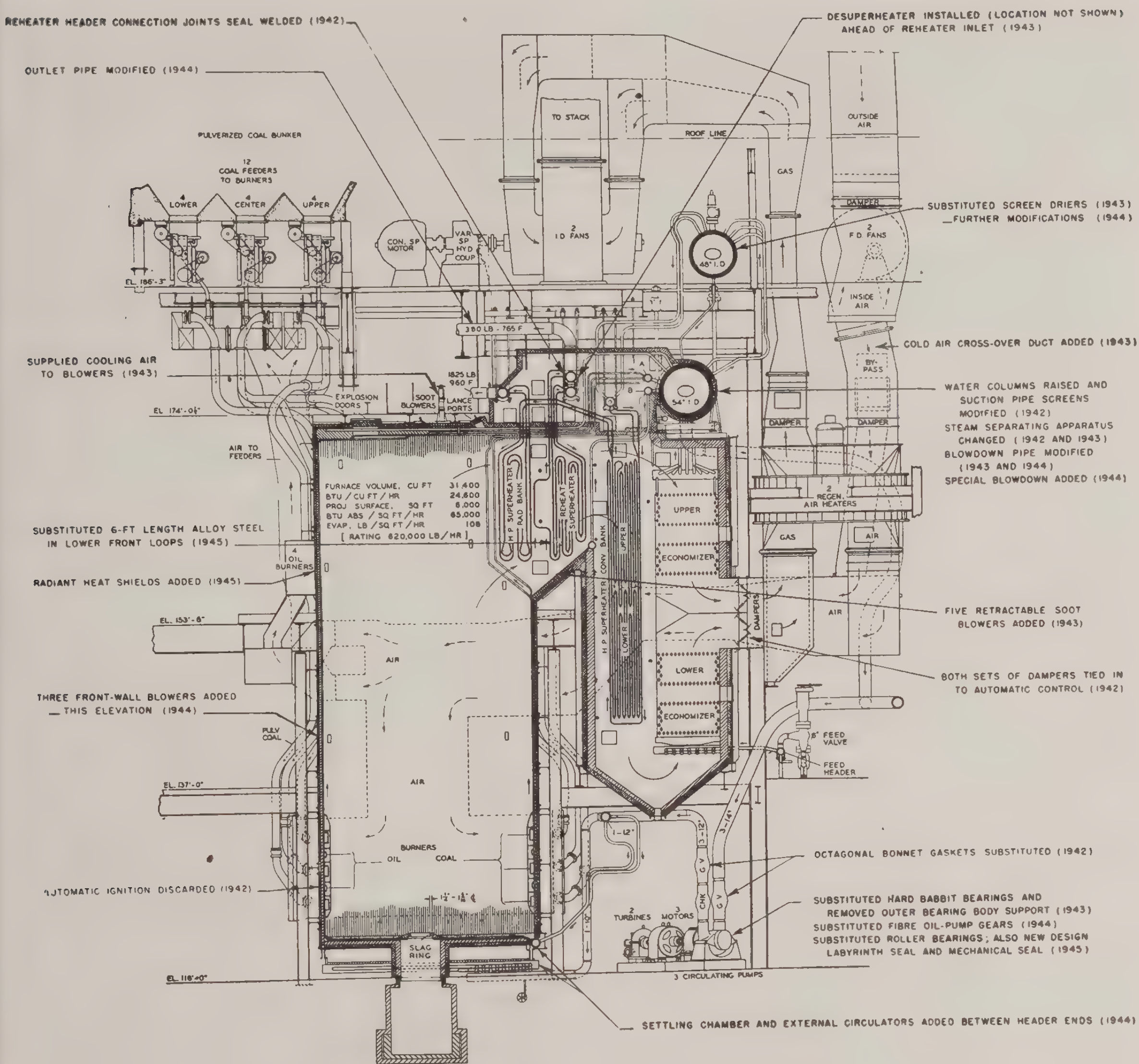


Fig. 2 Arrangement of Boiler Unit as Originally Installed With Changes From 1942 to 1945, Inclusive, Indicated by Marginal Notations

the ends. Prior to the second tube failure, a sample of header washings was found to contain 66 per cent iron shot, 17 per cent mill scale, and miscellaneous oddments such as steel shavings, turnings, chips, and weld splatter. A sample collected at the time of the second failure contained 30 per cent weld splatter, 20 per cent mill scale, and 13 per cent iron shot, plus oddments.

The 54-in. boiler drum had been shotblasted in the shop after the junction headers had been attached and, although the headers had been blown out with compressed air, it was impossible to do an adequate job, since the header ends were closed at the time. The boiler had been carefully washed out and boiled out, with the orifices and strainers removed, prior to operation. Headers were washed at every available opportunity but the recurrence of foreign particles and substantial amounts of mill scale persisted for some time in the lower headers. There was also evidence of chemical hide-out whenever the boiler was taken off the line.

In order to clean tube surfaces thoroughly and remove mill scale, the boiler was washed with inhibited hydrochloric acid in April, 1943. All orifices were removed and the circulating pumps were not operated during the washing and it was therefore impossible to maintain uniform temperatures. In spite of this, tests showed that substantial amounts of mill scale and foreign material had been removed. Prior to this time feedwater was normally treated with standard sodium chemicals (phosphate, hydroxide, and sulphite). After the acid cleaning the potassium treatment developed by Dr. R. E. Hall was introduced (3, 4).

In December, 1943, after 7 months' operation with potassium treatment, including silica feed, the fourth tube failure occurred. This was the third failure in the same tube (second tube from the left side wall), and was in the section of the tube which passed across the roof of the furnace. The inside surface of several other tubes examined at the same time was coated with thick

sludge and scale, but the actual failure was considered to be caused by a plugged orifice strainer. The evidence of such plugging was missing, due, no doubt, to the disturbance created in the header when the tube failed; but repeated failures only in tubes served by orifices close to the dead end of the headers indicated the most probable cause to be sludge accumulations at the header ends.

Study of orifice-header pressures showed that there was sufficient pressure difference between the rear and the side headers so that if a small circulator were installed between each end of the rear header and the rear end of each side-wall header, there would be sufficient flow between the interconnected headers to preclude any sludge accumulation near the orifice strainers which serve the tubes in which failures had occurred. Circulators of $1^1/_4$ in. were installed between the rear header and the side headers, each provided with a small baffled settling chamber with sampling connections and blowdown valves and upper closure plugs to permit measuring the depth of sludge in the chambers. After establishing the fact that they collected a considerable quantity of sludge, they were connected into the existing drain lines, and are blown down once a week as part of the operating routine. Since the headers have been interconnected, very little sludge accumulation within them has been evident and very few of the strainer openings are found closed. In fact the strainers have not been removed in nearly 2 years except a few for inspection purposes.

At the time of the fourth failure representative tubes were sampled. There were substantial sludge deposits on all of them and acmite scale on most. Therefore, before the boiler was returned to service it was again acid-cleaned, using inhibited hydrochloric acid, supplemented by a fluoride. In order to insure a thorough acid cleaning at recommended temperatures, one circulating pump was operated 1 min during each half hour of the cleaning period, and thermocouples installed for the purpose showed that recommended temperatures were being maintained. Immediately after the cleaning and washing period the circulating pump was dismantled and the impeller removed for examination. There was no evidence of any attack by the acid.

Since this fourth failure the potassium treatment of feedwater without silica has been used: Tube samples taken in February, 1944, August, 1944, and September, 1945, from several parts of the walls showed very little sludge accumulation, and what did exist was not adherent but easily brushed away. There was evidence of very little, if any, corrosion or pitting of wall surfaces.

Periodically throughout the 3-year operating period of this boiler the external surfaces of representative furnace tubes have been checked for external corrosion. Although an adherent "enamel" coating of fused slag has been found, similar to that which has been credited with causing corrosion in other furnaces, actual corrosion has been negligible or nonexistent up to October 26, 1945.

On October 26, 1945, about 22 months after the last internal cleaning of the boiler, three small leaks occurred in the rear furnace wall, one located less than 1 ft above the floor and two about 20 ft above the floor in a different tube. The tubes which had failed were found to be heavily coated with sludge and, on the fire side, iron-oxide scale had formed between the sludge and the tube wall, indicating that the sludge had caused overheating of the tubes. An adjacent tube near the floor in the rear wall was in similar condition on the inside but a tube almost directly opposite, near the floor of the front wall, although heavily coated with sludge, had most of the deposit on the casing side where it could not cause overheating. This latter tube was perfectly clean at an elevation 17 ft above the floor, and a tube in the right side wall 15 ft above the floor was also nearly as clean as the same tube had been a month earlier.

There is evidence to indicate that a recent disturbance has caused this trouble because on this last outage there was appreciable external tube corrosion on some of the rear-wall tubes in the region of the lowest leak. Due to failure of the water seal below the slag hole and malfunctioning of the flue-gas oxygen recorder a deficiency of air near the walls and consequent abnormally high gas temperatures probably occurred.

Before the boiler was again placed in service it was acid-cleaned, using one circulating pump a few minutes of each hour to maintain uniform solvent temperature throughout the unit. Two tubes were sampled after the cleaning to make sure of a thorough job.

Superheater: High Pressure. There have been no failures, changes, or maintenance in either the convection bank or radiant bank.

It had been originally expected to obtain automatic superheat control by having the controller operate only the by-pass dampers located at the outlet of the upper economizer. However, it was found that manual operation of the main-passage dampers at the outlet of the lower economizer was necessary to obtain quick response. The controls were therefore modified so that both sets of dampers are automatically operated, that is, as the by-pass dampers open the other dampers close a proportionate amount.

Superheater: Reheat. The steam reheater which operates at 400 psi was provided with bolted ball-and-socket joints where the elements are attached to the headers. On account of numerous leaks during the first 2 months and the difficulty of access to the joints for refacing, it was decided to seal-weld all these joints.

Operation at high rates of output indicated that reheated-steam temperatures would be too high at maximum load for continuous operation with carbon-steel reheater elements. It was decided not to reduce the heating surface but rather to correct the difficulty by installation of a manually controlled steam-atomizing spray-type desuperheater. This was installed between the high-pressure turbine and the reheater in July, 1943, and has given satisfactory control.

It will be noted from the side-elevation view of the boiler that the lower portion of the front loop of the reheater is not effectively screened from the radiant heat of the furnace, particularly if the rear-wall screen tubes are kept clear of ash accumulation. Soon after a soot-blower overhaul in August, 1944, reheater-tube failures in this front loop began to occur. These were quickly repaired on each occasion by welding-in a new piece of carbon-steel tubing, but at the end of April, 1945, the lower end of the front loop, including the lower bend, of all 58 elements, was replaced with chrome-molybdenum-titanium alloy tube sections about 6 ft long. This is the same material as used in the front section of the superheater for 960 F steam temperature and since the reheated-steam temperature is only 765 F, no further trouble from this cause is anticipated. All other parts of the reheater are quite effectively shaded from radiant heat.

Economizer. There have been no failures or changes in the economizer. There has also been no maintenance except to seal up a few air leaks at the economizer doors and in the suspended arch where the hanger rods pass through the roof.

Steam Drums and Drum Internals. Because of the forced-circulation feature it was possible to operate the pumps with cold water and observe the flow of water into and out of the drum with the manholes open. Before placing the boiler in operation the flow of water toward the two suction pipes at the ends of the drum was photographed, and it was observed that a vortex was created over each suction-pipe nozzle, and that air was drawn into the suction pipes when the water level was low. The design of suction-pipe screens was changed and a coverplate was incorporated to eliminate the undesirable turbulence.

In reference (1) Fig. 13, it will be noted that the feedwater from

the economizer outlet discharges into the drum through tubes entering the bottom of the shell and that a deflecting baffle was installed directly above the outlet of these tubes. It was found that this resulted in low shell temperatures between this region and the suction-pipe nozzles. This baffle was removed in December, 1942, in order that the feedwater, which is at a temperature 100 deg F below that of the boiler water, can mix with the latter more quickly and thus eliminate this undesirable condition.

The original steam-separating means consisted of a combination of "hydraulic baffle" and metallic baffles in the wet drum and a combination of baffles and rod-type drier in the dry drum. During the first few months of operation the boiler was sensitive to carry-over when the water level approached the center line of the drum and, because the feedwater regulators were not functioning properly, the capacity was limited to 400,000 lb per hr, although higher capacity could have been carried with better water-level regulation. Minor modifications were tried but showed little promise of correcting the trouble so an entire new set of wet-drum internals of the "reversing-hood" type was substituted in October, 1942 (see Fig. 13 of reference 1 of Bibliography).

The blowdown line in the boiler drum as originally installed was found to be unsatisfactory after the reversing-hood type of internals had been installed because considerable feedwater was entering the blowdown piping. To correct this condition, six open-ended collecting pipes were extended upward above the water level so that concentrated boiler water from the hoods would discharge directly into the collecting pipes.

Although the reversing-hood steam separators in the wet drum resulted in satisfactory steam purity at loads up to 500,000, it became obvious in June, 1943, that further improvements would be necessary to meet the guarantee of 0.50 ppm solids in steam. Furthermore, the gaskets of the drum internals had deteriorated during the acid wash of the boiler in April, 1943, so that in August, 1943, the wet-drum internals were regasketed with copper-covered asbestos and screen driers were substituted for the rod driers in the dry drum.

Early in 1944 special laboratory evaporating apparatus, which permitted the determination of solids in steam by evaporating a 4-liter sample every 24 hr, was used to check the solids in steam. It was found that at maximum guaranteed output the guaranteed 0.50 ppm was being exceeded. Part of the difficulty was believed due to the fact that the steam-discharge tubes from the dry drum are not uniformly spaced along the drum but are arranged in three groups. A new set of low-velocity screens was substituted and a perforated baffle plate was installed ahead of each group of steam-discharge tubes to produce uniform velocity through the screens.

The steam line supplying the high-pressure soot blowers was originally connected to a nozzle near the center of the dry drum. The steam was taken out of the drum from the space between the screen driers and the steam-discharge tubes, and it was found that when the soot blowers were operated there was an increase in moisture in the steam in the saturated header of the superheater. To improve this condition the soot-blower steam nozzle was extended to take steam from the drum before it passes through the final drying screens. Gasketed joints in the wet drum were seal-welded at this time.

Excellent steam quality is now obtained from this unit and the results are included elsewhere in this paper.

Although there are generally only 10 to 20 lb of sludge in the wet drum after several months of operation, it has been noted that when the fires are extinguished preparatory to taking the boiler out of service, the blowdown water sample always becomes very turbid. The theory was advanced that the most quiescent zone in the wet drum during the steaming period is on the bottom of the shell between the 18 feedwater discharge nozzles which project into the drum about 2 in. This would then be a logical place for sludge to accumulate; but when the feedwater flow rate is reduced on shutting down the boiler the flow of water toward the suction pipes might wash this deposit away. Accordingly, in February, 1944, a special blowdown pipe was placed on the bottom of the drum and connected externally, through valves, to the continuous blowdown pipe. A study was made of the increase in turbidity of the blowdown sample on opening these valves and it is now part of operating routine to operate this special blowdown once a week.

In a boiler of this type it is possible to check the circulating ratio quite accurately by determining the solids concentration in the water at the circulating pumps and in the concentrated water leaving the steam-generating circuits, provided a representative sample of the latter can be collected. Fig. 3 shows a novel means

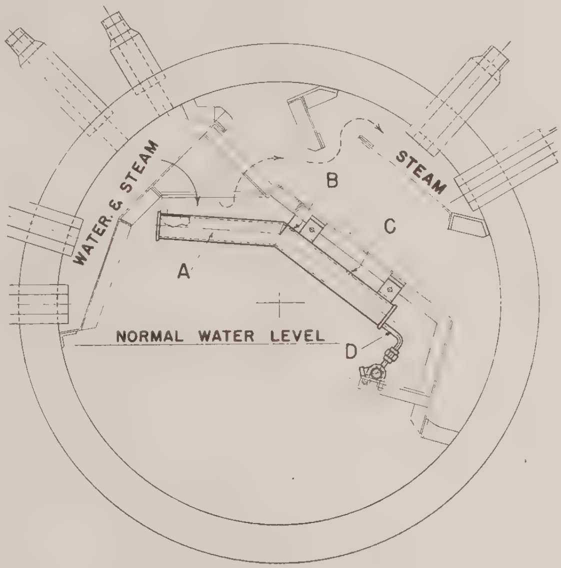


Fig. 3 Arrangement of Continuous Blowdown Collectors

of collecting this sample from the concentrated water discharged into the drum above the water level. Six such blowdown collectors are uniformly spaced across the drum. The horizontal portions *A* are located directly under the slots of the reversing hoods which discharge water and steam into the drum. The upper surface of pipe *A* is perforated and designed to collect water at a rate several times greater than the maximum rate of blowdown. Any steam which enters pipes *A* is vented through an opening at *B* and the excess water is discharged through an opening at *C*. By proportioning the latter opening properly, a high collecting rate can be maintained in pipe *A* without filling the pipe, and the collecting rate is also independent of the rate of blowdown from the lower end of the collector at *D*. These blowdown collectors have been very satisfactory and result in minimum blowdown rate since they collect only the concentrated water which is discharged from the steam-generating tubes.

Air Heaters. The air heaters have given excellent service and there has been practically no maintenance. The steam soot blowers have never been used nor have the heaters been washed. They have been air-lanced on three or four occasions during scheduled outages but the necessity for this has never been urgent. In accordance with manufacturers' recommendations, compressed-air blowers were recently installed to bring the installation up to date.

Soot Blowers. Fig. 2 shows the location of the low-pressure soot-blower elements and also the position of the high-pressure retractable blowers located in the roof. These latter blowers had a sleeve around the blowing nozzle providing an annular passage to permit room air to be drawn in by furnace draft for cooling the blowers. After about a year of operation it was found that the portion of the sleeve projecting into the furnace was burned off and that the blowing elements were also badly damaged. In October, 1943, air under pressure was piped to these blowers to provide additional cooling.

When the boiler was operated at 550,000 lb per hr or higher, continuous output, it was found that an accumulation of ash occurred in the crotch and on the slope below the reheater. There was a tendency for this accumulation to grow quite rapidly and extend up the sloping screen tubes to the superheater. Five retractable mass-type blowers were installed on the slope in August, 1943.

In 1944 the coal was poorer than normal and quite variable. Slag was sometimes difficult to remove from the furnace walls. The soot blowers in the superheater-reheater zone were badly damaged and practically useless and it was found that the steam pressure at the high-pressure blowers was considerably below the recommended operating pressure. Since the coal situation might get worse before it got better, it was decided to review completely the soot-blower situation with the manufacturer and take steps to improve the situation during the August, 1944, annual outage.

During overhaul, additional bearings were provided for several of the low-pressure elements, particularly in the superheater-reheater zone. Replacement elements with larger nozzles were installed between reheater and front superheater and the wall boxes which were out of alignment were relocated. Operating mechanism of all blowers was overhauled and some changes made to facilitate operation. Two of the roof blowers were moved down to the side walls to replace wall blowers at the elevation of, and just in front of, the bottom of the slope below the reheater to assist the slope blowers which were not very effective near the side walls. The two side blowers thus replaced, plus one blower "stolen" from the upper front wall above the auxiliary burners, were located in the front wall at the elevation of the first row of observation doors above the burners.

During the process of overhauling all the retractable blowers it was found that part of the damage and malfunctioning was due to lack of lubrication. A lubrication schedule was set up and central lubricating stations were installed so that several blowing elements could be lubricated from each station. With the exception of the slope blowers there was very little soot-blower maintenance during the following year and at the end of that time all of the blowers except those in the slope were still in good condition. The cleaning effectiveness of the blowers as a whole has also been very good since this August, 1944, overhaul. Part of this improvement can be attributed to the higher steam pressure available at the high-pressure blowers after repairing the pressure-reducing valve through which this steam passes.

At the present time the nozzles of the slope blowers do not retract far enough to protect the blowing heads and there is considerable maintenance. Also, in spite of frequent lubrication, some of these blowers have become inoperative due to the action of heat on the lubricant. Correction of these difficulties is still being followed by the soot-blower manufacturer, but in fairness to him it should be pointed out that space limitations had made it necessary to install blowers with only 5-in. travel. The blowers must advance far enough so as not to cut the tubes in the slope and at the same time retract far enough to be out of the heat, and the blowing head is about 2 in. long with three large nozzles covering about 100 deg of blowing arc.

Circulating Pumps. Except for two leaks at the circulating-pump heads, the circulating pumps performed satisfactorily during the first 3 months of preliminary operation but toward the end of October, 1942, trouble developed in the labyrinth of one of the pumps. As described in reference (1) the labyrinth provides a means of sealing the pump against leakage of hot boiler water at nearly saturation temperature by introducing lower-temperature feedwater at a higher pressure at one end of the labyrinth passage. The labyrinth is a close-clearance passage between moving and stationary parts and as the sealing water flows toward the leak-off point adjacent to the packing gland the pressure breaks down to about 150 psi. An increase in leak-off pressure could be caused by increased flow through leak-off piping and is an indication of increased sealing-water quantity due to excessive wear in the labyrinth. This is how the first labyrinth trouble was detected. When the pump was opened up, the labyrinth surfaces were found to be badly worn, but new parts were installed without any changes except to realign the pump.

The *A* pump which had a labyrinth failure in October, 1942, had another similar failure in November, 1942, and toward the end of December, 1942, the labyrinth in the *B* pump failed. New sleeve bearings were installed in the *A* and *B* pumps during January, 1943, and the bearings of all three pumps were changed to hard babbitt in February, 1943.

During the overhaul for bearing replacement on the *C* pump, it was discovered that the pump could not be rotated until the support under the coupling end of the bearing body was removed, and in March, 1943, the supports were also removed from under the coupling end of the bearing body on both *A* and *B* pumps. Indicators were used to record the amount of vertical and horizontal movement of each bearing body when the support was removed. It was found that these supports had been exerting a strain on the bearing body which probably was resulting in misalignment of the shaft in the labyrinth. It was later found that temperature changes within the pump and/or piping would produce a few thousandths of an inch movement of the floating end of the bearing body and that the upper and lower parts of the bearing-body bracket were not at the same temperature. Insulation was added to the lower part of the bracket to correct this latter condition.

Also in March, 1943, the *A* pump labyrinth failed for the third time but the trouble was found to be corrosion or erosion of the casing in the labyrinth section. This was repaired by applying stainless steel and remachining. The other two pumps were overhauled during April and May, 1943, and there were no further failures in the labyrinths. However, there were several failures of *B* and *C* oil-pump spur gears later in the year; fiber gears were substituted and gave much longer life than the bronze gears.

Bearing troubles were encountered not only with the circulating-pump motors but on other pump motors in the station. The trouble in the case of the circulating pumps was diagnosed as being due to expansion of the pump shaft toward the motor a sufficient amount to put a thrust on the motor bearings. This movement had not been allowed for in the design of the motors. After the replacement of several sets of motor bearings, the pump manufacturer corrected the trouble by the installation of limit stops in the flexible couplings located between motor and pump.

Although there was no trouble or maintenance on the circulating pumps since the spring of 1943, except oil-pump drive gears, the amount of water used to seal these pumps greatly exceeded the manufacturer's predictions. The pipe carrying this water to the pumps from boiler feed pump discharge was therefore much overloaded, thus making it necessary to carry a pressure at the feed pumps about 200 psi above boiler pressure in order to have sufficient sealing-water pressure at the pumps. In order to be able to operate with closer clearances in the labyrinth, to reduce the sealing-water requirements, the pump manufacturer decided to rebuild the bearing body and to substitute roller bearings in

place of the babbitted sleeve bearings. The design of the labyrinth was also changed to accommodate a mechanical seal between the sealing-water injection point and the impeller chamber. This seal reduced the leak-in of seal water to the pump practically to zero. Further reduction in sealing-water requirements was accomplished by the close clearances in the labyrinth passages so that the over-all result was to reduce the injection water, leak-off and leak-in, to amounts far below the original predictions, and the high-pressure leak-off was eliminated completely. One of these pumps was rebuilt about February 1, 1945, and there has been no perceptible increase in sealing-water requirements. This pump was dismantled after 2 months' operation and the boundary surfaces of the labyrinth passage were in excellent condition. The second pump has just been changed over and the third pump will have also been rebuilt by the time this paper is presented.

Water-Level Indicators. Four water-level devices were installed on this boiler, consisting of a recorder and an Eye-Hye gage on the operating panel, a double-window Bi-Color gage with 21-in. visibility and mirrors on one end of the drum, and a double-window Micasight gage with 18-in. visibility on the other end of the drum. The indirect-type gages have been very satisfactory but they do not give a true indication when the pressure is below the design value, due to change in water density.

The center of both water-level gages was originally located 10 in. below the drum center line. This meant that in one gage the water level was still visible when there was only $6^1/_2$ in. of water in the drum, which would expose the modified suction-pipe cover plate, and result in steam entering these pipes. The water columns were both raised so that the normal water level and center of gage are now 5 in. below the center of the drum.

Illumination of the double-window Micasight gage was not sufficiently bright to permit the water level to be observed from the operating floor. The manufacturer decided that it would be a physical impossibility to line up the double-window gage for good visibility because there was only one spot on the operating floor from which the gage could be seen and part of the window was obstructed by a steel beam. A single-window gage of 12-in. visibility was substituted in August, 1943, but after a short period of operation the image of the water level became too faint to be seen from the operating floor. In October, 1945, the gage manufacturer installed a new improved illuminator and mirrors to reflect the water-level image to a convenient point near the operating floor. Visibility of the image with new mica windows was very clear but decreased rapidly. It remains to be determined how often the mica will have to be renewed to maintain satisfactory visibility.

The Bi-Color gage has required frequent maintenance. It has been necessary to have a spare gage available for replacement due to the frequency of failures. One of the gages was modified by the manufacturer in February, 1944, but the modified gage was no improvement. The other gage was modified in a different manner in September, 1945, but failed after 8 days in service. One of the gages was recently assembled in an improvised manner and at the time of writing, has been in service 5 weeks which is the best record so far. However, visibility of the water line at the end of that time is only fair due to dirty mica.

Burners. The automatic ignition system used light oil, mechanically atomized and electrically ignited. From a control station it was possible to start up the oil pump, insert the ignition torches and ignite the oil. The principal trouble encountered was with the mechanism used to insert and retract the oil burners. The automatic features were discarded in favor of torches of steam-atomizing type using heavy oil and which are moved into position by hand.

The pulverized-coal burners are located in the four corners and fire tangentially. No changes have been made except to incline the lowest row slightly downward to improve slag fluidity on the floor.

There are four auxiliary oil burners in the upper front wall which are required to maintain maximum steam temperature at reduced load when the unit is operating with a clean furnace and 100 per cent oil fuel. These burners are not required to supplement coal-firing nor are they necessary with intermittent coal and oil firing. They have never been used and were found to be damaged due to radiant heat from the furnace after a year of disuse. The vanes and other metal parts were burned or distorted. They were repaired and removable Fahrite protecting plates inserted to shield them from the furnace heat.

Miscellaneous: Valves, Piping, Ducts. During the period of preliminary operation, with frequent starting and shutting down, considerable trouble was encountered due to leaks in the bonnet joints of the large gate valves in the suction and dicharge lines of the circulating pumps. These flanged bonnets were originally installed with ring gaskets of oval section. Leakage troubles were corrected by changing to gaskets of octagonal section.

Two outages were charged to safety valves because two shutdowns were scheduled to lap-in the seats of leaky valves.

There are two forced-draft fans and two air heaters with a common preheated-air duct below the air heaters. Although dampers were provided at the inlet to each fan and air heater and, in the cold-air by-passes, these dampers did not prevent the preheated air from leaking through to the idle fan, when only one fan was in operation, and burning the paint on the fan. A crossover duct was installed in 1943, between the cold-air outlet ducts of the two fans to correct this condition.

A program involving several changes in the exhaust-steam piping between the high-pressure turbine and the low-pressure station header is in progress at this station for the purpose of reducing the pressure drop between those points. The first change was to modify the outlet connection of the reheater, in August, 1944, when it was realized that a reduction in pressure drop of several pounds per square inch could be obtained. The reheater originally had a single outlet near the center of the outlet header, and two additional 6-in. outlet pipes were added.

Another piping modification was to install a by-pass line around the high-pressure feedwater heaters permitting both heaters to be out of service without having to shut down the high-pressure boiler. The by-passline also increases flexibility of operation by facilitating an increase in the quantity of low-pressure steam from the high-pressure boiler without increasing the output of high-pressure steam. Under certain load conditions this would obviate the necessity of placing another low-pressure boiler in service or in any event decrease the urgency.

Research and Tests

Special Provisions. Before the boiler was placed in operation more than 100 thermocouples were installed at various points throughout the unit for the purpose of checking operating metal temperatures on furnace-wall tubes, superheater, reheater, economizer, steam drum (lower), feedwater connections to drum and circulating-water connections to pumps. Nearly one half of these wereinstalled on the hot face of the furnace-wall tubes at an elevation about 3 ft above the furnace floor. Pressure taps and Pitot tubes were also installed at various points in the circulating system, as illustrated in Fig. 4. Actually, the Pitot-tube locations were further from the elbow than this sketch indicates.

During the course of subsequent special studies, after the boiler had been placed in operation, additional thermocouples were added to lower drum, circulating pumps, reheater, and on the furnace tubes at different elevations, bringing the total number of thermocouples installed up to about 275.

Steam-sampling connections were initially provided on super-

heater inlet and dry drum, but for special studies additional sampling connections were made at dry drum, reheater outlet, high-pressure-turbine inlet, and high-pressure-turbine outlet.

A special calorimeter was installed to check for the presence of steam in the suction pipes to the pumps and its operation is described in reference (5).

The method of installing the thermocouples and the technique of using Pitot tubes has been described in references (6) and (7).

Circulation. One of the first research studies was to check the total circulation and distribution of circulation to the four separate groups of parallel circuits comprising the four furnace walls. In each wall one circuit had been chosen for installation of a pressure tap on the downstream side of the orifice. The pressure difference between header and the downstream tap beyond the orifice could then be used to determine the rate of flow through the orifice, using calibration data reported in reference (8) Fig. 39, which were determined for the orifice assembly shown in Fig. 38, this being an exact replica of the arrangement used in the Montaup boiler. All the orifices serving the circuits of a given wall are the same size; and, since the pressure drop across the orifices is large compared to the drop across the parallel heated circuits, very little error is introduced by using the flow through one orifice to calculate the flow for the entire wall. Furthermore, all heated tubes of a given wall are nearly the same length and all have approximately the same exposure to heat. By measuring the pressure drop across four furnace-wall orifices, it was therefore possible to determine the total circulation of the unit since there is no other steam-generation surface or no other stray circuits except a connection to the economizer inlet; this is used to circulate boiler water through the economizer when the unit is being brought up to pressure preparatory to placing in operation.

The Pitot tubes in the two suction pipes to the circulating pumps provided another means of determining the total circulation, except that this would not include the sealing water which

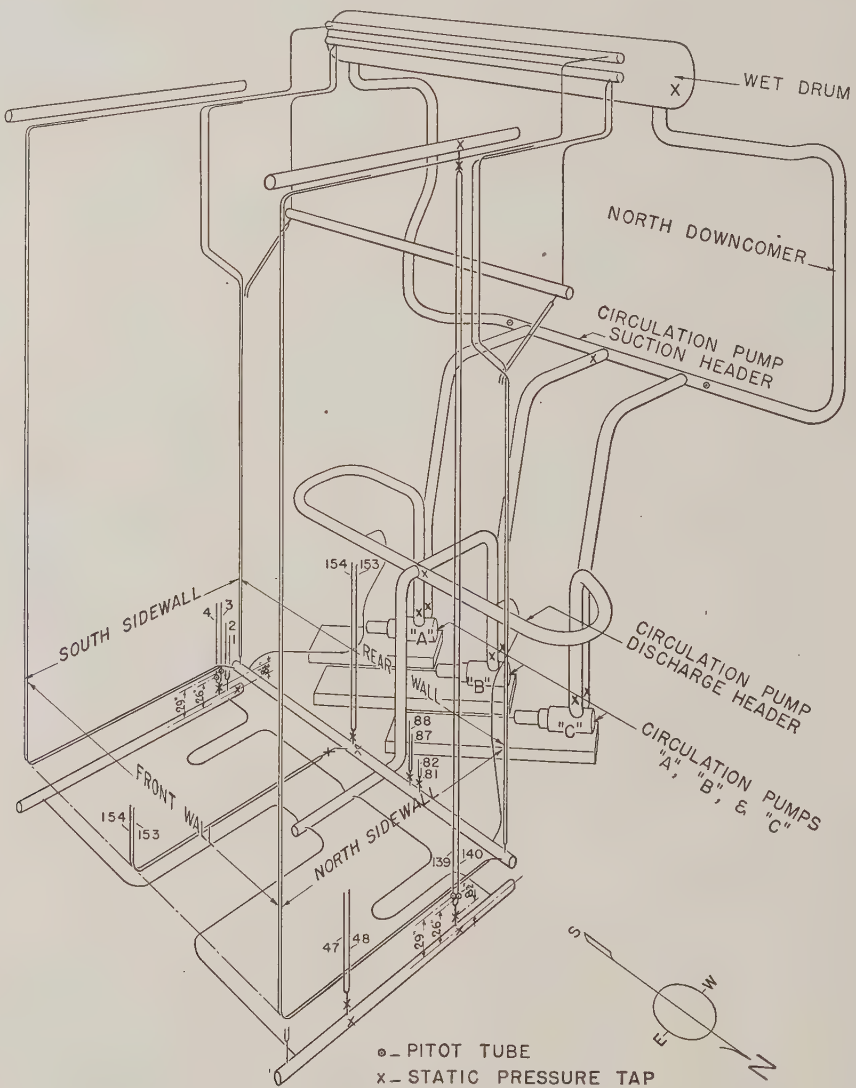


FIG. 4 ISOMETRIC SKETCH SHOWING LOCATION OF SPECIAL PRESSURE CONNECTIONS USED IN TESTS ON CIRCULATION

leaks into the pump. This leak-in can be checked by determining the difference between rate of feedwater flow and steam flow and of course making an allowance for blowdown. This was done on certain tests during 1944, but the flow data were taken during 1942 and 1943, and no correction for leak-in was made because the quantity was of the order of 0.50 to 1 per cent of the water being handled by the pumps, and the accuracy of the flow data did not warrant this refinement since it is not claimed to be correct within 1 per cent.

Measurement of the pump head provided a third means of checking the pump capacity because a pump characteristic curve had been determined by shop test. Unfortunately, however, these tests were conducted up to a capacity of only 5000 gpm and the curve had to be extrapolated for comparison with some of the field data.

A fourth method of checking the total circulation was by determining the difference in solids concentration in the water at pump discharge and leaving the evaporating circuits. The following equation shows the relationship of the variables involved

$$W/S = C_1/(C_1 - C_e)$$

where W = weight of water entering furnace circuits
S = weight of steam generated in furnace circuits
C_e = concentration of solids in water entering, ppm
C_1 = concentration of solids in water leaving, ppm

Since the feedwater is introduced at the bottom of the drum and does not mix with the steam in the drum, it is not heated to saturation before entering the suction pipe. There is, of course, some condensation of steam in both drums, and this would tend to make the weight of steam generated in the furnace circuits greater than the weight of steam metered at the superheater outlet. Consequently, the determination of W by using the weight of metered steam for S would give a value of W which would be lower than the true value.

When this method was first used the difference in concentration of chloride, sulphate, and total solids were all determined for comparison of the results but the time required for analysis of the water samples was too great to permit frequent checking. The results reported herein are based on analysis for chlorides only. The chloride concentration at the pumps was generally between 200 and 260 ppm, and it was considered possible with this concentration to determine the chloride by titration without involving an error of more than about 2 ppm.

Fig. 5 shows a plot of the total flow of water through the orifice headers, based on measured orifice pressure drop using a mercury manometer. The data were collected at different operating pressures and rates of output, but the total flow appears to be independent of these variables over a wide range. There is, however, a marked increase in circulation when the boiler is not steaming and when the water temperature is low.

Fig. 6 is a similar plot of flow data based on pressure differential measured by a mercury manometer across the Pitot tubes in the suction pipes. There is a greater spread in the data probably due to the effect of variable leak-in of sealing water, and the effect of the economizer circulating line which was open during the period when most of the downtake flow data were taken. However, this plot substantiates the previous statement that the total flow is quite independent of load and pressure. Figs. 5 and 6 illustrate the fact that the flow with one pump is about 70 per cent as great as with two pumps. Three pumps were not operated during the period when these data were taken, except at atmospheric pressure with cold water (100 to 125 F) under which conditions three pumps produced only 15 per cent more flow than two pumps.

Fig. 7 is a plot of measured flow rate in the suction pipes and

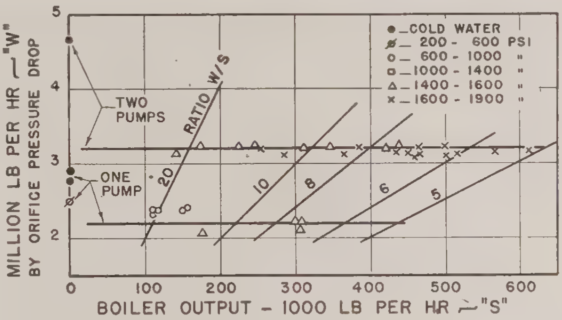


Fig. 5 Total Circulation to Lower Furnace-Wall Headers Based Upon Pressure-Drop Measurements Across Orifices

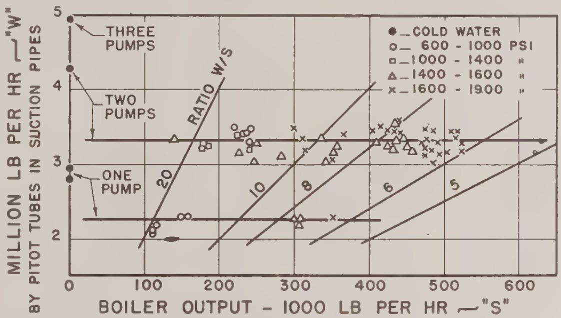


Fig. 6 Total Circulation Entering Pumps Based Upon Pressure Drop Measurements Taken on Pitot Tubes in Suction Pipes

measured pressure difference between pump outlet and pump inlet, uncorrected for acceleration across the pump, superimposed on characteristic pump curves. The pressure difference was determined by dual connections to a 12-in. Crosby test gage having 5-lb graduations which may have introduced considerable error in the pressure data. This may explain why some of the data do not fit the characteristic curves too well, but a small error in the flow rate may also be partly responsible. Also, the pump-head curves are plotted for 1750 rpm, and the pumps are motor-driven and operate at considerably less than full load amperage, particularly when the water density is low, so the actual pump speed may have at times approached 1790 rpm. The pump speed was not determined but the extent to which this variation would displace the head curve is indicated on the chart. The system resistance lines drawn through the test data can be considered as only approximate since the data for one pump operation are meager and subject to large error in the pressure reading.

Fig. 8 is a plot of total water flow calculated from chemical concentration at pump discharge and leaving evaporating circuits, the latter based on water samples from the continuous-blowdown collectors shown in Fig. 3. The data are plotted for two-pump and three-pump operation and two chemists worked independently on these results which explains why two different kinds of points are plotted for each series of tests. No allowance was made for steam condensed in the drum, which explains why total circulation by this method is lower than the results of the two other methods.

It has been shown that the total circulating-water flow calculated from the pressure drop across four orifices checks very well with the total flow calculated from Pitot-tube measurements in the suction pipes. The distribution of circulating water to the

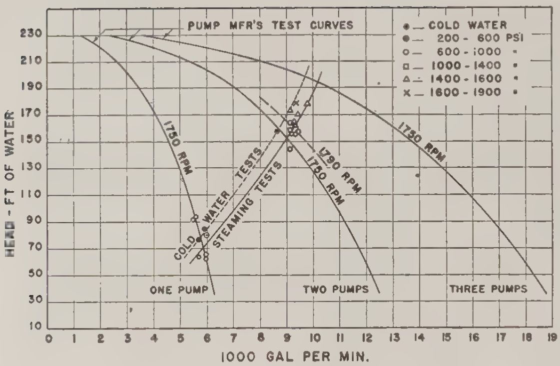


FIG. 7 CIRCULATING-PUMP CHARACTERISTICS, COMPARED TO MEASURED PUMP HEAD PLOTTED AGAINST SUCTION-PIPE FLOW RATE

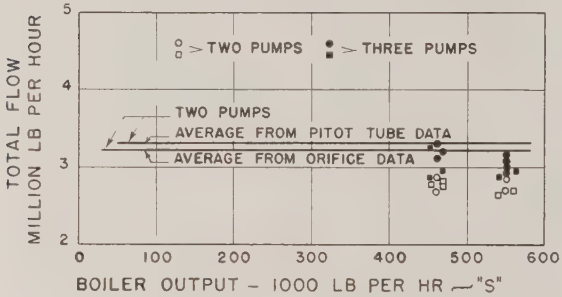


FIG. 8 TOTAL CIRCULATION CALCULATED BY CHEMICAL METHOD COMPARED TO AVERAGES BY OTHER METHODS

walls can therefore be quite accurately determined from the same orifice pressure-drop measurements and the results of several tests are given in Table 2. When this unit was designed the orifices were selected so as to control the water flow to each wall in proportion to the calculated distribution of heat to the walls. The table shows how closely this distribution and control are being realized with only two orifice sizes, namely, 0.34 and 0.40 in., and also illustrates that the distribution is practically the same for any load or pressure.

The distribution of water to the two suction pipes was found to be practically 50–50 when the two outside pumps or the center pump was operating. It was sometimes nearly equal when the center and one of the outside pumps were used but on other occasions the distribution was 45–55.

Fig. 4 shows that pressure drop could be measured across two orifices in the right side wall. Equal pressure drops were often noted but the maximum difference in flow rate on any test was 1.5 per cent.

A study was also made of water distribution to the two legs of bifurcated circuits in the right-hand and left-hand walls. These circuits are indicated in Fig. 4. In each circuit the orifice pressure drop was measured to determine water flow to the circuit and average velocity in the two tubes, assuming equal distribution. A calibrated Pitot tube was installed in each leg to check the distribution of flow. The average velocity based on Pitot-tube differential was lower than the velocity calculated from the measured orifice flow but the distribution was nearly equal as indicated by Table 3.

A calorimeter similar to that described in reference (5) was used to check for the presence of steam in the suction pipes. These tests involved not only the measurement of water temperature in the pipes but a check on the heat exchange in cooling this water

TABLE 2 COMPARISON OF MEASURED DISTRIBUTION OF CIRCULATING WATER WITH CALCULATED DISTRIBUTION OF HEAT TO STEAM-GENERATING SURFACES

Test Date	Operating Pressure	Boiler Output	Distribution of Circulating Water Front	Rear	Sides
6-20-43	1825	285,000	37.2%	30.5%	32.3%
6-16-43	1840	450,000	37.8	30.1	32.1
6-19-43	1850	515,000	37.6	30.2	32.2
6-21-43	1875	565,000	37.7	30.5	31.8
6-24-43	1880	610,000	37.7	30.5	31.8
* 6-5-43	380	0	37.4	28.9	33.7
6-5-43	1840	252,000	37.4	29.9	32.7
6-5-43	1850	385,000	37.3	29.9	32.8
6-4-43	1850	465,000	37.3	30.1	32.6
4-22-43	1500	175,000	37.7	29.7	32.6
4-23-43	1500	245,000	37.6	29.5	32.9
4-23-43	1500	340,000	37.4	30.0	32.6
4-23-43	1500	420,000	37.3	30.1	32.6
7-20-42	850	110,000	35.6	30.0	34.4
7-22-42	900	156,000	34.3	30.1	35.6
* 8-31-42	0	0	37.0	29.0	34.0
* 8-31-42	0	0	38.4	30.1	31.5
8-31-42	0	0	37.0	28.6	34.4
			Calculated Distribution of Heat		
			37.4%	29.6%	33%

*One pump used for these tests; all other tests based on two pump operation.

TABLE 3 DISTRIBUTION OF FLOW IN BIFURCATED CIRCUITS

Date, 1943	6-16	6-18	6-20	6-21	6-24
Boiler output, 1000 lb/hr	450	515	285	565	610
Boiler pressure, psig	1840	1845	1825	1875	1880
Right-hand Furnace Wall -	Circuit No. 70 Tubes No. 139 and 140				
Flow Thru Tube 139 - %	49.8	50.0	49.1	49.0	48.1
" " " 140 - %	50.2	50.0	50.9	51.0	51.9
Left-hand Furnace Wall -	Circuit No. 2 Tubes No. 3 and 4				
Flow Thru Tube No. 3 - %	50.5	50.2	51.7	51.0	51.0
" " " 4 - %	49.5	49.8	48.3	49.0	49.0

about 200 F in the calorimeter. This gives a check on the presence of entrained steam and none could be detected by this method.

Other tests were conducted to check the water temperature at several points in the suction pipes between the drum and the pumps, by the use of iron-constantan thermocouples peened into the metal of the pipes under the insulation. A precision potentiometer, using an ice cold junction, was used and the data of Table 4 are typical of the results obtained with two-pump operation. The difference between saturation temperature in the drum and the measured temperatures was 6 to 9 F which is about one half the calculated difference assuming no condensation of steam to heat the feedwater in the drum. Therefore these tests indicate that the rate of steam condensation in the drum was 10 to 15 per cent of the rate of steam flow leaving the unit and this correction, if applied to the Fig. 8 data, would bring the results of the chemical method in very close agreement with the total flow determined by Pitot tube and orifice pressure drop.

Furnace-Tube Temperatures. The measurement of the fire-side skin temperature of furnace-wall tubes is of considerable interest to boiler manufacturers and users. Thermocouples if carefully installed will give reliable readings over a period of several months and some have been in service more than 2 years. The skin temperature can be used to give an approximate evaluation of the rate of heat absorption at different levels in the furnace and to study the rate of slag or ash accumulation, as well as the effect of slag removal by blowers or by lowering the rate of steam generation. Several papers discussing the effect of slag and ash on heat absorption are to be found in a recent publication (9), and some of the results of the studies at Montaup are included in these papers.[5] Another reference (10) discusses the effect of metal thick-

[5] Reference (9), Fig. 9, p. 13; (see Fig. 14 of this paper).

TABLE 4 CIRCULATING-WATER TEMPERATURE MEASUREMENTS

Date, 1943	July 6	July 13	July 14	July 15
Boiler output, 1000 lb/hr	550	530	550	550
Boiler pressure, psig	1875	1870	1875	1875
Feed temperature to drum F	510	520	500	525
Saturated Steam Temp. F	628	628	628	628
Temp. N. Pipe below upper elbow, F	—	621	619	621
Temp. at suction header, F	622	622	619	—
Temp. at "A" pump inlet, F	621	622	619	621
Temp. at "B" pump " , F	621	—	—	—
Avg. (to nearest deg.) F	621	622	619	621
Difference from saturation F	−7	−6	−9	−7

ness, heat absorption, and internal-surface factors on the tube-metal temperature.

The data obtained at Montaup and already published show that even in a bare-tube furnace and in the zone of highest flame temperature with pulverized-fuel firing, the hot face metal temperature with normal slag covering was only about 50 deg F above saturation temperature. The tubes are $1^1/_4$ in. OD 0.165 in. minimum wall thickness, but the average wall thickness is probably more nearly 0.180. It may therefore be readily calculated from the charts and equations of reference (10) that the heat absorption at the point of temperature measurement was only about 50,000 Btu per hr per sq ft, allowing for reasonable drop at the evaporating film. These data also show the effect of deliberately removing the slag and ash coating from a portion of the wall, thus exposing a small surface to the full radiation of the flame at a temperature approaching 3000 F. Under these conditions the skin temperature rose to a maximum value 172 deg F above saturation (786 F — 614 F saturation temperature), and the corresponding heat-absorption rate would be about 150,000 Btu per hr per sq ft, allowing for temperature gradient between inside surface and the evaporating water. These values represent the extreme rate of heat absorption to which these bare tube walls could be subjected because if a larger area in the zone of high temperature were clean the radiating temperature and the average heat-absorption rate would both be lower.

In June and July, 1943, a study was conducted to learn the effect of circulation rate on the polished interior surface of several tube specimens installed in several furnace-wall circuits. In each of three circuits having different-size orifices a pressure tap was available on the downstream side of the orifice so that the water-flow rate in each could be determined by measuring the orifice pressure drop. The results of the flow measurements appear in Table 5, section A.

There were three thermocouples at the 30 ft elevation on side-wall tubes Nos. 3 and 5, and at the 24 ft elevation in the rear wall there were three couples on tube No. 87 and two on tube No. 81. When tube-temperature measurements on the side-wall tubes are compared to those on the rear-wall tubes, it should be considered that the side-wall tubes are very close to a corner of the furnace. The couples on tubes Nos. 81 and 87 were quite close to a furnace wall blower, and the temperature readings reflected the effect of blower operation. At the end of 2 weeks continuous operation, with coal-firing, and with the boiler operating at maximum guaranteed output, tube-temperature data were collected; some of the readings are included in Table 5, section B.

All these thermocouples were read daily at less than hourly intervals for a month. It was noted that tube No. 81 was more sensitive to wall blowing than No. 87 but never did the temperature of either tube exceed 720 F even immediately after operating the blowers. On some occasions after soot-blowing, the temperature spread for the five couples was less than 10 deg F. From these observations it was concluded that variation in circulation from 57 to 230 per cent of normal had no effect on the tube temperatures.

TABLE 5 EFFECT OF VARIABLE ORIFICE SIZE ON CIRCULATION AND TUBE TEMPERATURE AS SHOWN BY (A) FLOW DATA AND (B) TEMPERATURE DATA

A - Flow Data - June 14 & 15, 1943 - 1845 psi - 465,000 lb/hr output				
Wall Location	Left	Left	Rear	Rear
Circuit Tube No.	3 & 4	5 & 6	87 & 88	81 & 82
Orifice diameter, in.	0.50	0.34	0.34	0.25
Water Flow, lb/hr	12,000	*5,150	6,400	3,625
Inlet Tube Velocity, ft/sec.	9.1	3.9	4.82	2.73
B - Temperature Data - June 22, 1943 - 1880 psi - 650,000 lb/hr output				
Tube No.	3	5	87	81
Elevation above floor, ft	30	30	24	24
No. of couples	3	3	3	2
Max. Temp., F	663	657	690	702
Min. Temp., F	639	644	652	644

* There was no downstream orifice pressure tap on this circuit but the water-flow figure given was obtained on the same size orifice in the opposite side wall.

In October, 1944, the boiler was placed in operation with oil-firing after an outage. Under these conditions the furnace walls were clean and remained clean while tube temperatures were taken. Oil was fired tangentially through burners interposed between the coal burners, and it would therefore be expected that tube temperatures in the lower part of the furnace would be much higher than with coal-firing. Fig. 9 illustrates the results of these tests and shows the variation in tube temperature at different elevations above the furnace floor when the boiler output was varied from 260,000 to 560,000 lb per hr. At each elevation noted two thermocouples were installed 1 in. apart, one as a check against the other, because previous experience had indicated that a couple sometimes becomes shorted against the cover plate, thus giving a high and erroneous reading. At the time these studies were made a panel of eight tubes in the right-hand side wall was being used for another special study of the effect of the orifice size on internal tube conditions, but tubes No. 133 and 139 were the only two tubes of the group which had couples installed at the 5-ft and 11-ft elevations. Tube No. 133 is one of a pair served by a normal-sized 0.34 in. orifice in the trunk of the bifurcate; tube No. 139 was one of a pair having no orifice below the bifurcate but having an individual orifice of 0.25 in. in each tube above the bifurcate. The calculated water-flow rate to each of these tubes is the same. The location of this group of tubes is one where maximum flame temperature would most likely occur but it will be noted that the maximum tube temperature measured was 720 F.

The object of the special study started in 1944, and just mentioned as involving a group of eight tubes in the right side wall, was to confirm over a 1-year operating period, with routine boiler-water control, the general conclusions resulting from the shorter study of tube temperature and interior-surface condition of the rear-wall and left side-wall tubes. The four bifurcated elements chosen were side by side in a zone of high heat absorption and actually straddled a wall-blower element located about 34 ft above the furnace floor. Test specimens of new tubing were installed in all eight tubes, extending from approximately 18 to 25 ft above the floor, and two thermocouples were installed on each tube at each of the following elevations: 18 ft, 24 ft, and 26 ft. One bifurcated circuit had a normal orifice, one an oversize, one an undersize, and one element had individual orifices in each leg which were installed at two butt-welded tube joints in the form of a diaphragm-backing-ring with the orifice hole drilled through the center of the diaphragm. During the first month the boiler was coal-fired at design pressure and daytime load generally be-

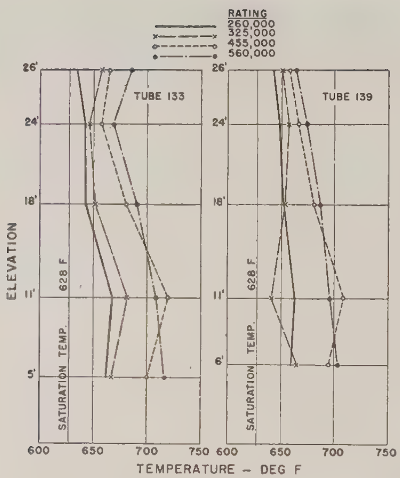


FIG. 9 PLOT SHOWING EFFECT OF RATING AND ELEVATION IN FURNACE ON TUBE TEMPERATURES WITH OIL-FIRING

tween 550,000 and 610,000 lb per hr, followed by 4 days' operation with 100 per cent oil at 550,000 lb per output. Table 6 presents the range of readings at each elevation on each of the eight tubes during these two periods of operation.

It will be noted that tube No. 139 had abnormally high temperatures at the 18-ft elevation, although they were normal at the two higher elevations. The boiler was taken off the line July 16 for a scheduled outage, and a blister was found on tube No. 139 at about 10 ft above the floor. A short piece was welded in and the boiler placed in operation the following day, but temperatures were still somewhat higher than normal at this one elevation. Subsequently, a probe was run up into these two individual orifices to make sure they were clear and after that the boiler was operated for over a year with normal temperatures. In October, 1945, these individual orifices were removed and it was discovered that the probable cause of the blister was welding scale which had formed on and near the orifice during welding and which had subsequently flaked off and restricted the flow. In further discussion of the temperatures in Table 6, the figures given for tubes Nos. 139 and 140 will not be considered.

Table 6, section A, shows a wide variation in temperature due to the effect of ash building up and falling or being blown off the tubes. At the lower elevations the metal was practically at saturation temperature when covered with ash and never exceeded 705 F under normal operating conditions. At the highest elevation which was still about 8 ft below the nearest wall blower the minimum and maximum temperatures were higher due to more effective cleaning and lower rate of accumulation after cleaning. Table 6, section B, shows that with oil-firing the spread in temperature at all couples was much less than with coal, due to more uniform ash conditions at the absorbing surface. This part of the table also shows that with oil-firing the tubes are hotter at the lower elevations as would be expected with a cleaner furnace. In no case is there any significant difference in temperature that can be attributed to orifice size.

Steam Purity. Because of the several changes and modifications which were necessary before satisfactory steam purity was obtained, considerable data were collected on this subject. During the early part of this work samples were taken from the "dry" drum to determine the effectiveness of the separating apparatus in the "wet" drum and from the inlet header of the superheater to determine the final result. However, it was found that

TABLE 6 FURNACE-TUBE TEMPERATURES AT DIFFERENT ELEVATIONS ON TUBES HAVING DIFFERENT SIZE ORIFICES WITH (A) COAL-FIRING AND (B) OIL-FIRING

A – Coal Firing – 4 Weeks at 550,000 to 610,000 lb/hr Load

June 12 – July 12, 1944

Orifice Size		.34		.30		.50		.25	.25
Circuit Tube No.		133	134	135	136	137	138	139	140
Elev. 18'	Max.	705	705	665	680	675	690	790	700
	Min.	630	630	630	630	630	630	630	630
Elev. 24'	Max.	705	695	685	695	680	680	690	685
	Min.	630	630	630	635	635	635	635	645
Elev. 26'	Max.	725	---	710	720	---	---	695	720
	Min.	645	---	650	650	---	---	645	655

B – Oil Firing – 4 Days at 550,000 lb/hr Load

July 12 – July 16, 1944

Tube No.		133	134	135	136	137	138	139	140
Elev. 18'	Max.	690	715	700	710	720	710	785	695
	Min.	680	670	665	665	670	670	710	655
Elev. 24'	Max.	690	685	680	680	675	695	690	690
	Min.	655	660	660	650	640	655	650	650
Elev. 26'	Max.	690	---	670	685	---	---	680	690
	Min.	665	---	655	660	---	---	655	660

under certain conditions the steam conductivity was lower (indicating better purity) entering the final drying screens than leaving the screens, as determined by the superheater inlet-header sample. Switching the conductivity cells at the two sampling points did not change the results. Some time later it was discovered that if steam conductivity data were collected during a period when saturated steam was being taken from the dry drum for operation of the soot blowers, the conductivity of the superheater inlet-header sample fluctuated rapidly, and the average was much higher than when the soot blowers were not being used. This led to the installation of a sampling nozzle in the superheated-steam line near the high-pressure-turbine inlet, and it was found that this sample had a lower conductivity and did not fluctuate or increase very much during soot-blowing periods.

When the final changes were made to the dry-drum internals, a long sampling nozzle was installed at the outlet of the final drying screens between the screens and the several off-take tubes leading to the superheater. Furthermore, the off-take nozzle for supplying soot-blower steam was extended through the screens, as mentioned elsewhere in this paper. A sampling connection was made in the side of a horizontal run of this small soot-blower steam pipe (dictated by convenience), and subsequent checks on steam purity entering the final drying screens were made only during periods of soot blowing.

It was found that the source of make-up water and the season of the year had a very marked effect on steam conductivity leaving the high-pressure boiler. For example, there was a period during the spring of 1944, when it became necessary to use one of the low-pressure boilers as an evaporator for the make-up water. Raw water was evaporated in the boiler and condensed in the low-pressure units and it was found that the conductivity almost immediately increased about 1 mmho all through the station at points where it was being measured, such as at condenser hot wells, feed line and high-pressure-boiler steam outlet. This increase was found to be due to an increase in ammonia. There is also a seasonal variation in ammonia caused by the make-up water from the evaporators, even though the make-up is only about 2 per cent. For example, the uncorrected conductivity of steam in August, 1945, was about 3.5 times greater than the corrected value but less than 1.5 times greater in the month of May, 1945. The corrected value is currently obtained by passing the condensed sample of steam through a degassifier; earlier values of the gas correction were obtained either by checking for ammonia and carbon dioxide in the sample or by comparing the uncorrected conductivity with the results of gravimetric measurement of the total solids.

Fig. 10 shows the relation between total solids, determined by evaporation, and the conductivity of undegassed samples of condensed steam. The generally accepted conversion factor by which conductivity of a degassed-water sample is multiplied to obtain ppm of dissolved solids, is 0.60. Fig. 10 shows that when the evaporators were used for make-up the factor was about 0.40 in the cold-weather months, and 0.20 or less in the warm-weather months; but when the low-pressure boilers were used to evaporate the make-up, the factor was only about 0.20, even in the cold-weather months.

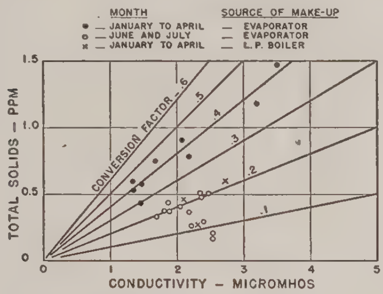


FIG. 10 RELATION BETWEEN CONDUCTIVITY AND TOTAL SOLIDS DETERMINED BY EVAPORATION

All the condensed-steam samples which were evaporated for total-solids determination after the final changes were made in the drum internals were obtained at boiler loads between 550,000 and 650,000 lb per hr. For each of these determinations a 3-liter sample was evaporated to dryness in the special steam-sample evaporator which is described in another paper of this series (11), and the results are given in Table 7.

The conductivity of undegassed-steam samples taken from the dry drum ahead of the final drying screens was found to be 4.23 mmho and 4.56 mmho at boiler loads of 620,000 and 650,000 lb per hr, respectively. Other samples, evaporated during the same week, indicated that the conversion factor was 0.20 or less, so the corresponding total solids in the dry drum would be 0.85 and 0.91 ppm, respectively. This would indicate that without the dry drum the purity would be equal to or better than the minimum guarantee of 1 ppm which has now been recommended by the American Boiler Manufacturers Association and Affiliated Industries (A.B.M.A. and A.I.). The guarantee on this unit was 0.50 ppm, and the screens in the dry drum removed enough moisture to bring the final results below the guarantee as will be seen from the results given in Table 7.

TABLE 7 STEAM-PURITY DETERMINATIONS BY EVAPORATION

June 13 to July 20, 1944

Boiler Output lb/hr	Number of Determinations	Boiler Water Avg.* T.S. ppm	Steam Sample Avg.* T.S. ppm
550,000	3	593	0.34
572,000	3	1107	0.34
610,000	7	598	0.37
650,000	1	1137	0.37

* T.S. is total solids determined by evaporation.

Effect of Circulation Rate on Internal Surface of Tubes. In June, 1943, the boiler was placed in operation with one oversize and one undersize orifice. The location of the circuits, orifice size, and relative circulation rates are given in Table 5, section A. Table 5, section B, lists the number and location of thermocouples which were installed in 2-ft-long sections of new tubing having polished inside surfaces. These sections were installed to determine whether corrosion would occur on polished surfaces with the new potassium treatment, including the feeding of silica, and whether rate of circulation had any effect. A polished section was also installed in the adjacent rear-wall circuit near the discharge end, in the zone above the furnace roof where the tube would not be exposed to heat.

At the end of 1 month of operation at 470,000 lb per hr average load and 630,000 lb per hr maximum load, one half of the polished sections were removed from the two rear-wall tubes. A superficial examination revealed no difference between the two tubes; each was lightly and uniformly coated on the inside with a film of gray sludge, and under the coating of each the metal had been slightly attacked on the fire side but less so on the side toward the casing.

At the end of another month of operation at an average load of 510,000 lb per hr, the remaining half of the original polished sections were removed from the rear-wall tubes. Again a superficial examination revealed no difference between the two sections but there were a few small pits on each. The polished sections from the side wall were not removed for examination at this time since the study of the rear-wall tubes had shown no difference in appearance even with relative circulation rates of nearly 2 to 1. The results of a more elaborate examination of the foregoing and other polished sections removed at later dates are discussed in another paper of this series (12).

In June, 1944, when it had become apparent that the omission of silica from the water treatment had resulted in improved conditions, another special study was commenced involving four circuits in the right side wall as previously mentioned and described (see Table 6). After a year of operation tube sections were removed from these circuits in September, 1945. Again, superficial examination of the tubes indicated that the rate of circulation, within the range of 200 to 75 per cent of normal, had little effect on the appearance of the surface. Regardless of rate of circulation, all the tubes had a light coating of black sludge of less thickness than had been found the year previous; the side toward the fire was much cleaner than the side toward the casing; metal under the film was very smooth on both fire side and casing side except for numerous small pinpoint pits which occurred more frequently on the unheated side than on the heated side.

Table 8 shows the marked difference in weight of deposit for the fire side compared to the casing side; in each case the tubes were split along the vertical axis separating the fire side and the casing side, and all the sludge deposit from a 5-in. length was removed and weighed. It will also be noted that the tube which had the highest rate of circulation also had the largest deposit of sludge.

TABLE 8 COMPARISON OF WEIGHT OF SLUDGE SCRAPINGS FROM TUBES WITH UNDERSIZE, OVERSIZE, AND NORMAL ORIFICES

Orifice Designation	Oversize	Normal	Undersize
Orifice Diameter - in.	0.50	0.34	0.30
Tube Number	138	134	136
	Weight of Sludge - Milligrams		
Fire Side	162.1	96.0	97.5
Casing Side	244.1	179.0	146.7
Total	406.2	275.0	244.2

The results of a careful examination of the surfaces of other specimens from the same tubes are discussed in another paper of this series (12).

PERFORMANCE

Circulating Pumps. The boiler is normally operated with two

pumps running and the third pump idle with suction and discharge valves closed. Normally, at least one dual-drive pump is always in service and after 3 years of operation a power failure did occur and trip out both pump motors; the turbine drive on one pump took over the load and maintained full speed on one pump. Test data plotted in Figs. 5 and 6 show that two pumps handle about 3,200,000 lb per hr and since the operating water content of the boiler is about 30,000 lb (exclusive of economizer), this represents a recirculation rate of more than 100 times per hr. The ratio of water to steam for any rate of output can also be readily determined from either Fig. 5 or Fig. 6. It will be noted that at maximum load of 650,000 lb per hr, this ratio is about 5 with two-pump operation, and the same ratio would be maintained with one pump up to a load of about 450,000 lb per hr. At 650,000 lb per hr load with one pump the water-to-steam ratio would be nearly 3 to 1, a condition which should not cause any trouble during an emergency period since it has been proved in experiments with circuits that operation at one half normal circulation over a period of 2 months produced no ill effect. It has, however, been the practice not to exceed 400,000 lb per hr output with one pump, particularly if the other two pumps were not in condition to operate. There were short periods early in 1943 when only one pump was in operating condition, but Table 1 shows that no outage of the boiler was ever caused by the circulating pumps.

The circulating-pump specifications rate each pump at 3500 gpm 50 psi head (184 ft) when handling water at 630 F and operating at 1750 rpm, but shop tests by the manufacturer indicated the total head to be 192 ft at the capacity mentioned. Owing to conservative pressure-drop calculations for the circulating system, the actual resistance for a given flow rate is lower than anticipated, and consequently, the pumps operate at higher than design capacity. Fig. 7 would indicate that when two pumps are operating, each is handling about 4600 gpm rather than 3500 gpm and that with single-pump operation the capacity is nearly 6000 gpm.

With two-pump operation each pump uses an average of 37 amp at 2300 v which is equivalent to about 3,500,000 Btu per hr in fuel fired, or a gross input to the two motors equal to less than 0.5 per cent of the fuel fired at maximum load. The hydraulic horsepower under the same conditions would be 240 for two pumps, which represents input of energy to the water at the rate of about 600,000 Btu per hr, so the net consumption of the pump drivers would be equivalent to 0.35 per cent of the heat in the fuel fired at maximum output.

Heat Absorption and Distribution. No attempt was made to check the actual distribution of heat to the various furnace walls but it can be calculated from average heat-absorption rate and exposed surface with due allowance for surface effectiveness. Table 2 shows that measured distribution of water to the various furnace walls is almost identical to calculated distribution of heat.

Distribution of heat to various other parts of the unit such as superheater, reheater, economizer, and air heater, is readily checked by measuring temperature rise or drop of one or both fluids and calculating or measuring the quantity of the fluids. Total heat absorption of each part of the unit was very close to manufacturer's predictions, except in the case of the reheater which was too conservatively designed. The method of correcting the reheated-steam temperature has been discussed elsewhere in this paper.

Figs. 11 and 12 show the results of informal tests in October and November, 1943, with both coal and oil firing, compared to predicted performance for coal-firing represented by the curves. The steam-temperature controller was set to maintain 950 to 960 F, according to the Micromax recorder, and the reheat temperature was also being regulated according to recorder, but at the end of the tests the recorder was checked and found to be reading 15 deg F too high. The corrected temperatures are plotted and therefore fall below the predicted values. However, for the tests above 450,000 lb per hr output higher steam temperature could readily have been obtained, because during these tests the by-pass dampers were being used to reduce the temperature of high-pressure steam, and the reheated-steam temperature was being reduced by desuperheating.

In comparing the results of the oil-firing and coal-firing tests it should be noted that oil was used only for the duration of the tests and coal was used during about 16 hours of each day. Therefore the furnace walls and other heating surfaces were not as clean as would be the case with prolonged used of oil.

Although the average gas temperature leaving the two heaters was close to contract predictions, the air temperature during these tests was higher than predicted. The three factors responsible for this were: (*a*) lower ratio of air to gas (*b*) higher gas-inlet temperature, (*c*) higher air-inlet temperature; but because of conservative selection of air-heater surface, the air-heater heat absorption was close to predictions. Air leaks into the economizer were responsible for the low ratio of air to gas, and these have been stopped, thus reducing both air-outlet and gas-outlet temperatures.

Superheat Control. Steam temperature from the high-pressure superheater is automatically controlled and the equipment used is described in another paper of this series (13). Automatic regulation of both sets of dampers at the economizer outlet results in very close regulation. Starting with the conditions which exist immediately after operating the furnace wall blowers, the by-pass damper is fully closed or slightly open depending upon the rate of output. As the furnace accumulates ash and the gas temperature increases, the by-pass dampers automatically open and maintain the steam temperature within ±5 F of the desired limit. At the end of an 8-hr operating period the by-pass dampers will be 50 to 80 per cent open, depending upon the output rate and the nature of the coal. Then, as the wall blowers are operated, the by-pass dampers operate automatically to compensate for the effect of removing the ash from the walls and maintain the same close regulation of steam temperature during the cleaning period. Tests were conducted to determine the full effect of operating the by-pass dampers from closed to 75 per cent open. As mentioned elsewhere the lower economizer dampers close as the by-pass dampers open, but a stop on the regulator cylinder prevents the former from closing fully, because full closing has never been necessary to obtain the desired control. Fig. 13 shows the results of these tests and illustrates how the heat absorption of the combined economizers increases as the by-pass dampers open, thus compensating for the decreased superheater absorption and resulting in very little increase of final gas temperature. Oil-firing was used in order to maintain uniform conditions of furnace cleanliness during the tests.

Capacity and Flexibility. The maximum peak capacity of the boiler has never been determined. The maximum demand has been 670,000 lb per hr but the feeders, burners, fans, and circulating pumps are adequate for higher output.

At reduced pressure there is no decrease in circulation, and the volumetric percentage of steam in the mixture leaving the generating tubes will increase so the average velocity in the tubes is greater. The unit is therefore very flexible as regards operation at variable pressure, and the upper limit of capacity would be governed by considerations of steam purity and steam temperature rather than circulation. These upper limits were not determined. During special tests on the high-pressure turbine, wide swings in pressure, output, and steam temperature were accomplished very rapidly and with an excellent degree of control.

The boiler has a very stable water level even under extreme op-

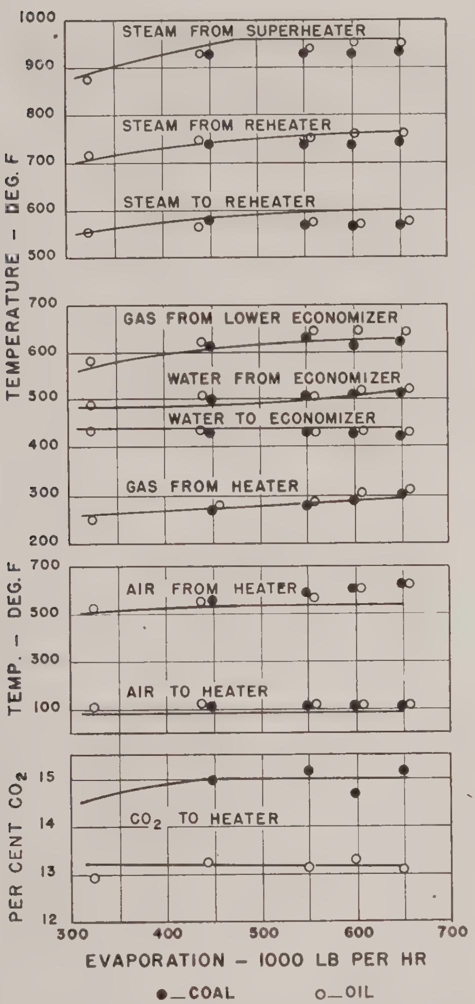


Fig. 11 Results of Tests With Coal- and Oil-Firing Compared to Predicted Performance

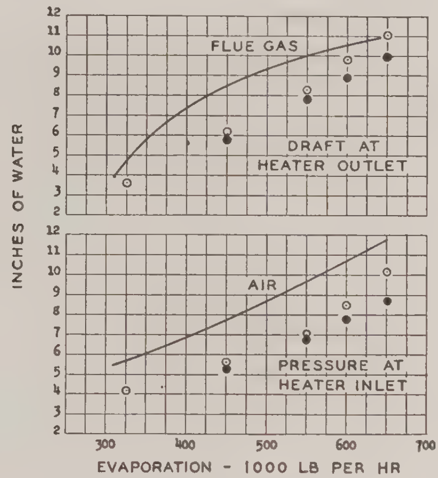


Fig. 12 Results of Tests With Coal- and Oil-Firing Compared to Predicted Performance

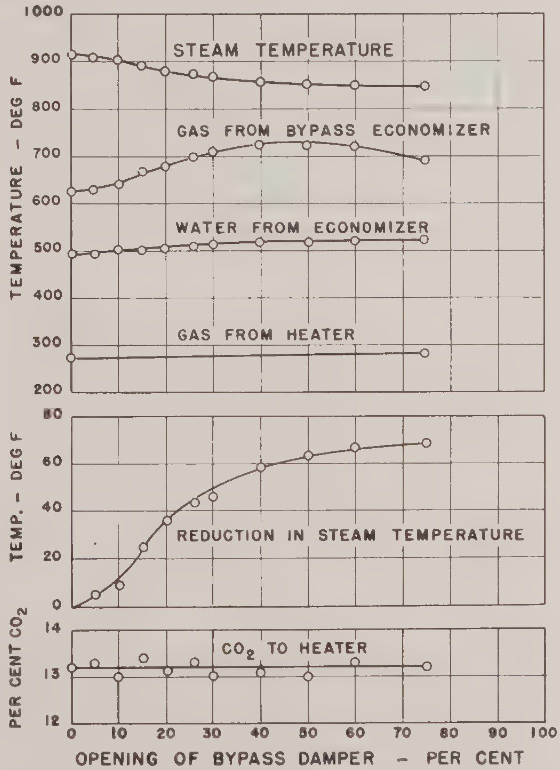


Fig. 13 Results of Superheat-Control Test, Oil-Firing, 560,000 Lb Per Hr Output, 1900 Psi Drum Pressure

erating conditions of sudden load increase or decrease. In fact, sudden emergency load changes have occurred without any perceptible change in water level. Also, on the occasion of furnace-tube failures, it has not been difficult to maintain water level while reducing load and pressure in an orderly manner.

Efficiency. Coal and ash samples were not collected during the informal tests conducted in 1943, so complete heat-balance figures are not available.

The contract over-all efficiency with coal-firing at 650,000 lb per hr output was 89.3 per cent, based on coal of 14,200 Btu per lb, with 2.8 per cent moisture as fired; and based on 15 per cent CO_2 in gases at the economizer outlet, with 290 F gas temperature leaving the air heater, 80 F entering-air temperature, and with 0.6 per cent allowance for unburned combustible.

During the reported tests at 650,000 lb per hr output, the CO_2 at the economizer outlet averaged 15.2 per cent, the air entering the heaters was 110 F, and gas leaving the heaters 306 F. Tests to determine heat loss due to combustible in fly ash were conducted at a later date when the maximum load demand on the boiler averaged 572,000 lb per hr, the CO_2 entering the convection superheater section was 16 per cent, and the average heat loss in the fly-ash combustible for two tests was 0.47 per cent. The results of two other tests with 15 per cent CO_2, at the same output, averaged 0.34 per cent heat loss in the fly-ash combustible.

Conclusion

The authors have revealed all the experiences with this high-

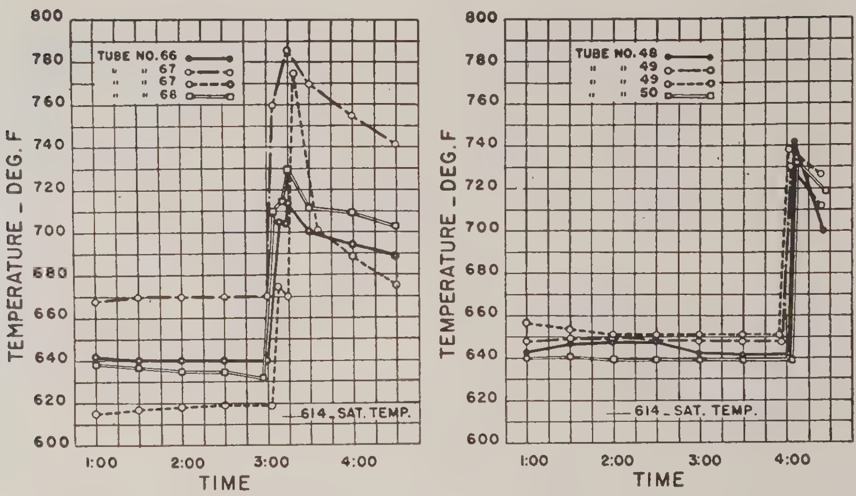


Fig. 14 Effect of Furnace Wall Cleaning on Hot Face Tube Temperature

pressure boiler in considerable detail so that others may benefit from a knowledge of what has been going on at this station during the last 3 years. The special test and research activities reported in this and its companion papers (11, 12, 13) represent only a part of the total number of special studies conducted. Many of these were associated more with operation problems of the station as a whole than with problems peculiar to the forced-circulation boiler. The judgment of those responsible for selecting a bare-tube, slagging-bottom, radiant, forced-circulation boiler for this high-pressure application has been verified by the operating record of this boiler.

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12 "Results of Potassium Treatment," by W. W. Cerna and A. K. Scott, published on pages 443–451 of this issue of Transactions.

13 "Experience With Instruments and Control Equipment for 2000-Psi Boiler at Somerset Station of Montaup Electric Company," by W. D. Bissell and E. B. Powell, published on pages 453–466 of this issue of Transactions.

Discussion

Lewis J. Dawson.[6] It may be in order at this time to restate the suction conditions under which the circulating pumps operate in order that the importance of the labyrinth design can be understood. The water handled by the impeller is at full boiler pressure of about 1900 psi and at the corresponding saturation temperature of 630 F. The pressure, temperature, and incipient flashing preclude the possibility of using a conventional stuffing box in which the packing is subjected to suction conditions. The injection and bleed-off arrangement is required to give safe stuffing-box conditions and normal packing life. The labyrinth must be of a reasonable length and of close clearance to reduce injection water requirements. These two factors make it important that labyrinth alignment be maintained.

After it became apparent that misalignment was causing the labyrinth failures, the support under the coupling end of the bearing body was removed which allowed the pump to move freely with the piping so that no internal misalignment could occur. Measurements were taken of the movement of the free end under various operating and stand-by conditions. These measurements revealed that pipe movement was the major factor, with uneven bracket temperature of secondary importance. Variations in these measurements, both vertical and horizontal, could be obtained by warming and starting one of the remaining two pumps, showing that temperature changes in the common piping affected alignment of the unit being measured. Equalization of bearing-bracket temperature did not restore alignment.

The units are equipped with spacer-type gear couplings, so that the resultant 0.030 to 0.040 in. misalignment between pump and driver can be tolerated. In connection with the couplings, the author has mentioned the limit stops which were installed to prevent motor bearing failure. It is a sad commentary on motor design that all large motors must be protected in this manner because of inadequate thrust-bearing capacity.

Failures of oil-pump spiral drive gears were found due to wear in the oil-pump shaft bearings. This resulted in improper alignment and tooth contact between gear members. The new-design bearing housing has incorporated a ball-bearing oil-pump shaft support.

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[6] Field Engineer, Cameron Engineering Service, Ingersoll-Rand Company, Phillipsburg, N. J. Jun. A.S.M.E.

With reference to Fig. 7 of the paper, it is probable that the actual operating speed is close to 1790 rpm under the reduced horsepower loading. The majority of the steaming test points fall reasonably close to the revised head-capacity curve. We note that the field points having greatest deviation are all from high-temperature water readings. Checking the point of greatest deviation, calculating the water horsepower and comparing it to brake horsepower input, we arrive at a pump efficiency 21 per cent greater than the manufacturer's shop test curve value at the corresponding capacity. This would lead us to believe that the Pitot tube calibration is off at high water temperatures and corresponding low densities.

Authors' Closure

The same data which Mr. Dawson used in arriving at his statement that pipe movement was the major cause of pump misalignment can be interpreted differently when all the operating procedures occurring at the time of measurement are also taken into consideration.

Fig. 4 shows the position of the suction pipes to each pump, and the suction header which is common to all three pumps. The horizontal movement of these three pipes was measured at an elevation 20 ft above the pumps during the process of raising pressure on the boiler and it was found that there was no movement of the center pipe while the movement of each outside pipe was about $^1/_4$ in., but in opposite directions, as would be expected due to expansion of the suction header. This being the case, one would expect the movement of the coupling end of the bearing to be in opposite directions on *A* and *C* pumps, and no movement on *B* pump, if due to pipe strain. Instead this end of the bearing body moved downward on all three pumps, from a cold to hot condition, and the cause of the pump misalignment therefore is a controversial subject. The fact remains that the trouble was corrected by merely allowing the coupling end of the bearing to float freely and by taking steps to equalize upper and lower bearing bracket temperatures.

The authors desire to comment further on the tube failures which occurred on October 26, 1945. The amount of sludge found in the drum after this failure was greater than had previously accumulated during any operating period of equal duration, and this sludge was nearly 70 per cent copper. Sludge deposits from the generating tubes were also higher in copper content than previously noted. This would indicate that prior to this failure unusual feedwater contamination in the form of corrosion products, including copper, had been prevalent.

The failures located 20 ft above the floor were diagnosed as having been caused by sludge deposits and subsequent overheating sufficient to cause weakened metal and intergranular rupture. The failure close to the furnace floor may also have been started by sludge accumulation which increased the outside surface temperature a sufficient amount to accelerate external tube corrosion. Concurrently corrosion was causing a thinning of the metal from the inside and laying down a deposit of the corrosion product, iron oxide, to further insulate the inside of the tube. When the tube became thin enough a small bulge occurred which quickly overheated locally until rupture occured.

After acid washing the boiler, thermocouples were installed at four elevations between 8 in. and 24 ft above the floor on the tube which had failed close to the furnace floor. During the past five months these couples have been read twice daily and some of them connected to a recorder for a continuous record. The metal temperature 8 in. above the molten slag on the floor varies between saturation temperature (about 630 F) and 765 F but there has been no gradual increase as would be expected if sludge were again accumulating on the inside surface. The maximum temperature at higher elevations has been 10 to 20 deg F lower. The furnace tubes were checked for external corrosion about January 1, 1946, and there was no evidence of any further attack on any of the walls.

Special Studies of the Feedwater-Steam System of the 2000-Psi Boiler at Somerset Station of Montaup Electric Company

By W. D. BISSELL,[1] B. J. CROSS,[2] and H. E. WHITE[3]

In conjunction with other studies on the high-pressure forced-circulation boiler at the Somerset Station of the Montaup Electric Company during the past 3 years, tests to determine, particularly, the extent of dissolution of iron throughout the system, and the formation of high-iron sludges were made. A summary of the findings of this investigation are given in this paper.

During the course of a complete investigation of the high-pressure forced-circulation boiler at the Somerset Station of the Montaup Electric Company covering the past 3 years, a special study was made on the feedwater and steam system particularly with regard to the dissolution of iron throughout the system, and the formation of high-iron sludges within the boiler. This paper gives a summary of the findings of these studies.

The period covered by this work was from January, 1944, to October, 1945. The boiler was thoroughly cleaned by acid treatment late in December, 1943. The boiler water has been continually on potassium-salt treatment since May, 1943.

Over the past few years the occurrence of iron oxide in boilers has become a matter of interest, if not concern, to operators of high-pressure boilers, as the presence of this substance must always indicate corrosion in some part of the system. Usually the amount of iron oxide is small by comparison with the large area of steel surfaces involved, and the indicated loss of metal, if uniformly distributed, is not large enough to be a cause of great concern. If, however, there are small areas of concentrated attack, the amount of iron oxide found may well indicate serious corrosion leading to failures.

Even in the case of uniform corrosion of iron which in itself may not be a serious matter, the accumulation of the solid products might lead to failures in the boiler by insulation effect on surfaces receiving heat and also by obstructing passages and thereby interfering with circulation of water.

Sludge deposits found in the high-pressure boiler at the Somerset Station were of particular concern because tube failures that had occurred were attributed to these deposits either on the surfaces of the tubes or on the strainers protecting the orifices in the tube circuits.

The examination of samples of tubes cut from furnace walls at periodic examinations disclosed a thin film of deposit on the entire inner surface of the tube. On the evaporative surfaces the deposit was usually baked on in a semiscalelike form. On the inactive surfaces it occurred as a slime which dried out to a dust. Both types of deposits could be removed with a stiff brush. Slime deposits of sludge also were found in the inlet headers, on orifice strainers and on other surfaces not exposed to external heat. Scalelike flakes of caked sludge have been found lodged in the holes in the strainers. Some of these flakes had a curvature indicating that they had formed in headers or downtake piping. Deposits found in the drum were more granular than those found in the tubes and probably represent the heavier components of the sludge.

The composition of the sludge varied somewhat with location but all deposits had a high proportion of black iron oxide. Typical analyses of sludges are given in Table 1.

TABLE 1 TYPICAL ANALYSES OF SLUDGES

SAMPLE NUMBER	1	2	3	4	5	6
IRON AS Fe_2O_3 - PER CENT	53.3	47.0	78.3	27	63	42
PHOSPHATE AS P_2O_5 - PER CENT	18.5	3.3	6.0	20-25	6-8	12-15
SILICA AS SiO_2 - PER CENT	.6	TRACE	.5	.5	1	TRACE
CALCIUM AS CaO - PER CENT	12.3	4.1	6.5	20.0	10	15-20
MAGNESIUM AS MgO - PER CENT	10.3	1.6	5	12-15	4-8	8-10
COPPER AS Cu - PER CENT	-	41.3	4.2	-	7	2

1. SUSPENDED SLUDGE AT BLOW DOWN.
2. DRIFT DEPOSIT IN WET DRUM.
3. DEPOSITS ON STRAINERS.
4. DEPOSIT ON DOWNCOMER SCREENS
5. DEPOSITS IN REAR HEADER COLLECTING CHAMBER.
6. SLUDGE FROM REAR HEADERS.

The high proportion of iron oxide in these sludges naturally led to some concern as, if it were formed entirely within the boiler, it might indicate severe local corrosion or pitting that would result ultimately in failure.

The accumulation of sludge, regardless of its composition, in the boiler might result in stoppages of furnace circuits or overheating of metal by insulation of heat-absorbing surfaces. Corrosion may occur as a result of the concentration of boiler-water salts under deposits of sludge on heating surfaces. It is known that adherent scales may form under such deposits on steam-generating surfaces as a result of the concentration of the components in the sludge "sponge" and the higher metal temperature resulting from the insulation of the metal.

The high-pressure boiler at this station delivers steam at 1850 psig and 950 F to the high-pressure turbine. This turbine exhausts through a reheater to the low-pressure system. The water supplied to the boilers is condensate with make-up supplied by evaporators. The principal outside sources of contamination were condenser leakage and evaporator carry-over.

A simplified diagram of the feedwater system is shown in Fig. 1. The condensate from the hot-well pumps passes successively through the oil cooler, 15th-stage heater, deaerating heater, 10th-stage heater, low-pressure feed pumps, exhaust heater, extraction heater, high-pressure feed pumps, high-pressure heaters, and to the economizer. The drips from the various heaters are re-

[1] Somerset Station, Montaup Electric Company, Fall River, Mass.
[2] Combustion Engineering Company, Inc., New York, N. Y. Mem. A.S.M.E.
[3] Stone & Webster Engineering Corporation, Boston, Mass. Mem. A.S.M.E.

Contributed by the Power, Industrial Instruments and Regulators Divisions, and the Joint Research Committee on Boiler Feedwater Studies, and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

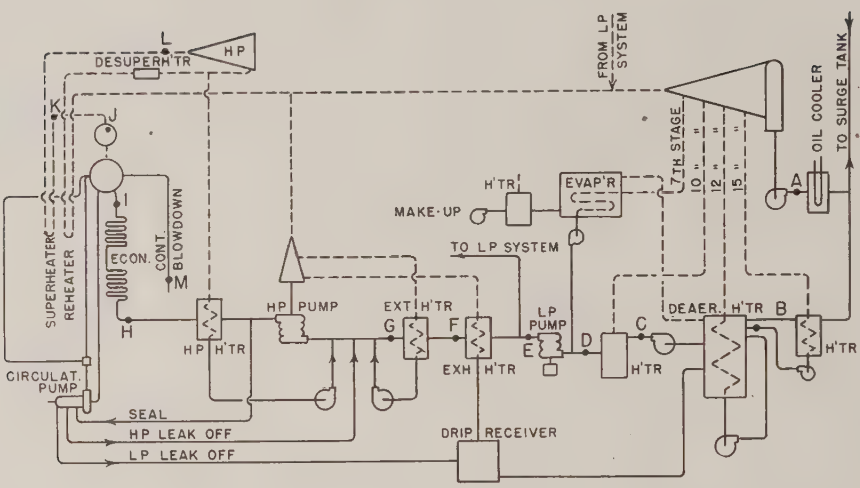

FIG. 1 SIMPLIFIED DIAGRAM OF FEEDWATER SYSTEM

turned to the main stream as indicated. The gland water for the circulating pumps is taken before the high-pressure heaters. The high-pressure leak-off is returned to the feed-pump suction and the low-pressure leak-off to the deaerating heater. Sampling points are indicated in this diagram and are also listed in Table 2. The fluid temperature and pressure and the type of cooling coil used are given in this Table.

TABLE 2 LOCATION AND DATA OF SAMPLING POINTS

	TEMPERATURE °F	PRESSURE LB/SQ IN	COOLING COIL
POINT A - HOT WELL PUMP	80-100	40	NONE
B - 15TH STAGE HEATER	160	40	ST. STEEL
C - DEAERATOR OUTLET	215	80	COPPER
D - 10TH STAGE HEATER	240	80	"
E - LOW PRESSURE PUMP	250	500	"
F - EXHAUST HEATER	290	500	"
G - EXTRACTION HEATER	340	500	"
H - ECONOMIZER INLET	450	2150	"
I - ECONOMIZER OUTLET	520	2150	"
J - STEAM DRY DRUM	630	1950	MONEL
K - STEAM SATURATED HEADER	630	1930	"
L - SUPERHEATED STEAM	950	1850	"
M - BOILER BLOW DOWN	630	1950	"

This diagram represents the condition at the time most of the work reported was done. Minor changes have since been made.

SCOPE OF INVESTIGATION

The feedwater study may be divided into several phases both on the basis of the nature of the work done and on the time periods. Because of wartime restrictions of both manpower and equipment, all of the work could not be carried on simultaneously. Each phase of the investigation is listed in the following tabulation and is discussed in order:

Period 1.

(*a*) The continuous measurement of hydrogen in feedwater and steam and oxygen in feedwater.

(*b*) The measurement of total solids in feedwater, steam and blowdown.

(*c*) The measurement of suspended solids in boiler water at shutdowns of the boiler.

(*d*) The study of gases in steam, particularly as affected by small variation in boiler-water conditions and the normal irregularities in the functioning of auxiliary equipment.

Period 2. The exploration of the feedwater system for dissolved hydrogen, dissolved iron and changes in alkalinity (pH).

Period 3.

Miscellaneous studies as follows:

(*a*) Copper in feedwater.
(*b*) Bead column and ion exchanger column tests.
(*c*) Tube-temperature measurement.
(*d*) Corrosion detector.

The work under Period 1 was done during January, February, and March, 1944.

Period 1 (*a*). Recording hydrogen meters were set up to give a continuous record of the dissolved hydrogen in feedwater and in the steam delivered by the boiler. As hydrogen is one of the products of the reaction of water with iron, the amount of hydrogen evolved may be taken as a measure of the rate of corrosion. As irregularities in the performance of the deaerating heater were suspected, a dissolved-oxygen meter was also installed to sample the feedwater to the boiler. With the exception of short outages for adjustments and maintenance, these instruments have been in continuous operation since January, 1944. Originally they were installed close to the point of sampling but later moved, when stainless-steel tubing became available, to a more convenient location for observation by the boiler operators.

The hydrogen record for the 3-month period, January, February, and March is plotted in Fig. 2. Steam flow and boiler pressures are also shown on this chart. It may be seen that while the hydrogen content of the feedwater is substantially constant, the hydrogen in steam showed relatively high values in the early part of January, with a gradual reduction in these values until they leveled out at $1^1/_2$ to 1 part per billion over the feedwater values. While the high hydrogen values during the first week of the month may be partly accounted for by the lower rating, it is believed that they are in greater part due to residual hydrogen in the system following the acid wash.

The feedwater sample was usually taken from the economizer inlet. It was, however, occasionally taken at the economizer outlet. The values for hydrogen at the economizer outlet fell between those of the inlet and the steam leaving the boiler.

The significant feature of these hydrogen curves is the fact that they show that a large part of the hydrogen in the steam enters the boiler with the feedwater. Also there is an indication from the few readings taken that there is an increase in hydrogen across the economizer.

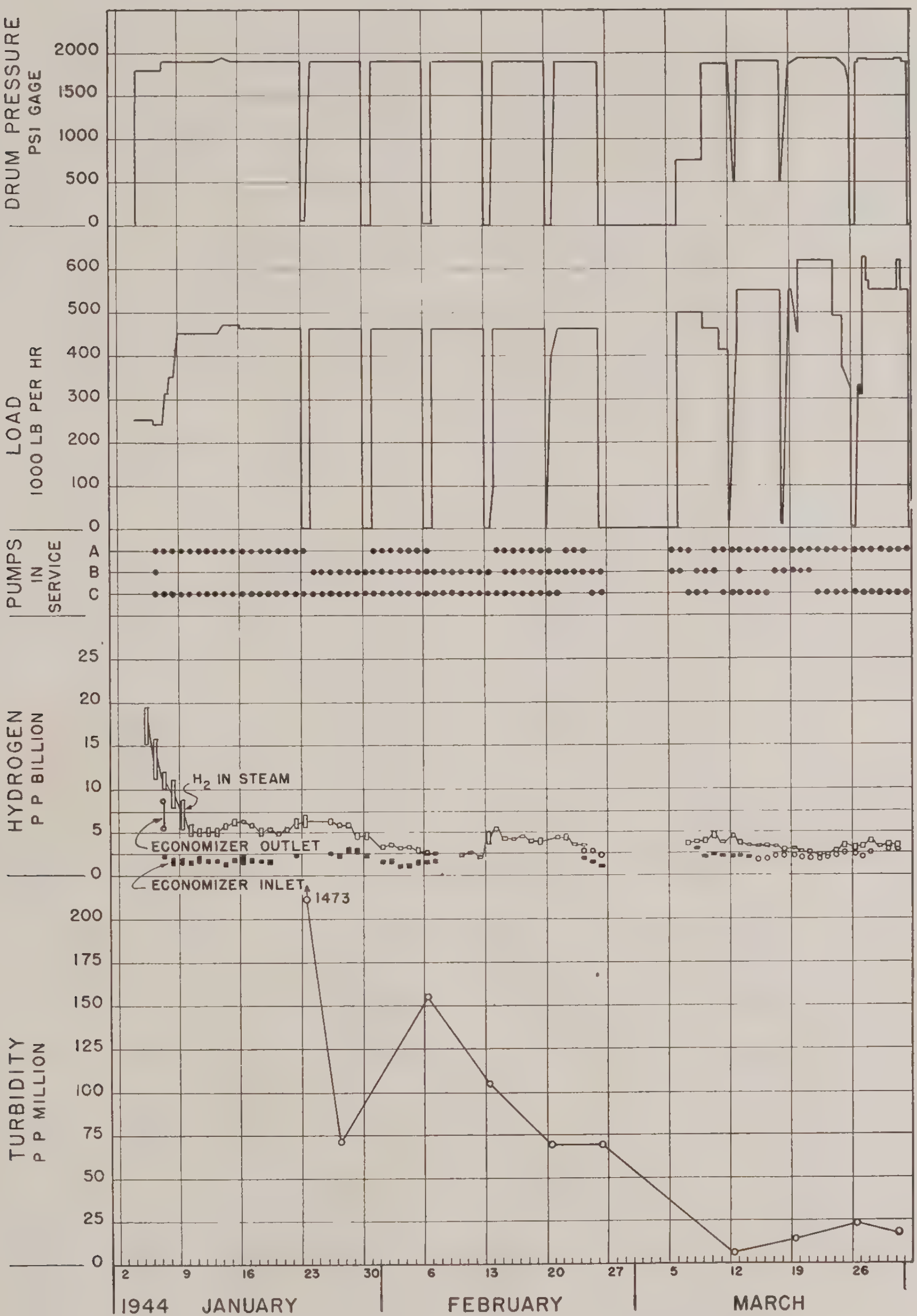

Fig. 2 Hydrogen Content Record of Feedwater for 3-Month Period

A consideration of the probable reaction of iron with water presents the possibility that hydrogen may be evolved in the economizer and boiler without any reaction involving the metal of the boiler. It has been proposed that in the lower-temperature areas the reaction may be

$$Fe + 2H_2O \rightarrow Fe(OH)_2 + H_2$$

At the higher temperature of the boiler and economizer the ferrous hydroxide may react to form magnetic oxide

$$3Fe(OH)_2 \rightarrow Fe_3O_4 + 2H_2O + H_2$$

If, however, all of the hydrogen formed in the boiler came from this second reaction, the relative amounts of H_2 in feedwater and

steam would be 3 to 4. Actually it is nearer to 2 than 3, which indicates some reaction with the boiler metal.

During the 3-month period represented by this chart there were sporadic occurrences of small amounts of dissolved oxygen in the feedwater to the high-pressure unit. Changes in the deaerating system, made in the spring of 1944, resulted in consistently good removal of air and since that time the feedwater has been substantially oxygen-free.

Period 1 (b): Total Solids in Feedwater, Steam, and Blowdown. The first approach to the sludge problem was a study of the amounts of total solids delivered to the boiler in the feedwater and leaving the boiler with the steam produced and the blowdown removed. Samples of feedwater, steam and blowdown were collected in hourly increments over an 8-hr period. These samples were evaporated to dryness and the residues weighed. The standard-size samples were 3 liters for the feedwater, 3 to 4 liters for the steam, and 1/2 liter for the blowdown. All samples were evaporated without boiling in closed containers ventilated with filtered air.

The results of a number of determinations are given in Table 3, with the calculations leading to the total amounts of solids delivered to the boiler and removed from the boiler for a 24-hr period. Feedwater and steam flows were taken from station meters. The boiler blowdown was passed through cooling coils and weighed. Because of the dilution of the feedwater with condensate from the high-pressure and the extraction heater after the point of sampling, it was necessary to correct the total solids in feedwater by analysis for this dilution. Also, because some of the steam produced is diverted through the high-pressure heater and returned to the boiler as feedwater, the measured steam flow must be reduced by that amount.

It will be noted that there is in each case a positive unaccounted-for balance which presumably remains in the boiler.

In a number of tests the iron content of the total-solids residues was determined. The calculations for iron were carried out in the same manner as for the total solids and a figure for iron remaining in the boiler was obtained. These results also are given in Table 1. As with total solids, there is a positive unaccounted-for amount of iron.

Period 1 (c): Suspended Solids in Boiler Water. A measure of the relative amount of sludge formed and remaining in the boiler was afforded by the determination of turbidity of the boiler water that occurs when the boiler is taken out of service. It had long been noted that on shutting down the boiler the water becomes turbid and that the maximum concentration of suspended solids occurred at the pressure interval of 600 to 400 lb pressure; this being at the time the fires were extinguished. Advantage is taken of this period to remove much of the accumulated sludge by blowing down. During January, February, and March, 1944, each week was a test period during which modifications in operating procedure were tried. After each week the boiler was taken out of service and the turbidity measured while the boiler was being blown down.

A graphic log of a typical shutdown is shown in Fig. 3. The various observations and measurements are plotted against time.

The rate of reduction in load and pressure is shown in the lowermost graphs with notation of various operations. The

TABLE 3 RESULTS OF TESTS TO DETERMINE TOTAL SOLIDS IN FEEDWATER, STEAM, AND BLOWDOWN

DATE - 1944	1/26	2/1	2/8	2/16	2/22	3/14
STEAM FLOW (METER) 1000 LB/HR	461	461	464	461	465	554
HP HTR DRIPS (EST) 1000 LB/HR	80	80	80	80	80	90
STEAM TO LP SYSTEM 1000 LB/HR	381	381	384	384	385	464
TOTAL SOLIDS IN STEAM - PPM	.76	.275	.47	.60	.40	.77
IRON IN STEAM - PPM	.026	-	.017	-	.029	.055
F.W. FLOW (STEAM + B.D.) 1000 LB/HR	461,500	461,487	464,475	461,596	465,590	554,594
T.S. IN FEED WATER (EXTR.HTR.) PPM	1.76	1.77	1.26	1.73	2.03	1.93
IRON IN FEED WATER (EXTR.HTR.) PPM	.146	-	.038	-	.06	.065
TOTAL SOLIDS CORRECTED (DILUTION BY H.P. DRIPS)	1.59	1.50	1.12	1.53	1.75	1.74
IRON CORRECTED (DILUTION BY H.P.DRIPS)	.123	-	.034	-	.055	.063
BLOW DOWN - LB/HR	500	487	475	596	590	594
TOTAL SOLIDS IN BLOW DOWN - PPM	454	458	591	668	600	690
IRON IN BLOW DOWN - PPM	.160	-	.05	-	.032	.076
FLOW - 1,000,000 LB/DAY						
FEED WATER AT EXTR. HTR.	11.075	11.076	11.147	11.150	11.175	13.310
STEAM TO L.P. SYSTEM	9.144	9.144	9.216	9.216	9.144	11.147
BLOW DOWN	.0112	.0117	.0114	.0143	.0142	.0143
SOLIDS - LB/DAY - IN FEED WATER	17.609	16.614	12.485	17.059	19.556	22.159
IN STEAM	6.949	2.515	4.331	5.500	3.696	8.583
IN BLOW DOWN	5.448	5.349	6.737	9.552	8.520	9.867
UNACCOUNTED TOTAL SOLIDS	5.212	7.874	1.317	2.007	7.340	4.709
IRON - LB/DAY - IN FEED WATER	1.362		.379		.625	.839
IN STEAM	.238		.157		.268	.613
IN BLOW DOWN	.003		.0006		.0004	.001
UNACCOUNTED - IRON	1.121		.221		.357	.225

boiler had been in operation approximately 1 week using the *B* and *C* circulating pumps.

The dissolved hydrogen in feedwater and steam are plotted only up to the time that reduction in pressure occurs. The readings beyond that time had no significance as the flow of sample was too much reduced. The slight rise in hydrogen as the steam flow is reduced is characteristic of most of the shutdowns and may be due to the increased concentration of hydrogen with reduced flow of steam.

The reduction in concentration of chloride and sulphate is the result of increased rate of blowdown and is a measure of the dilution of boiler water with the increase in feedwater rate.

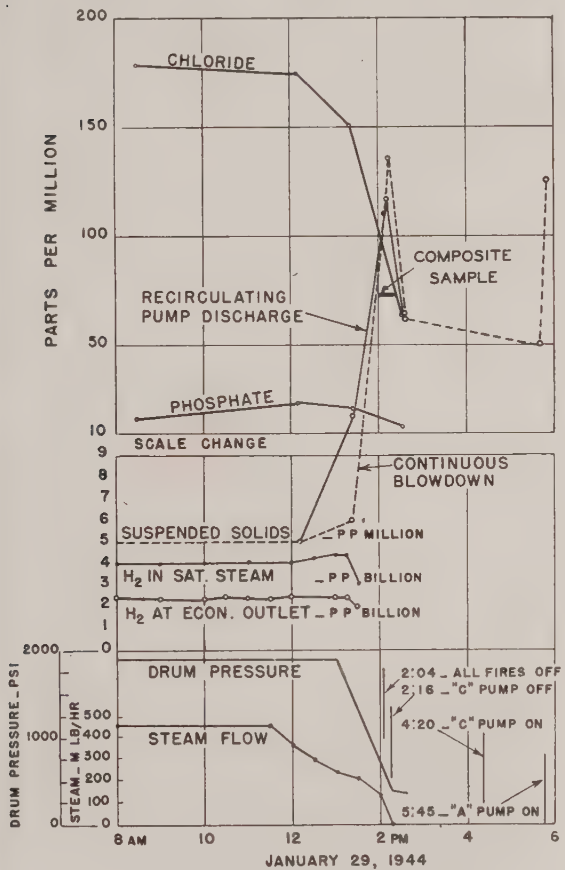

FIG. 3 LOG OF A TYPICAL BOILER SHUTDOWN

The phosphate does not show a corresponding reduction in concentration, indicating a slight "hideout" of that chemical. When corrected for dilution, a recovery of about 1 lb of phosphate as PO_4 is indicated. There is also a slight increase in the silica content of the water.

There are two graphs for boiler-water turbidity, one based on samples taken at the boiler blowdown, the other on samples taken at the discharge of the circulating pumps. While there is good agreement in the two sets of samples, the latter are regarded as the more accurate as the sampling lines were much shorter. The turbidity rose sharply to a maximum value of 135 ppm at the time the fire was extinguished, then decreased to 50 ppm by 5:40 pm. At that time the *A* circulating pump which had been out of service for a week was started. The increase in turbidity from 50 to 125 ppm was caused by the sludge that had settled out on the valves at the pump. As the water content of the boiler under nonsteaming conditions is about 50,000 lb, this increase in turbidity corresponds to $3^1/_2$ lb of sludge.

The composite sample taken over a 20-min period at the time indicated in the graph was assumed to represent the maximum sludge brought into suspension. The peak turbidity was of short duration and the sludge was probably not thoroughly distributed in the mass of the boiler water.

The shutdown procedure, as indicated on this chart, was established for the shutdown of January 29 and was followed as closely as possible on all subsequent tests.

The results of the turbidity measurements for January, February, and March, 1944, have been plotted in the lower graphs of Fig. 2 using, with the exception of January 22, the results of the composite samples. On the January 22 shutdown no composite sample was taken and the point indicated off scale was that of the maximum turbidity sample.

As may be noted, there is a decrease with time in the amount of sludge stirred up at shutdown periods. The high turbidity of the January 22 samples, which represented a full-pressure operating period of about 2 weeks, may be attributed at least in part to the effect of the acid wash. The gradual reduction in sludge over the 3-month period was principally the result of a closer control of condenser leakage and evaporator carry-over. Subsequent to this period of operation, sludge-collecting chambers were installed between the ends of the rear-waterwall inlet header and the side-wall inlet headers and also a sludge blowdown pipe was installed at the bottom of the main boiler drum. These were blown down once a week and the accumulated sludge thus removed during operation.

Period 1 (d): Gases in Steam. The study under this item was in greater part unrelated to the work covered in this report. A portion of the program, however, was a measurement of the gases in feedwater and steam and the results afforded an independent check on the hydrogen, indicated by the recorders. Samples of water and steam were passed through a special degasifier and the gases removed were accumulated until the quantity was sufficient for analysis by a modified Orsat apparatus. The results of these determinations indicated hydrogen values of the same order as those shown by the recording instruments. The readings of the dissolved-oxygen recorder were also checked using a modification of the Winkler test.

Period 2: Hydrogen, Iron, and pH. The work covered by this period was carried on during the interval, October, 1944, to April, 1945, after two additional hydrogen recorders had been acquired. The two original recorders had been set up permanently to sample the feedwater entering the economizer, and the steam leaving the desuperheater. One of the new recorders was set up at the hotwell-pump discharge, and the second one was arranged so that it could be moved to various locations along the line of feedwater flow.

The pH of the water was measured at the sampling points and samples were taken for the determination of iron. The pH was measured by an electrometric meter using a flow cell located at the point of sampling. It was found to be essential in the measurement of pH of water of high purity that the samples be not exposed to the air. Large errors may result from absorption and possibly the release of gases. Water samples of 1 liter were taken in hourly increments for the iron determination. A laboratory sample of 500 ml was evaporated to dryness in a dust-free atmosphere, then taken up with 1 ml of hydrochloric acid and diluted to 50 ml. The iron was determined by a colorimetric method using a photoelectric colorimeter.

In the determination of these very small quantities of iron

extreme care was necessary to avoid contamination of the samples. For this reason the samples were not filtered as this operation would involve a hazard of contamination through exposure to the air. Although the samples were all crystal-clear, occasionally one would show an inordinately high iron content, and it was suspected that an unnoticed particle of sludge had been included. In such cases the result was discarded.

The results of a number of tests are shown in the graphs in Fig. 4. Various conditions of load and pressure are represented. The values for dissolved hydrogen, pH, and iron are plotted for the various sampling points. The plotted points represent the average of daily samples taken over a 6- to 8-day period. It was not possible to take samples from all points during any one test period and usually 6 points were selected for each test. The selection was varied for different tests so that samples at all points were obtained during the course of the study.

For the study of hydrogen the plan was to have one meter stationed at the hot well and to advance the second one successively to stations along the line of flow. The hot-well sample consistently showed zero hydrogen throughout the study. No hydrogen was found at the deaerator outlet or the 10th-stage-heater outlet. Sporadic readings of low values were obtained at the discharge of the low-pressure boiler feed pump. From this point on there was a gradual increase in the quantity of dissolved hydrogen along the path of flow to the maximum values shown for the superheated steam. This study shows that all of the hydrogen which appears in the boiler feedwater originates in the feedwater system. The hydrogen in the steam is removed

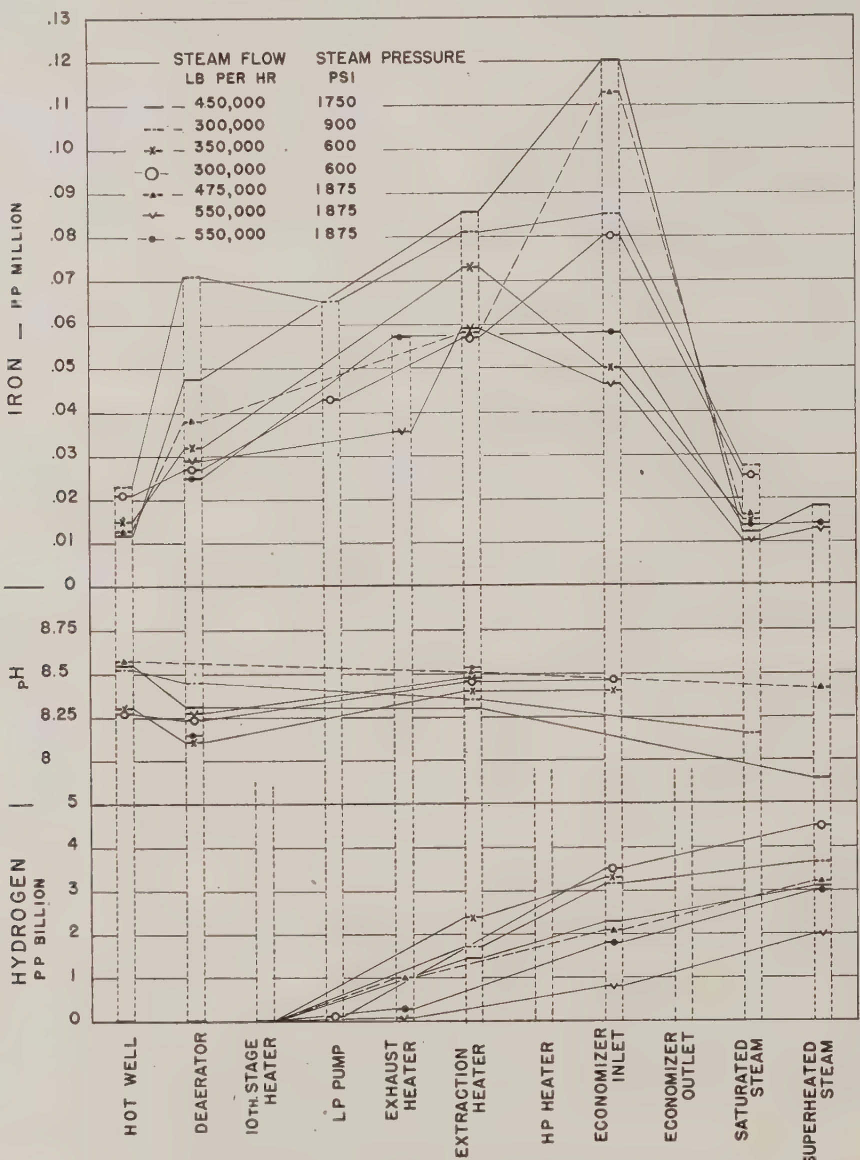

Fig. 4 Curves Showing Values for Dissolved Hydrogen, pH, and Iron Plotted for Various Sampling Points

TABLE 4 RESULTS OF TESTS TO DETERMINE AMOUNT OF COPPER IN FEEDWATER

COPPER - PARTS PER MILLION

SAMPLE	TEST NO. 3 9/19	9/20	9/22	9/24	9/27	AVERAGE TEST 3	AVERAGE TEST 2	AVERAGE TEST 1	TEST 1-2-3
15TH STAGE HEATER DRIPS	.007	.005	.004	.004	.007	.005	.009	.007	.007
HOT WELL	.009	.022	.007	.006	.009	.011	.009	.012	.011
DEAERATOR OUTLET *	.012	.006	.007	.007	-	.008	.011	.012	.011
DEAERATOR OUTLET **	.013	.008	.009	.009	.017	.011	.011	.014	.012
SATURATED STEAM	.008	.007	.006	.006	.013	.008	.009	.012	.010

AMMONIA NITROGEN - PARTS PER MILLION

SAMPLE	9/19	9/20	9/22	9/24	9/27	AVERAGE TEST 3	AVERAGE TEST 2	AVERAGE TEST 1	TEST 1-2-3
15TH STAGE HEATER DRIPS	.10	.10	.15	.10	.08	.11	-	-	.11
HOT WELL	.15	.15	.20	.20	.18	.18	-	-	.18
DEAERATOR OUTLET	.12	.15	.15	.18	.18	.16	-	-	.16
STEAM	-	.18	.20	.25	.25	.22	-	-	.22

* COPPER COOLING COIL
** STAINLESS STEEL COOLING COIL

in the condenser and none is recirculated to the feedwater.

The pH value of the samples varied over a relatively narrow range during this series of tests. The pH values at the hot well were between 8.3 and 8.6. There was a reduction in the deaerator due to removal of gases, apparently NH_3. The rise in pH at the extraction heater is ascribed to the effect of the treatment chemicals that are introduced ahead of that point, although it may be due in part to vapor from the evaporators which is introduced into the deaerator.

The values for the iron content of the water at the various sampling points are given in the upper group of points. The points of each test are connected by lines to form curves only the better to identify the several points of the test. While there is wide divergence in values, particularly at the economizer inlet, the points of each test form the same general pattern. No relation is evident between the iron content of the water and changes in pressure and boiler load. It may be concluded only that iron and dissolved hydrogen increase together. If we may be permitted to use the average of the rather scattered points for iron values at the economizer inlet, it may be pointed out that this average value corresponds closely to the amount of hydrogen in the superheated steam. From the equations given earlier, of the probable reactions for the dissolution of iron, one part per billion of hydrogen is the equivalent of 0.021 ppm of iron. The average hydrogen of the steam is 3.3 ppb corresponding to 0.069 ppm of iron. The average of the economizer feedwater is 0.079 ppm iron. There is as close an agreement as could be expected.

The miscellaneous projects listed under Period 3 represent work done at various times during the course of the investigation.

Period 3 (a): Copper in Feedwater. The measurement of copper in the feedwater was made because of the presence of that element in all deposits found in the boiler. As the condensers were suspected of being the principal source of contamination from copper, the study was limited to this equipment. Samples of condensed steam from the 15th-stage-heater drips were taken as representative of steam delivered to the condenser and a sample was taken at the discharge of the hot-well pumps for comparison. Samples were also taken of water leaving the deaerator and saturated steam leaving the boiler.

The analysis was made on 500-ml samples evaporated to dryness in the same manner as for iron. The determination of copper was by a colorimetric method. As there was a question of possible contamination of the samples by the use of copper cooling coils, a comparison was made between copper and monel coils and between copper and stainless-steel coils. There was no indication of contamination from the copper coils and in fact the very small differences in copper content of the samples from the three coils favored the copper coils. The sample reported is that taken with the stainless-steel coil.

The results of the series of tests are given in Table 4. Daily results for a 5-day period are given with the average for this period and the average for two similar periods. While the amounts of copper reported are quite small, there is a consistent increase across the condenser. The indication of copper in the condensed-steam sample is rather surprising. These samples were condensed in a monel coil, and there is a possibility of contamination from that source. The comparative tests of monel and steel coils at the deaerator outlet where the sample was cooled only in the coil cannot be taken as assurance that there is no contamination under condensing conditions.

It should be noted that there are other possible sources of copper contamination than the condenser that were not investigated. These are the 10th stage heater and low-pressure pumps having bronze impellers. There are also steam-air heaters in the coal-preparation house, the drips from which are returned to the feedwater system.

Period 3 (b): Bead and Ion-Exchange Columns. The plating out of iron-rich deposits on conductivity cells, glass tubing, and rubber tubing through which samples of feedwater and steam were drawn has been observed in a number of plants.

At the Somerset Station attention was first called to this phenomenon by the necessity of cleaning the conductivity cells and their glass containers at least once a week. It was first assumed that the deposit was a result of oxidation of a ferrous salt and the deposition of the more insoluble ferric compound. The cell and thermometer were inserted in the container through a rubber stopper, not necessarily an airtight seal. It was, however, noted that the glass and rubber connections between the cooling coil and the cell, where there was little possibility of air leakage, were also coated with a similar deposit.

This deposit was easily removed by washing with dilute hydrochloric acid. Usually a slight to strong odor of hydrogen sulphide was noted when the cell on the steam sample was washed.

In order further to study this plating-out effect, bead columns were made up consisting of $^1/_2$-in-ID pyrex tubing packed with $^1/_8$-in. soda-glass beads. These were set up at sampling points so that a sample could be drawn through them slowly and at a measured rate. The sample stream was arranged to flow across the top of the column into a spill tube, and the sample to the column was drawn from this stream. There was thus no syphon effect and the pressure at the connections was always above atmospheric. The rate of flow was adjusted to 1 liter per hr, and the tests were continued until 25 liters were drawn through the column. The effluent was measured in 10-liter graduated bottles.

A synthetic-resin ion-exchange column, operating on the hydrogen cycle, was prepared and tested together with the bead column. It had substantially the same dimensions as the bead column and took a sample from the same stream and at the same rate of flow. The purpose of the test was to determine the possibility of the quantitative removal of small amounts of dissolved salts and their recovery in the column wash in more concentrated solution. If successful, this method could be used instead of the usual one involving the evaporation of large samples, except, of course, where the determination of silica is involved. The exchange medium used was an analytical grade of a commercial product.

Daily samples were taken of the influent and analyzed for iron. Electrical conductivity and pH were measured for both influent and effluent. At the end of the sampling period the deposit in the bead column was dissolved with dilute hydrochloric acid, and the total solids as chlorides weighed. Iron and copper were determined in the residue. The ion exchange column was regenerated with dilute hydrochloric acid followed by a rinse with distilled water, and the extracted solids were recovered. Total solids as chlorides, iron, and copper were determined.

The results of two tests, one on a sample from the deaerating heater and the other on a sample from the extraction heater are given in Table 5. The ion exchange column is designated by the letter A and the bead column by B. The columns were in operation for 8 hr daily for a period of 3 days. Based upon the inlet and outlet samples, the recovery of iron on test 1 was 100 per cent for the A and 50 per cent for the B column. The actual recovery by extraction of the columns was 63 per cent for A and 25 per cent for B. It was found that the extraction of the ion-exchange bed was incomplete, and in the second test the acid strength was increased with resulting better recovery.

The recovery of iron for test 2 on the basis of inlet and outlet samples was 69 per cent for the A column and 39 per cent for the B column, compared with 99 per cent and 39 per cent, respectively, by extraction. The values by extraction are believed to be more nearly correct as at the higher concentration a greater accuracy of analysis is possible.

It will be noted that both conductivity and pH are reduced by the bead column which might indicate that the iron removed is in the form of ferrous hydroxide. The changes in conductivity and pH across the ion exchange column are what would be expected. When the positive ions are retained in the ion exchange process an equivalent amount of acid is produced.

The principle involved in the plating-out effect exemplified in the bead column is not known but is believed to be a surface-adsorption effect. Iron appearing in the deposit is in the form of ferric oxide; hydrated ferric oxide and magnetite. Silica also is present, its source probably being the soft glass beads. It is probable that the silica in the deposit is incidental and is not involved in any reaction with the iron. The iron deposits occur in pyrex tubing, rubber tubing, and on the inner surfaces of the various cooling coils. The entire feedwater system is undoubtedly lined with this deposit and probably much of the iron de-

TABLE 5 RESULTS OF TESTS TO DETERMINE TOTAL SOLIDS AS CHLORIDES, IRON, AND COPPER USING BEAD AND ION EXCHANGE COLUMNS AT DEAERATING HEATER AND AT EXTRACTION HEATER

TEST NUMBER	1						2					
SAMPLE	DEAERATOR OUTLET						EXTRACTION HEATER OUTLET					
SAMPLE VOLUME	25 LITERS						25 LITERS					
WATER TO COLUMNS	1	2	3				1	2	3			
IRON - PPM	.014	.012	.050				.069	.058	.056			
CONDUCTIVITY - MMHO	3.0	6.5	3.05				4.35	5.05	2.65			
PH	8.55	8.60	8.59				8.36	8.48	8.34			
	COLUMN						COLUMN					
	A			B			A			B		
WATER FROM COLUMNS												
IRON - PPM	0	0	0	.007	.006	.026	.011	.032	.013	.041	.043	.026
CONDUCTIVITY - MHO	8.0	15.5	6.3	3.5	7.25	2.95	9.84	9.85	3.70	4.0	4.7	2.35
PH	7.0	4.65	6.0	8.5	8.4	8.2	4.75	4.77	5.05	7.76	8.20	8.09
COLUMN EXTRACTION	MG	PPM		MG	PPM		MG	PPM		MG	PPM	
TOTAL SOLIDS	63.6	2.52		21.0	.84		68.0	3.72		3.50	.140	
IRON	.396	.0158		.159	.006		1.153	.060		.600	.024	
COPPER	.945	.0378		.635	.025		.530	.021		.325	.013	
RECOVERY OF IRON												
1 BASED ON INLET AND OUTLET	100 %			50 %			69 %			39.5 %		
2 BASED ON INLET AND RECOVERY	63 %			25 %			99.2 %			39.4 %		

posits noted in the boiler tubes may be accounted for by this property of plating-out on all surfaces.

The concentration of salt cations by means of the ion-exchange column did not work out well, principally because it was difficult to make a complete extraction of the retained material, and particularly the iron. While the residue remaining in the bed after regeneration was very small, even when the conventional wash was used, it represented a relatively large percentage of the total. This objection could be met by using a larger sample so that the unextracted portion would be negligible. The actual capacity of the bed was in only very small fraction realized by the 25-liter sample used. It is believed that this method of concentration of sample has promise and should be further explored.

Period 3 (*c*): *Tube-Temperature Measurement.* Tube-temperature measurements have been reported in another paper of this group. One purpose of these measurements was to determine whether or not there was any progressive increase in tube temperature that might be ascribed to internal deposits. Daily readings of temperatures of a group of tubes in the north wall gave no indication of appreciable internal deposits. The thermocouples were located at elevations of 5 to 25 ft above the hearth. A recent inspection following a tube failure at a 1-ft level, however, disclosed sludge deposits that would have been indicated by a couple at that elevation.

Period 3 (*d*): *Corrosion Detector.* In an attempt to obtain a quantitative check on the rate of solution of iron in the feedwater, a corrosion detector was installed in one branch of the boiler feed line leading to the economizer. This corrosion detector consisted of a short rod of carbon steel machined flat on two sides. It was mounted so that it could be removed without disturbing any of its original surface. This detector was recently examined after an exposure of 1 year. After being carefully cleaned in an ammonium-citrate solution, it was found to have lost 0.25 g of its original weight of 14 g. This loss if evenly distributed corresponds to 0.0001 in. penetration. There was, however, one small pit of about 0.005 in. depth. The surface was darkened and coated with a powdery film of what appeared to be an iron oxide. This corrosion detector was returned to its position in the feed line and will be again examined after a period of exposure.

The rate of metal loss, as indicated by the direct determination of iron content of the water and also by the measurement of the evolved hydrogen, is not large. Translated into inches penetration per year over the areas of the preboiler equipment and of the boiler itself, it would not be an impressive figure and would not lead to any great apprehension as to the imminent failure of any part of the equipment. There would, however, be cause for concern if the loss of metal were restricted to local areas such as has been noted in boiler feed pumps. While the hydrogen evolution appears to indicate a greater activity in the high-temperature end of the feedwater system, there is no evidence to indicate that localized attack occurs. It is, however, doubtful that loss of metal from erosion, such as may occur in feed pumps, is accompanied by the evolution of hydrogen.

The rate of loss of metal from the steam-side surface of the condenser tubes, as indicated by the measurements of dissolved copper, is extremely low. The loss on the cooling-water side is probably much greater, and also localized attack is known to occur. Failure of condenser tubes is more likely to occur as a result of attack on the outside, that is, the cooling-water side of the condenser.

One disquieting thing about the dissolution of iron and copper in the feedwater system is that the mechanism of the corrosion is not well understood, nor have preventative conditions been well established. It is generally agreed that dissolved ammonia is the principal factor in the corrosion of copper, and methods for the removal of this substance have been proposed. It is indicated that the dissolution of iron can be reduced by the control of pH. However, the permissible range of pH in the feedwater is limited by boiler-water conditions particularly when the rate of boiler-water blowdown is low.

The principal concern over the presence of iron and copper compounds in the boiler feedwater is caused by the propensity of these substances to form adherent sludges on the boiler surfaces. The iron is transformed probably completely to magnetic oxide and the copper is in large part at least reduced to the metallic state.

Slime deposits of sludge are found on all internal surfaces of the boiler and are often heavier on the unheated areas than on active steam-generating surfaces. It is indicated that this deposit is laid down mechanically, that is, there is no general chemical attack under sludge deposits. Small pits of pinpoint dimensions have been noted but are believed to be only incidental and not generally related to the deposits. While the sludge-collecting pots at the waterwall headers and a special sludge blowdown in the drum permit the removal during operation of accumulated sludge, they have not completely eliminated the hazard. The chief danger of the sludge deposit lies in its insulating effect on active steaming surfaces where it may build up to a thickness that results in overheating of the tube metal.

Basically the sludge problem in boilers results from the introduction into the boiler with the feedwater of solids or salts that form solids, at a higher rate than the rate of removal. Inevitably this must lead to trouble.

There are two obvious lines of attack on this problem: (*a*) to prevent or minimize the production of the metallic ions which are the principal components of the sludge in the feedwater system, and (*b*) to provide for a better continuous removal of sludge with the blowdown from the boiler.

The chances of complete solution of the problem by either or both of these efforts are not particularly promising and it is probable that a periodic cleaning of boilers will be necessary.

Apparatus and Methods

The hydrogen analyzer and recorder used in the investigation was a commercially available instrument utilizing the difference in thermal conductivity of gases. The analyzer consists of a Wheatstone bridge, two legs of which are formed by identical platinum-wire spirals. These spirals are electrically heated, a constant potential being applied. One of these spirals is surrounded by saturated air and the other by air that has bubbled through the sample which flows to the analyzer at a measured rate. Hydrogen of the sample is present in the latter air in proportion to its concentration in the sample. Because of the higher thermal conductivity of the air containing hydrogen, the temperature of the second spiral and hence its electrical resistance is reduced and the bridge is thrown out of balance. The balance is then restored automatically by a potentiometer which records the off-balance as hydrogen. At the usual low concentrations of hydrogen in feedwater the unit is parts per billion.

The principle of the dissolved-oxygen recorder is similar to that of the hydrogen recorder. Pure hydrogen which is generated electrolytically surrounds the reference spiral and hydrogen which scrubs the sample and has the equilibrium content of oxygen surrounds the analyzer spiral.

These instruments are extremely sensitive. Two parts of hydrogen in 10 billion parts of sample will give a scale deflection. It is necessary to check the zero of these instruments daily and it is desirable also to make frequent checks of the scale. This is done by introducing hydrogen to the sample at a measured rate. The hydrogen is made electrolytically and the rate of introduction of hydrogen is measured by current used.

While the meters are simple in principle, considerable attendance is required in order to assure their continuous accuracy.

The steam-sample evaporator is shown in Fig. 5. It consists essentially of a 5-liter flask having a tubulature extending downward with the open end in the evaporating dish in which the sample is being evaporated. The end of this tube is ground to a standard taper to which a platinum thimble is fitted. This thimble is weighed with the dish. As the water level in the dish recedes, the open end of the tube is uncovered allowing air to enter the flask and more water to flow into the dish. In starting the evaporator, the stopper is inserted at the end of the delivery tube and the measured sample, 3 to 4 liters, is poured in at the top opening. The cap with the manometer is then replaced and suction is applied to obtain an indication of about 1 in. on the manometer, and the stopcocks A and B are closed. The weighed dish is then placed in position on the heater assembly which is held on a ring stand below and to one side of the evaporator bell. The stopper is then removed and the heater and dish assembly quickly placed in the operating position. To do this without spilling the sample requires practice, and it is recommended that the operation be rehearsed with the flask empty until the necessary skill is acquired. The stopcock B is open during the evaporation.

The sample is evaporated without boiling, most of the heat being applied over the surface of the liquid. When the sample container is empty, the mercury switch closes and actuates a relay which shuts off the current to the heaters and closes the valve through which the air is admitted. There is enough residual heat in the heater and bell to complete the evaporation. At the end of the evaporation the flask is rinsed with a jet of distilled water and the washings added to the dish and evaporated. The rate of evaporation is about 200 ml per hr.

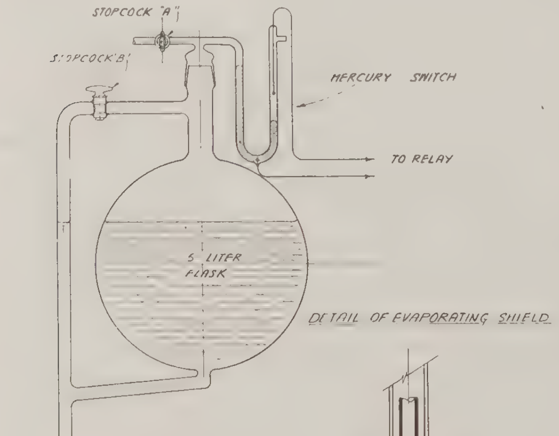

Fig. 5 Steam-Sample Evaporator

The sample concentrator for the determination of iron is shown in Fig. 6. It consists merely of a 1-liter Erlenmeyer flask covered with a vented hood through which filtered air is blown. A 500-ml sample is evaporated to dryness without boiling in about 4 hr. The residue is then taken up in 1 ml of concentrated acid and diluted to 50 ml for analysis.

Most of the analyses were made by a colorimetric method using thiocyanate. Measurement of color was by means of a photoelectric colorimeter. A second method using orthophenanthrolene was tried and although it proved more sensitive, it also required concentration of the sample and the simpler thiocyanate method was preferred.

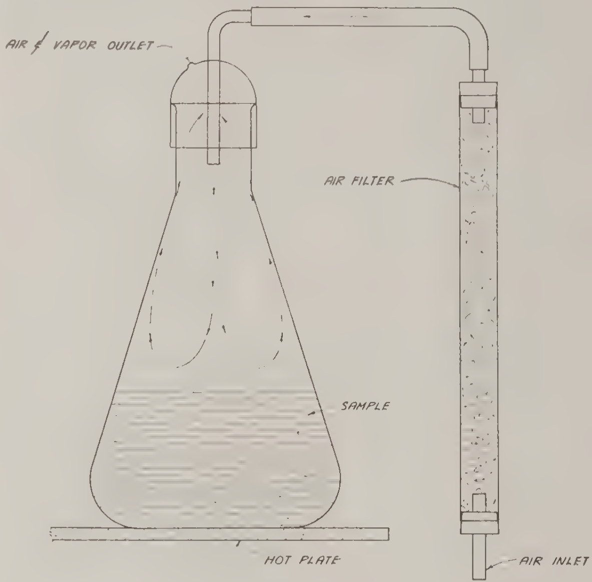

Fig. 6 Sample Concentrator for Determination of Iron

Extreme care is necessary to prevent contamination of the samples. It was found that if a sample was stored even for a day an appreciable amount of iron was lost because of the plating-out effect on the walls of the flask. To reduce the error due to the plating-out tendency, a small measured amount of acid and an organic reducing agent were placed in the flask in which the sample was collected, and the storage time was reduced to a minimum. Any iron present in reagents was corrected by means of blank samples.

Samples for copper determinations were prepared in the same apparatus used in the iron analysis. A colorimetric method using sodium diethyl-dithio carbamate in alkaline solution was used. This method was generally very satisfactory. In a very few instances an interfering turbidity resulted and it was necessary to extract the color with an organic solvent.

The bead-column and the ion-exchange-column arrangements are shown in Fig. 7. The columns were set up at the point of sampling. The sample flow was arranged so that a constant head was provided and also so that all connections were under pressure and no air would be drawn into the columns. The rate of sample flow was 1 liter per hr, and the effluent was collected in calibrated bottles so that the total flow could be measured.

At the end of a sampling period the assembly was taken to the laboratory where the extracts were removed by acid treatment and the columns prepared for another run. A 10 per cent

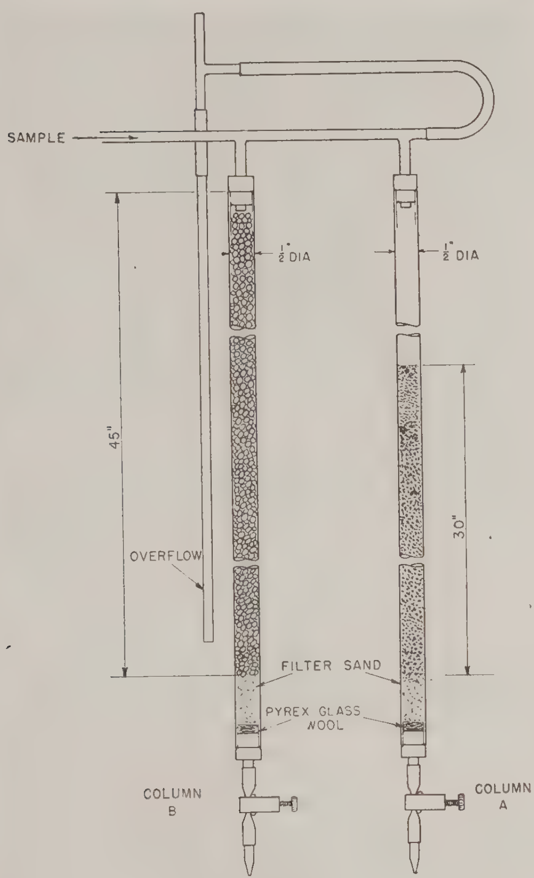

Fig. 7 Bead Column and Ion Exchange Column Arrangements

hydrochloric acid, 300 ml in amount, was used in regeneration of the ion exchange column, followed by a 300-ml water wash. Distilled water that had previously been passed through the bed was used for the wash water.

The 600-ml recovery sample was reduced to 500 ml for convenience and analyses were made on aliquot portions. Colorimetric methods previously described were used for the iron and copper determinations.

Discussion

R. C. Corey.[4] The excellent experimental data presented by the authors serve, among other things, to place one important phase of corrosion in high-temperature steam cycles on a factual basis, and those concerned with iron-oxide sludges now have available something more substantial than opinion or, as is frequently the case, " . . . . ill-begotten equations supported by bad data;" as it is stated by Lewis and Randall.[5] The experimental work conducted by the authors to establish the source of iron oxide in boilers and the probable mechanism of its formation, by means of hydrogen and dissolved-iron surveys of the steam and water cycles, is unique to the extent that no published data from similar work are to be found elsewhere. In some respects, which will be mentioned later, their results confirm certain conclusions of laboratory and theoretical studies of the fundamental reactions between iron and pure water. To this end there is support of the contention that there are yet many more questions than there are answers to the problem of corrosion by pure water at elevated temperatures.

It is a curious fact that although one may find in corrosion literature data on the rate of corrosion of iron and steel in contact with almost every conceivable inorganic or organic material, under widely different conditions of independent variables, there are no published data on the rate of corrosion of iron and steel in pure water. A possible reason for this may lie in the fact that in pure water the rate of corrosion is insignificant compared to that in acids, bases, and certain salt solutions. It is obvious, however, from the results of the present paper and the current problem of feed-pump corrosion which appears to be related in some way to pure condensate, that there is a definite need for an investigation of the rate of corrosion of iron and steel in pure water up to around 600 F, and under various conditions of flow and pH.

Evaporated feedwater make-up and highly efficient steam-washing and deaerating equipment result in feedwater of extreme purity, and as so clearly expressed recently,[6] "It dissolves to some extent anything that it comes in contact with on its way from the condenser to the boiler. This dissolving action starts in the last stages of the turbine where the first condensation occurs." It may be asked what is meant by "pure" water. Qualitatively it may be defined as water which contains no dissolved gases and which produces no residue on evaporation. A quantitative definition for pure steam condensate might be based upon its electrical conductivity when there are no ions present other than hydrogen and hydroxyl ions resulting from the ionization of the water.

On the other hand, it is possible that the conductivity of pure water may not change after contact with an iron vessel, due to the formation of Fe_2O_3 or FeOOH which are insoluble. In such a case the buffer index might be a useful criterion for purity, this being defined as the differential ratio dB/d(pH), in which d(pH) is the increment of pH when an increment of base dB, in terms of gram equivalents of hydroxyl ion, is added. An increment of hydrogen ion would be equivalent to —dB. Thus pure condensate would have a buffer index of 0.1429, and if iron oxide were added this value would increase due to the fact that the oxide would react with added hydrogen ion.

The authors have shown clearly that in the subject plant most of the hydrogen and dissolved iron result from corrosion of external piping and heat-exchange equipment, a very small amount being due to corrosion within the boiler. They postulate that the ferrous hydroxide formed in the primary corrosion reaction decomposes to Fe_3O_4 and more hydrogen. There is experimental and theoretical work to support this. Schikorr[7] found that when iron turnings were placed in contact with pure water, pressures of 18 atm and higher of hydrogen developed, and Fe_3O_4 was formed. In later experiments he demonstrated that ferrous hydroxide spontaneously decomposed to hydrogen and Fe_3O_4 at a rate inversely proportional to the pH.[8] Investigation of the

[4] Research and Development Department, Combustion Engineering Company, Inc., New York, N. Y.

[5] "Thermodynamics," by G. N. Lewis and M. Randall, McGraw-Hill Book Company, Inc., New York, N. Y., 1923.

[6] "Properties of Pure Boiler Feedwater," *Combustion*, Feb., 1945, p. 45.

[7] "On the System Iron-Water," by G. Schikorr, *Zeit. für Elektrochemie*, vol. 35, 1929, p. 62.

[8] "Ferrous Hydroxide and a Ferromagnetic Ferric Hydroxide," by G. Schikorr, *Zeit. für Anorganische und allgemeine Chemie*, vol. 212, 1933, p. 33.

reactions at room temperature between iron and pure water[9] demonstrated that Fe_3O_4, hydrogen, and about 0.2 ppm of dissolved ferrous iron were the final products, and the pH rose to 8.3. The dissolved iron was considered to be an intermediate stage in a chain of reactions,

$$Fe \rightarrow Fe^{++} \rightarrow Fe(OH)_2 \rightarrow Fe_3O_4$$

No attempt was made, however, to correlate the dissolved-iron concentration and pH because of (*a*) lack of information on the state of the dissolved iron, that is, whether it was colloidal or poorly ionized; (*b*) disagreement in the literature as to the solubility and the solubility product of $Fe(OH)_2$; and (*c*) inability to determine which of the foregoing reactions controlled the rate to the right. Thermodynamic calculations have shown[10] that iron reacts with oxygen-free water to form Fe_3O_4 more readily than $Fe(OH)_2$, although it is conceded possible that the latter may be an intermediate phase. Thus it is a necessary conclusion that $Fe(OH)_2$ will decompose to Fe_3O_4 and hydrogen.

All of these facts find confirmation in the results obtained by the authors if it is kept in mind that it is very unlikely that equilibrium between iron and condensate ever is attained in the short time of contact between the hot well and the boiler. Thus the relative amounts of Fe^{++}, $Fe(OH)_2$, Fe_3O_4, and H_2 may be expected to vary considerably from equilibrium values.

One question concerning these investigations with iron and pure water is whether the iron determinations represent only dissolved iron or varying mixtures of dissolved iron and an extremely finely divided form of magnetite.

In one of the afore-mentioned investigations,[9] there were indications that despite careful filtering a small amount of iron oxide, possibly of colloidal dimensions, occasionally contaminated the samples. This is merely a question of academic interest, related to the fundamental corrosion reaction, and therefore does not affect the results and conclusions of the authors.

It may be asked, why, if the rate of corrosion of iron and steel by pure water at elevated temperatures is insignificant when uniformly distributed throughout the system, it receives the attention it does in the present paper. The authors have very clearly resolved the problem in two parts: (1) There was the question whether the iron oxide originated in the boiler as the result of attack of steaming surfaces; and (2) the possibility that excessive deposits of iron oxide on steaming surfaces would lead to overheating due to a decrease in the over-all coefficient of heat transfer. In so far as the subject plant was concerned, it was demonstrated that the greater part of the iron oxide was carried into the boiler from external sources. This confirms the consensus of a number of operators. Regarding the ultimate effect of deposits of iron oxide on steaming surfaces, there is, in addition to the insulating effect, the possibility that boiler-water salts may concentrate in and beneath the oxide, and if sufficient free caustic is present localized, rapid attack may occur.[11]

Another factor that may be considered is the possibility that metallic copper, invariably associated with iron oxide in boilers, deposited on a baked-on sludge of iron oxide may set up an electrolytic cell, the tube metal being anodic, the copper cathodic, and the iron oxide between, serving as a porous matrix for salt-saturated liquid. This, of course, is hypothetical and until data are available on the emf between copper (or copper oxide) and iron, and on time-potential curves, which indicate whether anodic or cathodic polarization is controlling the rate of corrosion, this theory offers possibilities.

Thus as suggested by the authors, iron-oxide deposits in high-pressure boilers cannot be ignored, and if it has been established that excessive quantities of iron oxide are being carried into a boiler from outside sources, then steps should be taken to minimize corrosion in external equipment even though the rate is not sufficient to cause significant attack in such equipment itself. Theoretically attack can be minimized either by raising the pH of the feedwater or by applying a protective coating to the ferrous materials with which the feedwater comes in contact. Assuming, however, that there were no practical difficulties to the first of these steps, the fact remains that we do not know the optimum pH at which the feedwater should be maintained to minimize corrosion under operating conditions, because no investigation has been made of the rate of corrosion of steel under these conditions at various values of pH.

The data on hydrogen and iron throughout the system are extremely interesting. Regarding the increase in hydrogen across the economizer, not shown in Fig. 4 of the paper, it would be of interest to know what values were obtained at the economizer outlet and also the iron determinations, if available. If the iron and hydrogen did not increase together then it might be inferred that the increase in the hydrogen was the result of decomposition of ferrous hydroxide rather than corrosion in the economizer.

If the authors have made any conductivity determinations at the various sampling points, they would be a valuable supplement to the other data and may help to explain some questions of fundamental importance.

It was noted that the concentration of hydrogen in the steam did not vary with the load. This confirms experiments conducted by the American Gas & Electric Service Company at Logan, where the concentration was found to be constant at about 3 ppb between a steam output of 400,000 and 850,000 lb per hr.[12] The average concentration at Montaup was 3.3 ppb.

Regarding the results for dissolved iron, it is interesting to note that the iron in the steam was approximately the same as found in the hot well. It would be of interest if the authors described the method they adopted for determining the iron, that is, whether it was a modified thiocyanate, orthophenanthroline, or similar scheme.

At the present time little can be said about the manner of formation of metallic copper in the boiler or its effect with regard to corrosion. It has been stated[13] that the corrosion of copper or copper alloys by ammonia is negligible in the absence of oxygen. Relatively, this may be so but it appears that in oxygen-free condensate sufficient solubility, in the presence of less than 0.1 ppm NH_3, occurs to produce large amounts of copper in boilers. Dissolved carbon dioxide, either as free CO_2 or bicarbonate ion, also is a factor in the corrosion of copper.

The corrosion detector, installed in one branch of the boiler feed line leading to the economizer, offers excellent possibilities for determining, under actual operating conditions, the rate of corrosion of low-carbon steel. It is suggested, however, that it is not advisable to re-use the specimen for a subsequent test, more acceptable results being obtainable if new specimens are used for each test. Also, if possible, it would be desirable to install an identical specimen in a location where the temperature and pH of the liquid are the same but of different velocity.

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[9] "The pH, Dissolved Iron Concentration and Solid Product Resulting From the Reaction Between Iron and Pure Water at Room Temperature," by R. C. Corey and T. J. Finnegan, Proceedings of the A.S.T.M., vol. 39, 1939, pp. 1142–1260.

[10] "Thermodynamic Considerations in the Corrosion of Metals," by J. C. Warner, Trans. Electro Chemical Society, vol. 83, 1943, pp. 319–333.

[11] "Co-Ordination of Water Conditioning With Operating Problems," by R. E. Hall and C. E. Kauman, *Power Plant Engineering*, vol. 45, September, 1941, pp. 61–64, and October, 1941, pp. 59–61.

[12] Publication No. H-11, Edison Electric Institute, 1940.

[13] "The Reaction of Copper and Oxygen-Saturated Ammonium Hydroxide," by R. W. Lane and H. J. McDonald, *Journal of Corrosion*, Aug., 1945, pp. 1–8.

The experiments with bead columns to study the effectiveness of various materials in removing small amounts of solids from water are of considerable interest, particularly with regard to the comparative efficiency of the glass and the synthetic resin.

The authors suggest that silica is not involved in the action between the glass beads and iron-bearing water. The writer submits that in experiments with pure water and iron in pyrex vessels there seemed definitely to be reaction of dissolved iron with the glass. The pH of the water rose to 9.0–9.5 and a flaky anisotropic material formed which microchemical tests showed to contain iron and silicon; possibly being a hydrated iron silicate.

In conclusion, it is believed that this investigation has emphasized the need for more fundamental work on the reactions of iron with pure water. A program for such an investigation might be as follows:

1 Determination of the rate of corrosion of iron and steel in water as a function of pH, velocity, and temperature, up to 600 F.

2 Study of the time-potential curves for iron under the same conditions as item 1.

3 Preparation of pure ferrous hydroxide and determination of the following:

(a) Its solubility and pH in pure water.

(b) Rate of decomposition to Fe_3O_4 and H_2 as a function of temperature and pH.

The results of such an investigation should, among other things, determine the minimum pH of feedwater required to effect maximum protection of steel surfaces, at various temperatures related to operation.

E. R. Mueller.[14] Of interest primarily to chemists rather than engineers is the reference given in the subject paper to the use of an ion-exchange material as a tool in the analysis of feedwater or steam samples. The application discussed is a clear-cut example of the manner in which the analytical grade Amberlite resins can be used as analytical tools. The investigators are taking advantage of the unique properties of these resinous exchangers, in this case, the cation exchanger, one such property being the complete adsorption of heavy metal ions such as iron, copper, etc.

It has been stated in the paper that dilute hydrochloric acid was first employed with incomplete recovery of the full metal content from the exchanger. Although the concentration was not indicated in the paper, this was pointed out to be 5 per cent. By the use of 10 per cent acid, the higher concentration referred to in the paper, better results were obtained. It is suggested that in both cases more complete recovery of the metal ions from the exchanger could have been accomplished by the recycling of a relatively small but adequate portion of either concentration of acid. This technique would have the added advantage of keeping the eluting solution at the smallest possible volume, thereby reducing the time required later for evaporation.

Since the resinous exchanger is entirely organic in composition, it is suggested that the analytical technique might be improved upon by wet-ashing the entire sample of exchanger upon which the metal ions have been adsorbed. This would eliminate the elution step altogether, thereby insuring the analyst against an error in the results brought about by incomplete elution of the metal ions.

In general, the particular problem represents an unusual application of the properties of ion-exchange resins.

[14] The Resinous Products & Chemical Company, Philadelphia, Pa.

W. F. Ryan.[15] A becoming but regrettable modesty has restrained the authors of this paper from acclaiming the outstanding achievement of their part of the investigations at Montaup. The statement "recording hydrogen meters were set up" is a masterpiece of condensation. The "setting up" of these meters required many weeks of painstaking engineering, involving a degree of scientific knowledge, technical skill, patience and determination more readily associated with the instrumentation of the atomic-bomb project than of a steam boiler. Perhaps Mr. White can be persuaded at a later date to publish in detail this story of measuring hydrogen in fractions of a part per billion under the pressure and temperature conditions prevailing in this station, and securing accurate and continuous records, in spite of the difficulty in maintaining reasonable equilibrium between the fluids being analyzed and the metals which contained them.

The hydrogen-meter installation was expensive not only in money but in engineering manpower, and there were many discouraging periods during which the patience of operators and contractor's representatives was sorely tried, and many of us would cheerfully have thrown the project overboard. Fortunately, however, the work was carried to successful completion, and the importance of the results well repaid the time and effort expended.

The boiler, presumably in common with all boilers working at elevated temperatures, is subject to continuous, but infinitesimal, widely distributed, and therefore substantially harmless, oxidation. It may also be subject, at irregular and unpredictable intervals, to intensified localized oxidation. When the meter shows a sudden or abnormal evolution of hydrogen in the boiler it is a clear indication that such oxidation is taking place; the meter is raising a danger signal as significant as low water in a gage glass and calls for equally prompt and vigorous action.

We are feeding all of the high-pressure boilers in the country with water contaminated by oxides. The deleterious effects may not be serious in some instances but it is not of record that it ever did a boiler any good. We are now groping in the dark to combat, in the boiler, a contaminant which should never have been allowed to enter it, and there have been widespread difficulties with boiler feed pumps and high-pressure heaters that are all a part of the general problem. Mastery of the hydrogen-meter technique offers an opportunity at least to locate the source or sources of the trouble. It is hoped that some utility or combination of utilities may sense the importance of this possibility and undertake to measure hydrogen evolution throughout a high-temperature cycle from the superheater outlet to the condenser and back again through the feed heaters, feed pumps, economizer, boiler, and superheater. When this work is undertaken, the project will be greatly facilitated by the solution at Montaup of the many and difficult technical problems involved.

T. J. Finnegan.[16] The record of hydrogen throughout the cycle given in this paper by Bissell, Cross, and White adds confirmation to the opinion held by many persons that much of the loose black-oxide sludge in the boiler is of external origin. They do not accept the theory voiced by some extremists that none of the black oxide is formed in the boiler, except perhaps where overheating occurs. They support their opinion by the implied acceptance of the two equations which would explain the formation of hydrogen by decomposition of $Fe(OH)_2$ rather than by direct attack of the metal. In other words, they admit the possibility that the fundamental corrosion reaction

[15] Assistant Engineering Manager, Stone & Webster Engineering Corporation, Boston, Mass. Fellow A.S.M.E.

[16] Chemical Engineer, Buffalo, Niagara Electric Corporation, Buffalo, N. Y. Mem. A.S.M.E.

$$3Fe + 4H_2O = Fe_3O_4 + 4H_2$$

might occur in two steps, the velocity of which is strongly influenced by temperature.

While it is almost certain that the fundamental reaction is a chain one, it has yet to be established that the separate steps of the chain can occur at a delayed rate. For example, it is easily shown that when iron is placed in oxygen-free water at room temperature the products are loose Fe_3O_4 and hydrogen. Whether or not a large amount of $Fe(OH)_2$ may be mixed with the Fe_3O_4 and have potentiality for decomposing at higher temperatures is an interesting question.

Theoretical speculations of this sort as well as those referring to the reaction of copper given in the comment on the limitations of the hydrogen recorder in the paper by Bissell and Powell[17] make one admire the attempts which are made to give a rational explanation to the observed phenomenon, but they also emphasize the need for more fundamental work on the pure chemistry of the water-metal systems. We are continually being confronted with the results of chemical reactions, the nature of which we do not understand because they have not been thoroughly studied, especially over extended ranges of temperature.

An investigation made some years ago showed that the products of the reaction of iron with oxygen-free water at room temperature are hydrogen and Fe_3O_4.[18] The oxide did not form a protective coating but settled as a loose powder on the bottom of the reaction vessel. This investigation was made in a power-company laboratory which was not equipped for carrying out the obvious extension of the work, and it would be interesting if some research organization would repeat the experiments and continue them into higher-temperature regions, with particular attention to the matter of protective coatings, the existence of $Fe(OH)_2$, and to any factors which tend to limit the reaction.

Authors' Closure

The authors appreciate Mr. Corey's very carefully prepared discussion. His observations on the possible reactions that may take place in the feedwater system emphasize the need for further work on that subject. While no quantitative data were obtained, there is evidence that iron in the feedwater at the Somerset Station occurs both in colloidal and dissolved states. The iron is reported as total iron as the samples were evaporated to dryness and the residues were taken up with acid. In some samples where the iron content was high enough to be determined without concentration of the sample, lower values for iron were obtained on the sample before evaporation than on an evaporated sample. When the sample was acidified and heated increasing values for iron were obtained with increasing time of digestion.

[17] "Experience With Instruments and Control Equipment for 2000-Psi Boiler at Somerset Station of Montaup Electric Company," by W. D. Bissell and E. B. Powell, published on pages 453-466 of this issue of the Transactions.

[18] "pH, Dissolved Iron Concentration and Solid Product Resulting From Reaction Between Iron and Pure Water at Room Temperature," by R. C. Corey and T. J. Finnegan, Proceedings of the A.S.T.M., vol. 39, 1939, p. 1242.

There is also evidence that much of the iron is in the ferrous state. For example, in bead column tests a heavier deposit is obtained when the sample is aerated. The deposit is also of a deeper red color.

Mr. Corey suggests that there may have been a reaction between the glass of the beads and the iron in the water. Our conclusion that the silica was only incidental was based on the fact that the same type of deposit occurred in rubber-tube connections and also in the copper tubes of the cooling coils. This, of course, does not preclude the possibility that an iron-silica reaction does take place—the source of the silica being other than the glass.

While the hydrogen content of the steam is reasonably constant in the record shown in Fig. 2, the variation in rating is not large. There is, however, a definite trend of increase in hydrogen concentration with decrease in load. This may be noted in the hydrogen curves of Fig. 4 and also in Fig. 3. The change in the hydrogen values is not large and is sometimes masked by other unexplained variations.

As reported in the paper the corrosion indicator was removed for weighing at the end of one year and was at once restored to service in the pipe line. A duplicate corrosion indicator of carbon steel was machined, however, and was installed in October, 1945, in place of the original one, just as Mr. Corey has proposed.

His suggestion is that an additional corrosion indicator be installed where it would be exposed to identical conditions except for velocity would yield informative data. Because of daily and seasonal variations in boiler load, however, the present indicator is exposed to appreciable changes in velocity, and under existing conditions it appears impracticable to select a second location where the feedwater velocity would remain in approximately constant ratio to that at the existing corrosion indicator.

The valuable suggestions of Mr. Mueller are appreciated. The advantages of the ion exchange method of sample concentration are many and the results so far are encouraging. The alternate method of evaporation is laborious and time-consuming and it requires a very careful technique for its successful application.

The authors are gratified by Mr. Ryan's very kind discussion and appreciate his comments on the broader significance of the investigation.

The authors must agree with Mr. Finnegan that in what little experimental work has been done on the reactions of water and iron, there is little or no evidence of ferrous hydroxide in the corrosion products. The experiments, however, have been conducted under conditions such that chemical equilibrium was established. It is doubtful that such equilibrium could be attained in the feedwater system at the Somerset Station.

Recent analyses of bead-column deposits indicate that 50 to 70 per cent of the iron is in the ferrous state. The work is being continued and the feedwater system will be explored for the ferrous ferric ratio. There is no question but that the final product is Fe_3O_4. Under nonequilibrium conditions the ratio of ferrous to ferric iron should depend upon the relative reaction rates of the formation and decomposition of ferrous hydroxide.

Water Conditioning for the 2000-Psi Boiler at the Somerset Station of Montaup Electric Company

By W. W. CERNA[1] and R. K. SCOTT[1]

Considerable interest has been aroused in the chemical treatment used for the high-pressure boiler at the Somerset Station of the Montaup Electric Company. The theoretical aspects of the present treatment, which has been in use since January 1, 1944, are given in a paper by Hall.[2] Details of the results obtained with the different types of treatment used on this boiler have been given in a paper by Parks.[3] It is largely the purpose of this paper to present the reasons for the various methods of chemical treatment which have been used, and for the changes made, which have resulted in stabilization of the boiler-water conditioning since January 1, 1944.

Period of Sodium Treatment

When the high-pressure boiler of the Somerset Station first went into operation in the summer of 1942, a fairly standard treatment, using sodium chemicals, was used with the following control limits:

Alkalinity as sodium hydroxide, NaOH, ppm	50–60
Sodium phosphate, Na_3PO_4, ppm	50–70
Total solids, ppm (max)	1000

Some months later it was also decided to feed and maintain a positive reserve of sodium sulphite in the boiler water. During the period this treatment was used several tube failures occurred, as discussed in a previous paper of this series by Parks, Patterson, and Ryan.[4] Internal boiler conditions were also found unsatisfactory as described in the reference paper by Parks.[3] Evidence of the "hide-out" of sodium sulphate and sodium phosphate was also found which indicated that some of the boiler water was being concentrated very much beyond the limits of the normal over-all boiler water.

For example, using the published data of Schroeder, Berk, and Gabriel[5] for the solubility equilibria of sodium sulphate and sodium phosphate at boiler temperatures, a boiler water at 595 F, containing 350 ppm of sodium sulphate (Na_2SO_4) and 70 ppm of trisodium phosphate (Na_3PO_4) representative of that in the Montaup boiler at the time, would have to concentrate about 185 times before the first deposition of these substances would commence. Neglecting the effect of the sodium hydroxide on the solubility equilibria, which would actually be in the direction of increasing the number of concentrations, no deposition of sodium sulphate or sodium phosphate from the boiler water would be possible until the concentration of sodium hydroxide had likewise increased 185 times. If the initial concentration were 60 ppm of sodium hydroxide, the water just about to lay down solid sodium sulphate and sodium phosphate at the evaporative surface would necessarily contain 11,000 ppm of sodium hydroxide or a 1.1 per cent solution of NaOH. Thus it is apparent that in establishing the water conditioning for such high-pressure boilers, it is necessary to think in terms of the concentrated films, and not merely of the over-all boiler-water analysis, such as represented by the analysis of a representative sample taken from the boiler.

In so far as the over-all boiler water was concerned we had protective conditions. But the change which could occur on concentrating the boiler water from 60 ppm NaOH to 11,000 ppm NaOH is well illustrated by the curve in Fig. 1, originally developed by Partridge and Hall[6] from the experimental data of Berl and van Taack.[7] With 60 ppm caustic soda, the concentration is in the region of the most protective-to-iron zone. At 11,000 ppm NaOH (275 epm), the rate of attack found by Berl and van Taack on steel powder at 590 F was as great as in an acid solution containing about 25 ppm, HCl (0.7 epm).

The 11,000-ppm caustic-soda solution may seem fancifully high and difficult to attain. Actually, this is not the case. Dissolved substances lower the vapor pressure of a solution at any definite temperature. Correspondingly, under an externally applied temperature increase, the concentration of dissolved solids in a film of boiler water must necessarily increase to maintain constant pressure. As the terms Δt and Δts will be used frequently in this paper, a graphic representation of the meaning of the terms is given in Fig. 2 which consists simply of the vapor pressure-temperature curves of water and a saturated solution of potassium chloride. The Δt is the difference between the temperature of the concentrating film at the heat-transfer surface and the boiling temperature of pure water at the same pressure. The Δts is the Δt which corresponds to the existence of a saturated solution. If Δt is increased beyond this point, a solo solution will evaporate to dryness. Thus from Fig. 2, the film temperature can exceed the over-all boiler-water temperature at 1950 psia by about 140 deg F before potassium chloride will go to dryness in solo solution. At lower film temperature or Δt

[1] Hall Laboratories, Inc., Pittsburgh, Pa.

[2] "A New Approach to the Problem of Conditioning Water for Steam Generation," by R. E. Hall, Trans. A.S.M.E., vol. 66, 1944, pp. 457–488.

[3] "Experience With Sodium and Potassium Chemicals for Boiler-Water Conditioning at Montaup Electric," by G. U. Parks, Trans. A.S.M.E., vol. 67, 1945, pp. 335–338.

[4] "Operating History and Performance of 2000-Psi Forced-Circulation Boiler at Somerset Station of Montaup Electric Company," by G. U. Parks, W. S. Patterson, and W. F. Ryan, published on pp. 411–427 of this issue of the Transactions.

[5] "Solubility Equilibria of Sodium Sulphate at Temperatures From 150–350°C. III Effect of Sodium Hydroxide and Sodium Phosphate," by W. C. Schroeder, A. A. Berk, and Alton Gabriel, *Journal of the American Chemical Society*, vol. 59, 1937, pp. 1783–1790.

Contributed by the Power, Industrial Instruments and Regulators Divisions, and the Joint Research Committee on Boiler Feedwater Studies and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[6] "Attack on Steel in High-Capacity Boilers as a Result of Over-Heating Due to Steam Blanketing," by E. P. Partridge and R. E. Hall, Trans. A.S.M.E., vol. 61, 1939, pp. 597–622.

[7] "The Action of Caustic and Salts on Steel at High Pressures," by E. Berl and F. van Taack, *Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, V.D.I. Verlag, Berlin, Germany, No. 330, 1930.

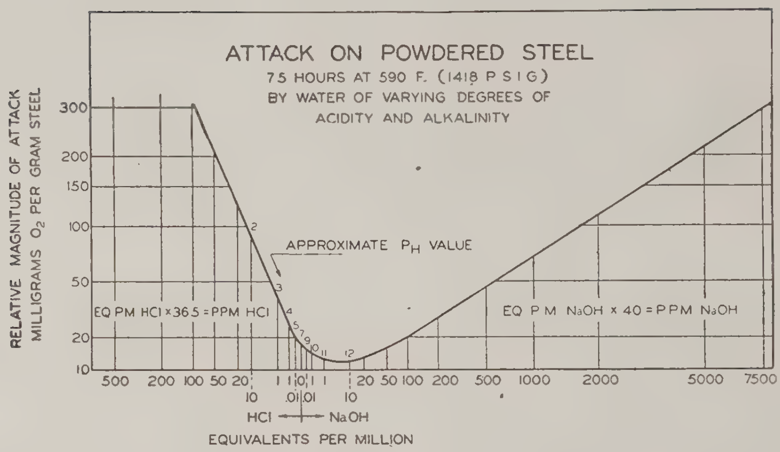

Fig. 1 Relative Attack of Steel at 590 F by Hydrochloric Acid and Sodium Hydroxide
(From laboratory data of Berl and van Taack.)

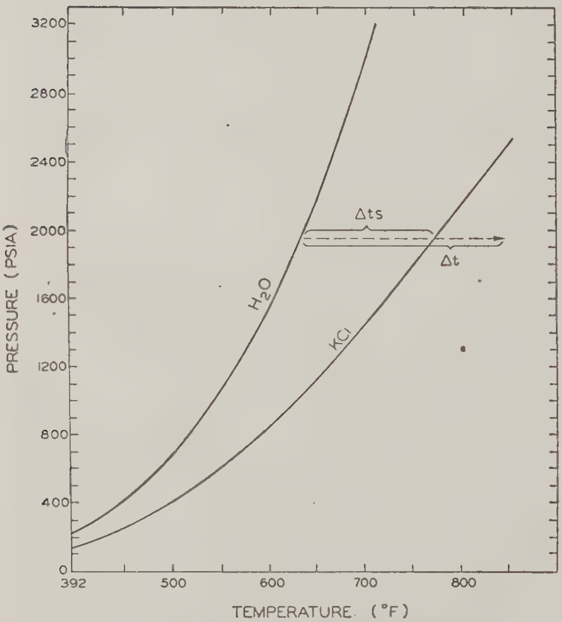

Fig. 2 Pressure-Temperature Relationships

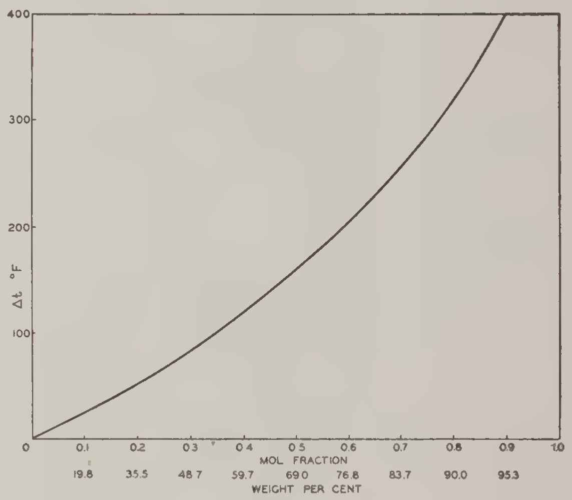

Fig. 3 Caustic-Soda Relation of Concentration in Solo Solution to Δt

differential, the film boiler water will simply concentrate to keep its vapor pressure the same as that of the over-all boiler water. If the film temperature or Δt exceeds 140 F the water will evaporate from the solution leaving a dry residue of potassium chloride.

Fig. 3 shows the caustic-soda concentrations produced in solo solution by various Δt values. How much Δt would be required to produce the 11,000 ppm NaOH concentration? The answer is only a little more than 1 deg F. With a Δt of 5 deg F the film concentration would be about 45,000 ppm (1125 epm) NaOH, with corrosive action (from Fig. 1) at relatively the same rate as an acid solution of about 300 ppm (8 epm) HCl.

That such small or even larger increases in film temperatures can result in a high-pressure boiler is hardly to be denied. Fortunately, for the most part such concentrated films are not continuous, but when they are, failure soon results. However, even with relatively good circulation such small increases in film temperature, or Δt, and concentrated solutions if soluble solids are present, can occur for at least short periods of time. Examples of such periods would be occasions when the slag falls off of a section of waterwall, exposing the tubes to higher temperature than the surrounding area which is insulated by the slag covering, or in certain tube sections immediately following soot blowing.

Obviously, in view of such factors, it becomes the duty of the water-conditioning engineer to think in terms not only of protective conditions for the over-all normal concentration of boiler water, but also in terms of maintaining the concentrating film as protective as possible.

Fortunately, by the time the need for improving boiler-water conditions at Montaup became apparent, considerable of the work developed in the paper by Hall[2] had already been made available. Therefore the boiler was cleaned of existing deposits by means of inhibited acid (Dowell) and a different treatment instituted.

Preliminary Potassium-Treating Period

The studies developed by Hall[2] indicated that while sodium silicate would "hide-out" and be scale-forming, the same was not true for potassium silicate as long as the silica-to-alkali ratio, in equivalents, was maintained at 1.6 or less, so that the highly soluble potassium metasilicate or disilicate would exist, even though considerable concentration due to film-boiling might occur. This made the maintenance of some silica in the boiler water, if it could be kept in the soluble form, desirable as a means of decreasing the amount of sodium-hydroxide concentration which could develop in a concentrating film. Furthermore, considerable magnesium phosphate had been found in various boiler deposits during the period of sodium treatment, and the maintenance of some silica in the boiler water would be advantageous for precipitating the magnesium as magnesium silicate, which has been proved to be the most desirable form of magnesium sludge in boiler water, with minimum tendency for the formation of adherent deposits. To obtain the advantage of these developments the boiler-water-conditioning control, started after the acid cleaning in April, 1943, was established as follows:

pH value	10.2–10.7
Phosphate (PO_4), ppm	30–50
Sulphite (SO_3), ppm	2–5
Silica (SiO_2), ppm	7–10
K to Na ratio, equivalents (min)	2:1
SiO_2 to alkalinity ratio, equivalents (max)	1.2:1

Inasmuch as the plant uses tidewater for cooling purposes, so that even small amounts of condenser leakage introduced considerable sodium in the boiler water, it was necessary to feed potassium chloride, in addition to the potassium phosphate, sulphite, and silicate, to maintain the desired potassium to sodium ratio.

A number of tests which were conducted after the boiler was returned to service with the use of potassium chemicals showed quite promising results. One of the most interesting features was the virtual elimination of sulphate and phosphate hide-out. The results of typical hide-out tests during the periods of sodium and potassium treatment, respectively, are shown in Fig. 4. The upper curves show the chloride concentrations as the load on the boiler was decreased. The lack of hide-out with potassium-treated boiler water contrasts sharply with the large amount of hide-out indicated by the sodium sulphate and sodium phosphate, obtained during the period of sodium treatment.

Several boiler inspections after relatively short runs, following the use of potassium chemicals, also showed promising results inasmuch as the boiler internals proved to be quite clean. Therefore the boiler was operated at full design pressure and the maximum load desired by the plant, generally between 550,000 and 600,000 lb of steam per hour.

A series of special tests were also made during this period which produced a number of interesting results. For example, it was found that in this boiler high-silica concentrations were unstable, and it was therefore inadvisable to try maintaining more than 10 ppm of silica in solution in the boiler water.

The work of Hitchens and Purssell[8] indicated that very little decomposition or auto-oxidation of sodium sulphite occurred up to a steam pressure of 1775 psi. However, in the Montaup boiler, considerable decomposition of sodium sulphite had been indicated whenever efforts were made to maintain more than a few ppm thereof in the boiler water. In this respect the potassium sulphite proved similar, and therefore the decision was made to maintain only a few ppm of sulphite in the boiler water, to serve primarily as an indicator of whether or not dissolved oxygen was entering the boiler with the feedwater.

[8] "The Behaviour of Sodium Sulphite in High-Pressure Steam Boilers," by R. M. Hitchens and J. W. Purssell, Jr., Trans. A.S.M.E., vol. 60, 1938, pp. 469–473.

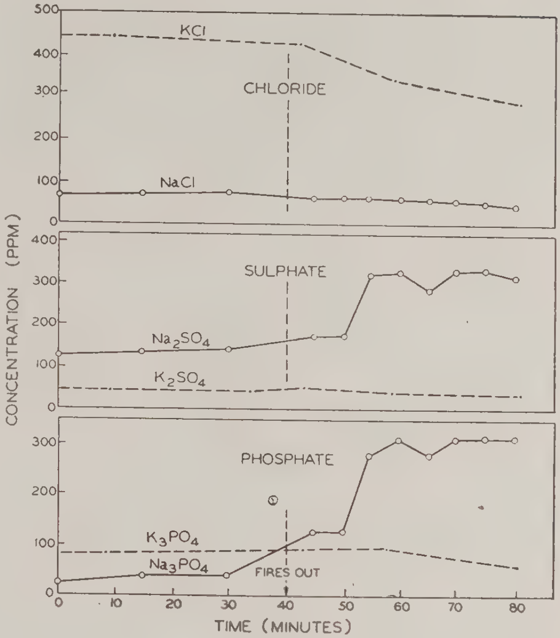

Fig. 4 Presence and Absence of Hide-Out With Na and K Salts

In December, 1943, No. 2 roof tube failed, apparently due to sludge accumulation in the rear-wall header around the strainer supplying the orifice which fed this particular tube. A number of tube sections were cut out at this time and considerable iron-oxide formation was found on the tubes, particularly on the fire side. Pickling some of the tube sections also disclosed a thin yellow ribbon of acmite, sodium iron silicate scale, on the fire side of a number of the tubes. Details of this are given in the reference paper by Parks[3] and need not be repeated here. It is perhaps noteworthy that during the period of potassium and silicate treatment the boiler had been operating at full design pressure and had generated considerably more steam with but one tube failure than it had during the period of sodium treatment during which three tube failures had occurred.

The boiler was again acid-washed by Dowell, using inhibited hydrochloric acid with some bifluoride added in order to remove the silicate scale as well as the iron-oxide formation. In order to eliminate further sludge accumulations at the ends of the rear header, arrangements were also made to interconnect each end of the rear-wall header with the adjacent side-wall header.

Potassium Treatment in Use Since January 1, 1944

In putting the boiler back on the line, it was decided to discontinue the feed of silica because of the slight but definite acmite formation which had been produced on the fire sides of a number of the tubes. It was also deemed advisable to continue with potassium boiler-water treatment because of the lack of hide-out when potassium chemicals were used in the boiler-water conditioning. Furthermore, a number of other plants which had substituted potassium chemicals in place of sodium chemicals for high-pressure-boiler water conditioning, had also found considerably less phosphate sludge in the boilers following this change

even though silica feed had not been used. This factor was also an influence in the decision to continue with potassium treatment.

Our laboratory has been doing considerable work with various solo solutions and mixtures of various salts in high-temperature bombs to determine the vapor pressures of the solutions at various temperatures and pressures. The results of a series of such tests are plotted in Fig. 5, from the data of Kaufman,

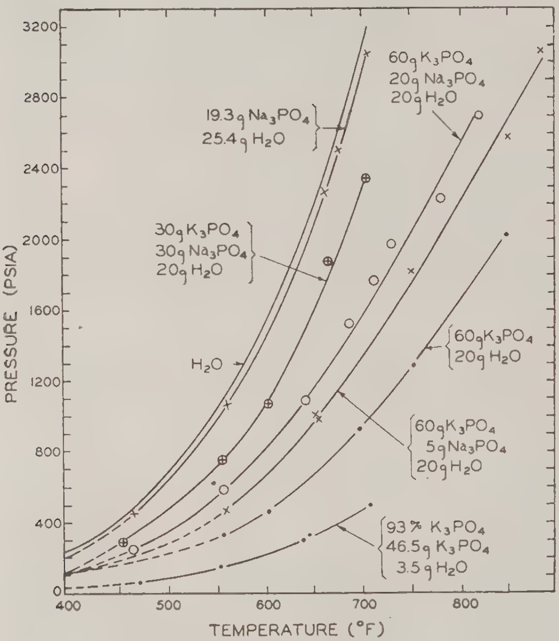

Fig. 5 Vapor Pressure of Sodium Phosphate, Potassium Phosphate, and Mixed Phosphate Solutions

Marcy, and Trautman.[9] These curves show why hide-out is readily obtained with sodium phosphate and not with potassium phosphate. For example, the Δt for sodium phosphate is only approximately 5 to 9 deg F for the entire temperature range from 400 to 3200 psia. Thus a film temperature of only 5 to 9 deg F above that of the main body of boiler water would be necessary to produce Δts, or a saturated solution. If the film temperature exceeds this, the solution will go to dryness, or, in this case, result in hide-out of sodium phosphate. With solo solutions of potassium phosphate, a Δt of several hundred degrees would be required before saturation and hide-out could occur. Mixtures of sodium and potassium phosphates show intermediate behavior, with an increasing Δts as the ratio of potassium to sodium is increased.

Fig. 6 shows the vapor pressures for a number of solo solutions compared to those of water at various temperatures. It will be noted from Fig. 6 that many of the sodium salts such as sodium phosphate, sodium sulphate, and sodium silicate have very low Δts values, or would go to dryness at a film boiler temperature of 10 deg or less in excess of the temperature of the saturated steam. A considerably larger Δts value is indicated for sodium chloride and even greater values are obtained for potassium chloride. As mentioned in connection with Fig. 5, the Δts of potassium phosphate is so large that it would seem almost impossible to obtain a film temperature sufficiently high to precipitate potassium phosphate out of solution unless actual overheating of a tube occurs to a considerable degree. With either sodium or potassium hydroxides, concentration of the solution occurs without going to dryness. In other words, in a boiler, or in a water-hydroxide system, Δts is practically nonexistent for sodium or potassium hydroxides.

As pointed out in connection with Fig. 1, concentration of hydroxide, such as obtained by only a few degrees elevation in temperature due to film boiling, can produce conditions decidedly corrosive to boiler metal. Therefore in thinking in terms of the concentrated solution which can develop, it is desirable to introduce other salts of sufficient Δts to remain in solution in case of film boiling, and thereby reduce the concentration of caustic which can result. In order to accomplish this, the joint means were used of decreasing the range of alkali to be maintained in the boiler water and increasing the potassium chloride concentration. The effectiveness of this was checked in the laboratory by means of accelerated tests. Bombs were charged with 250 g of solids (anhydrous) and 30 g of water. The temperature within the bomb was raised to approximately 1022 F in a period of 3 hr, after which the heating was discontinued and the closed bomb allowed to cool to room temperature. The residual pressure due to the hydrogen generated during this test was measured after cooling. The results obtained for mixtures of potassium chloride and potassium hydroxide, potassium phosphate and potassium hydroxide, and mixtures of both potassium chloride and potassium phosphate with potassium hydroxide are plotted in Fig. 7. It will be noted from this figure that the amount of hydrogen formed, which is a direct indication of the amount of attack on iron, dropped very rapidly as the molar ratio decreases below 0.2 for mols KOH over the total mols of chemicals present. Thus for benefit to be obtained from this source it is indicated that the chloride concentration, in equivalents, should be at least 5 times that of the hydroxide concentration, and for added safety in this respect a ratio of 10 to 1 chloride to alkalinity was

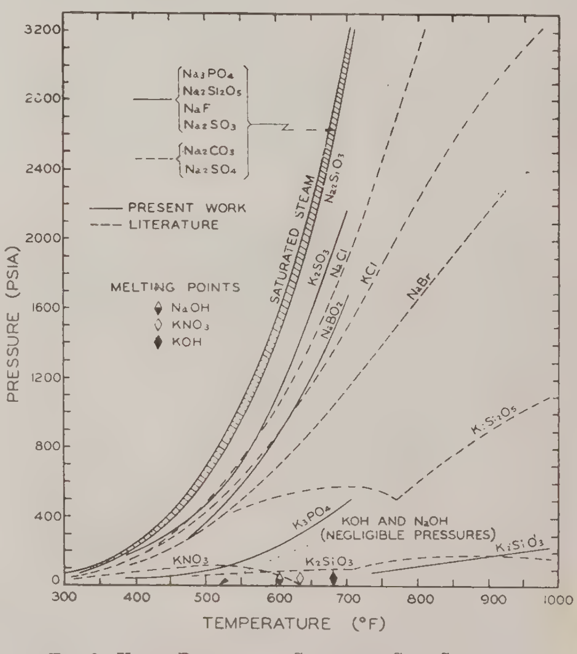

Fig. 6 Vapor Pressure of Saturated Solo Solutions

[9] "The Behaviour of Highly Concentrated Boiler Water," by C. E. Kaufman, V. M. Marcy, and W. H. Trautman, Proceedings of the Sixth Annual Water Conference, Engineers Society of Western Pennsylvania, October, 1945.

established as a minimum for the chemical control of the Montaup boiler water.

The detailed control limits which were established and have been maintained since January 1, 1944, are as follows:

pH value	9.8–10.4
Phenolphthalein alkalinity (ml, N/30 acid required for 100-ml sample)	0.5–1.6
Phosphate (PO_4), ppm	8–20
Chloride: hydroxide ratio, epm	Minimum = 10:1
Potassium: sodium ratio, epm	Minimum = 3:1
Total solids, ppm	Maximum = 1000

Actually, the control has worked out so that the chloride to hydroxide ratio, in equivalents, is seldom below 50:1. Thus in a film-boiling solution with Δt of 5 deg F, instead of obtaining a caustic concentration of 4.5 per cent, this concentration will be only about 0.09 per cent (900 ppm or 22.5 epm), or still within

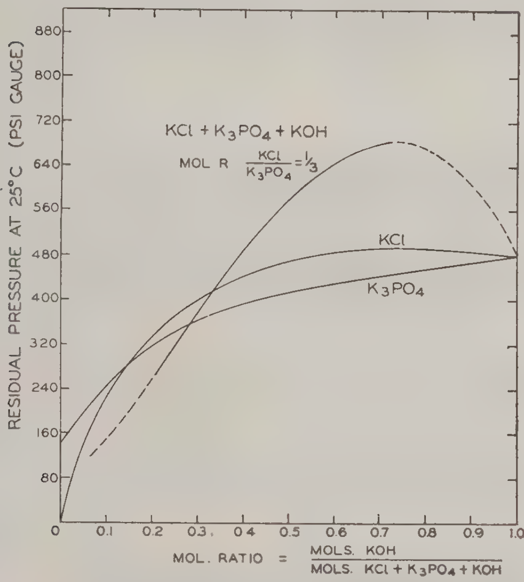

Fig. 7 Residual Hydrogen Pressure Developed by Action of KCl-K_3PO_4-KOH Mixtures

the protective range of minimum attack by hydroxide on steel at high-pressure-boiler temperatures, as indicated by the data of Berl and van Taack plotted in Fig. 1.

High-Pressure-Boiler Feedwater

The nature of the boiler feedwater is also a factor which cannot be ignored, especially in the operation of high-pressure boilers.

The boiler feedwater at Montaup is obtained from the turbine condensers plus a small amount of evaporated make-up water. Control to maintain good feedwater conditions is normally maintained at the plant, as proved by some 15 years' operation of the 400-psi boilers without difficulty in the preboiler equipment. However, with the advent of the early difficulties encountered in the operation of the high-pressure boiler, steps were taken further to improve the quality of the feedwater wherever possible. Operation of the two deaerators was adjusted so that they would always be under positive pressure to assure good elimination of dissolved oxygen. The turbine condensers were also retubed in 1944, to assure minimum contamination by the tidewater used for condenser cooling.

These steps have resulted in feedwater which generally shows zero values for dissolved oxygen. Carbon dioxide in the system is also very low, due to the feedwater to the evaporators being a sulphate rather than a carbonate water. Tests made on the steam indicate that the carbon-dioxide content is of the order of 0.01 ppm. Measurement by means of a Leeds & Northrup pH meter on flowing cooled samples, right at the sampling points after the deaerators, generally shows pH values of about 8.5.

Condenser leakage is immediately indicated on the condensate-conductivity recorders, and if this becomes at all high, the condenser is taken out of service as soon as possible and the leaking tubes located and plugged. As a result of this control the turbine condensate used for feedwater has averaged less than 5 micromhos per cm conductivity, since the summer of 1944.

Examination of Boiler-Tube Sections

The removal of tube sections for examination rather than merely examination and analysis of such deposits as are found in the wet-steam drum, has greatly facilitated the determination of internal conditions of the Montaup boiler. We have thus been able to study the distribution of constituents of the deposit between the fire and cold sides of the tubes, as well as between the surface of the deposit and the area next to the metal. Examples of this variation are the presence of acmite ($Na_2O.Fe_2O_3.4SiO_2$) on the fire side of some tubes in a thin layer next to the metal. In some instances this layer was so thin that it would never have been detected had the tube section not been pickled. The bulk of the deposits was iron oxide and contained little or no acmite. Another example is the presence of a considerably higher percentage of phosphate sludge on the cold side of some tubes rather than on the fire side. A third example of variation is found in the appearance of the internal surfaces of some tubes. Many of these specimens examined showed smooth continuous films of magnetic iron oxide, while others exhibited considerable roughness and thicker surface deposit. Pickling of the specimens showed that those tubes which exhibited a rough surface very frequently showed evidence of metal attack.

The first set of specimens examined by us was removed from the boiler on January 13, 1943, at the time of failure in the No. 2 roof tube. The general condition of these tubes has already been reported in the paper by Parks.[3] Also in this paper were presented for comparison photographs of tubes examined after the boiler had been on potassium-silicate treatment. In both of these sets of tubes a hard dense deposit, consisting chiefly of magnetic iron oxide, was found on the fire side of the specimens examined. This was, at least in the specimens removed after potassium-silicate treatment, found to overlie a very thin film of acmite ($Na_2O.Fe_2O_3.4SiO_2$). The specimens removed after sodium treatment were not pickled to determine if a layer of acmite adjacent to the metal was present. The cold side of the tubes, while showing a relatively rough surface, did not exhibit the hard, dense iron oxide found on the fire side in either of these sets of specimens. The material on the cold side of the tube was soft and powdery, as was the surface layer on the fire side which covered the hard, dense material.

Appreciable roughening of the metal surface of a number of the tubes was evident. This roughening was believed due in part, at least, to small tears produced in the tubes during cold-drawing. A specimen of this sort of tubing, carefully polished to remove the tears before installation, did not show as much attack on the metal surface although this tube section was not in operation as long as the unpolished tube. It is not known whether acid-cleaning of the boiler contributed to this roughening.

The first set of specimens removed after the discontinuance of silica feed still showed some patches of the dense iron oxide, which patches were coated with metallic copper. Since these specimens had been in the boiler during silica feed and the acid-cleaning which was done following the silica treatment, it seems quite

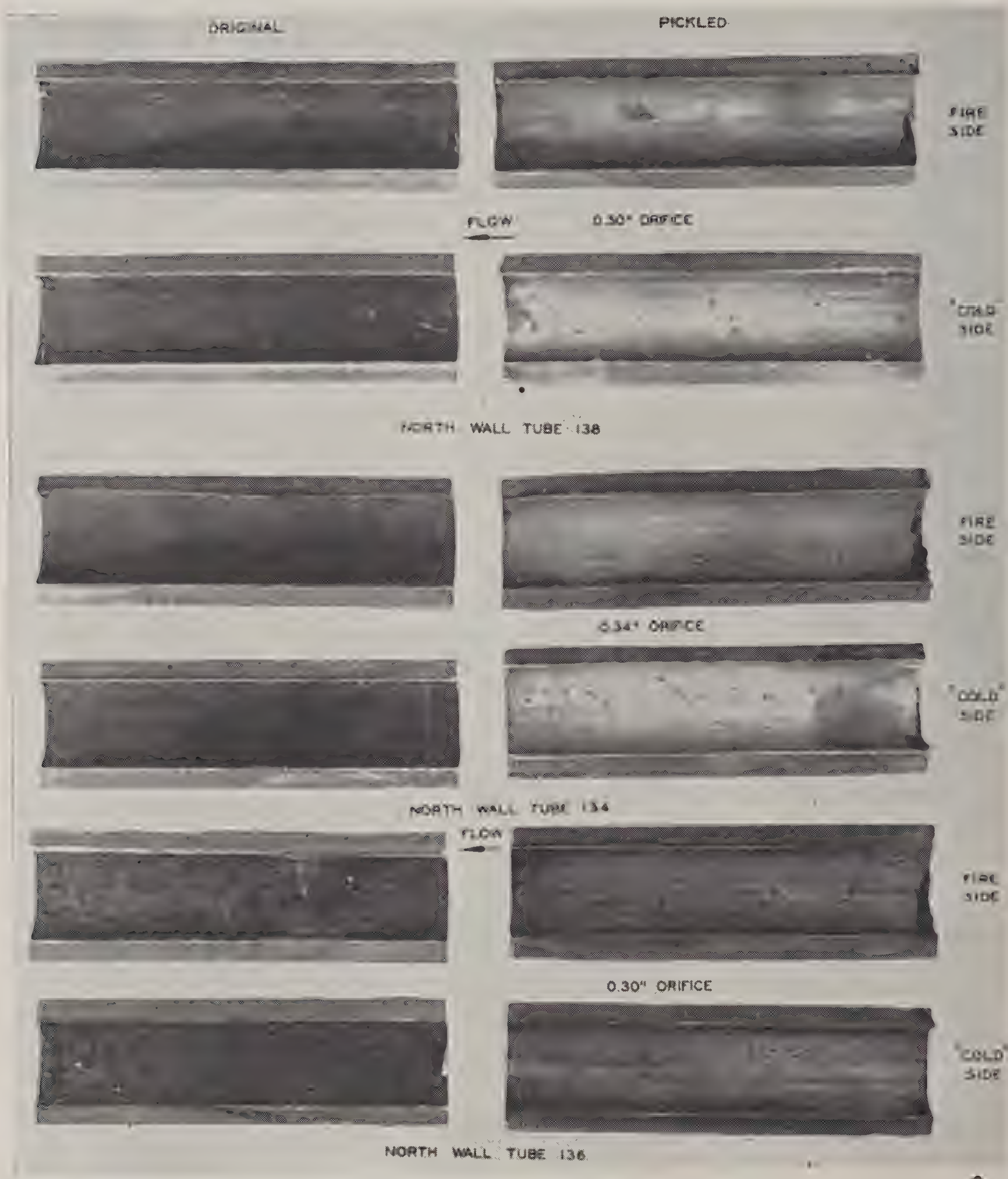

Fig. 8 Appearance of Boiler Tubes From Bifurcated Sections After 14 Months of Operation With Present Potassium Treatment

(Each tube is one of a pair of bifurcated tubes fed by a single orifice. Tube No. 138 is fed from a 0.50-in-diam orifice; tube No. 134 is fed from a 0.34-in-diam, or normal size orifice; tube No. 136 is fed from a 0.30-in-diam orifice.)

possible that the deposition of copper and even part of the pitting might have occurred during the acid-cleaning. Tube sections removed up to September, 1945, showed that none of the tubes which have been installed since the December, 1943, acid-cleaning contained any significant amount of this hard dense iron oxide. One or two localized patches of scale had developed, but this is believed due to restricted circulation from welding slag in the particular tube orifice, as will be explained later.

Since the surfaces of the tubes originally installed in the boiler were found quite rough, particularly on the fire side, it was decided, in order really to evaluate the present method of treatment, to examine only specimens which have been installed since the December, 1943, acid-cleaning. Special orifices were installed in the strainers feeding six of the north-wall tubes. A fairly large section of these tubes, plus an adjacent pair, fed by the normal orifice, was replaced in June, 1944, with new tubing and we were therefore able to determine the general effect of treatment and also the effect of the special orifices on the same set of samples.

The specimens from this section examined during the latter part of 1944, contained considerably more calcium and magnesium phosphate than had previously been seen on the tubes since the institution of potassium treatment, although there was no evidence of marked attack on the tubes such as had been experienced with the tubes examined previously. It was found that much of the sludge and iron oxide could be removed from the boiler by operating for a 24-hr period at 400 psi and high blowdown. Until the October, 1945, failure of two rear-wall tubes, this treatment of the boiler indicated that accumulation of deposit was insignificant.

Prior to the use of tubes installed in 1944, for examination, it was almost impossible to tell from the condition of the metal surface or from the amount of sludge accumulation just what the extent of attack really was. This was because the amount of metal loss and consequent roughening of the surface seemed to be a function of the number of tears occurring in the metal surface during cold-drawing, as well as operating conditions. The accumulation of iron oxide and also the precipitation of phosphate sludge appeared to be proportional to the amount of roughening of the metal surface. Of particular interest along this line was the fact that the phosphate sludge tended to pile up on the downstream side of a high spot in the iron-oxide surface, especially behind a tubercle on the cold side of the tube. We are at a loss to explain why there should be more phosphate on the cold side, unless the increased turbulence on the fire side of the tube prevented it from settling out. The phenomenon has, however, been observed too frequently to be pure coincidence.

Fig. 8 shows portions of tubes removed from the north wall after approximately 15 months of the present potassium treatment. Tube No. 134 is one of a pair of tubes being fed through a normal 0.34-in. orifice. It will be noted that the surface of the

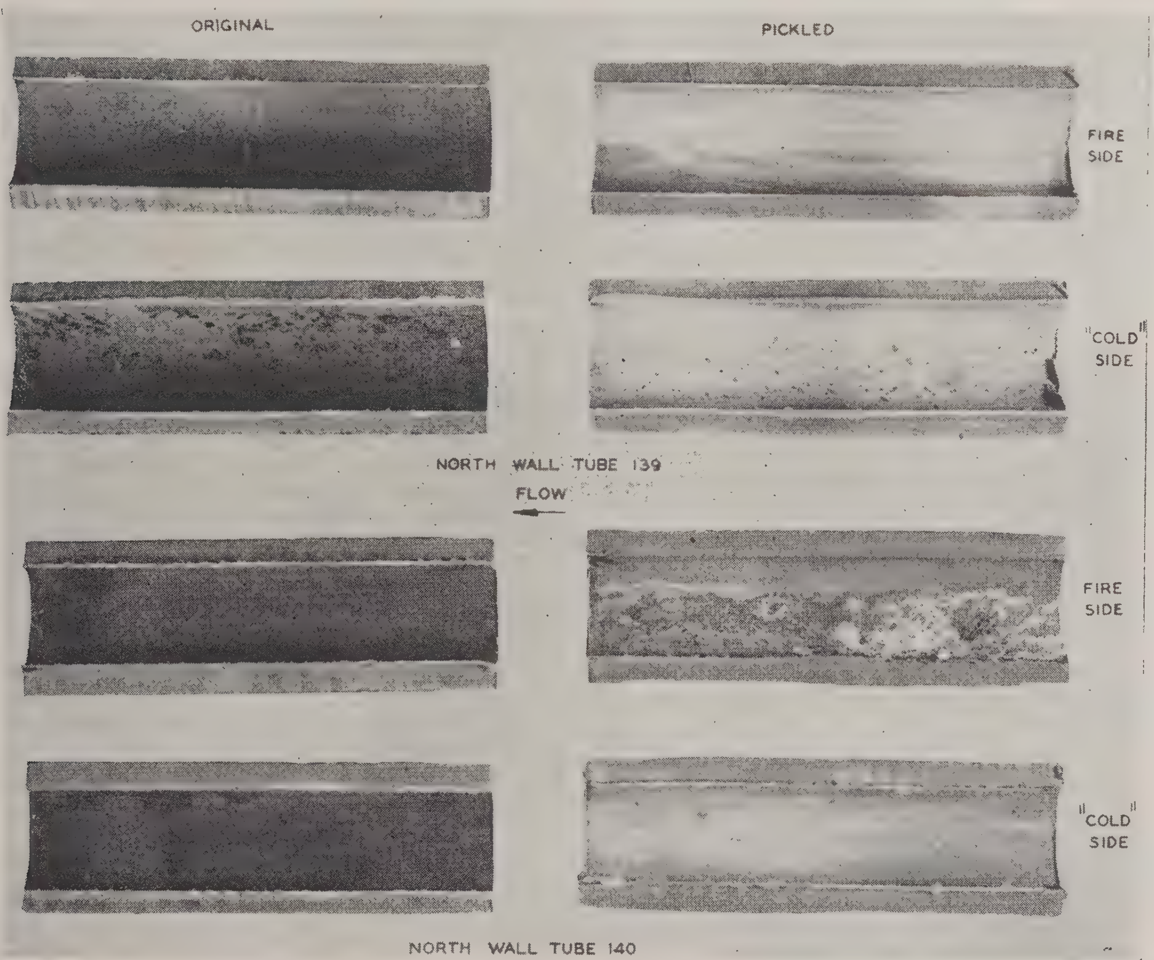

Fig. 9 Appearance of Boiler Tubes Fed by Single, Separate, 0.25-In-Diam Orifices After 14 Months of Operation With Present Potassium Treatment

oxide film is fairly smooth and that there is no evidence of significant attack on the metal in the pickled specimen, either on the fire or cold side. Tube No. 138 is one of a pair of tubes being fed by a special 0.50-in. orifice and is similar in appearance to No. 134 tube with the normal orifice. Tube No. 136 is one of a pair of tubes being fed by a reduced-sized orifice, 0.30 in. In the original specimen from this tube the definite roughening of the surface of the deposit can be noted on both the fire and cold sides. In the pickled specimen there is a little more roughening of the surface on the fire side, Fig. 9.

Tubes Nos. 139 and 140, Fig. 9, normally a pair fed by a single 0.34-in. orifice, are fed by two separate 0.25-in. orifices. These two separate small orifices would transmit about the same total water flow as through a single 0.34-in. standard orifice. It will be noted that there is no more attack evident on the pickled specimens of tube No. 139 than in the normal tubes. The flaking off of deposits on the cold side of tube No. 139 is believed to be due purely to mechanical causes during shipment of the tube, since light tapping of the tube in the laboratory removed more of the deposit. The particular section of tube No. 140 shown was photographed because of the apparent difference in the nature of the deposit. This is the only hard patch on any of this set of tubes, and does not extend over the remainder of the specimens from this tube. The main constituent of this very thin, hard, scalelike material is still iron oxide, although there is considerably more hematite (Fe_2O_3) than magnetic iron oxide (Fe_3O_4) present. There is also some phosphate sludge, some form of silica which was not sufficiently crystalline or present in sufficient amount for identification by x ray, and a small amount of metallic copper. Traces of sodium and potassium salts were also found in the deposits, apparently trapped in the scale. The poor appearance of the fire side of tube No. 140 is not considered significant as this condition was probably produced as a result of the orifice restriction which caused the No. 139 tube failure in July, 1944, as discussed in the first paper of this series by Parks, Patterson, and Ryan.[4]

The freedom from sludge and general appearance of most of these tubes is encouraging. It should be remembered, however, that the amount of sludge or scale which would cause little if any difficulty in the average boiler would likely result in a large number of tube failures in a high-pressure boiler with high heat-absorption rates.

The two tubes which failed on October 26, 1945, and a number of other tubes as well, again showed considerable deposit formation. In view of the long period of operation with little in the way of deposits from January 1, 1944, to September 14, 1945, it appears that some unusual circumstance may have developed in the operation prior to October 26. On the other hand, this may simply indicate that inasmuch as only the tubes cut out can be inspected for internal conditions, it would be advisable to acid-clean the unit at regular intervals such as annual shutdowns, rather than to operate indefinitely without certain knowledge of boiler cleanliness.

Conclusions

Operation of the Montaup high-pressure boiler since April, 1943, when potassium treatment was first instituted, has been decidedly successful, with operation at full design pressure and generally near design load as required by the plant.

There have been only two forced boiler outages since April, 1943, associated with internal boiler deposit. This record for $2^1/_2$ years' operation can be equaled or bettered by very few boilers operating at 1950 psi or higher drum pressure, which is the operating pressure of the Montaup unit.

In arriving at the boiler-water control for high-pressure boilers, not only the over-all boiler-water conditions must be considered, but attention must also be focused on the type of water which will develop when abnormally high concentrations occur, such as follow with but a few degrees film-temperature increase above the over-all boiler water and saturated-steam temperatures.

It is felt that the advantages of potassium equilibrium in the boiler water have been definite contributing factors to the operating record of this boiler, similarly to a number of other boilers operating at pressures of 600 psi or above, where definite improvements in internal boiler conditions or maintenance of turbine capacities, or both, have resulted.

Discussion

T. J. Finnegan.[10] This paper describes the application of potassium treatment to the Somerset Station and presents curves which show that under that treatment hide-out did not occur, while under sodium treatment it was present. This is in agreement with our experience with three high-pressure natural-circulation boilers which operate at 1265 psia and 900 F at the turbine inlet and deliver 900,000 lb of steam per hr. In our experience, however, it has been found that hide-out of phosphate is absent only when the PO_4 content of the boiler water is kept at a low figure and consequently we do not allow it to exceed 10 ppm. There is probably a relation between boiler design and the amount of hide-out which will occur, and this must be considered along with the purely chemical features of the problem.

We can also support the statement regarding the decomposition of sulphite in the boiler water and can demonstrate that the condition which permits it to occur may be very delicately established and difficult to define. In one of two 900,000-lb per hr high-pressure boilers, which had been operated with about 15 ppm of excess SO_3, sufficient decomposition occurred so that hydrogen sulphide could be smelled easily in the steam, and copper sulphide deposits were found on the high-pressure heaters, while in the other boiler no evidence of sulphite decomposition was found. Both boilers were operated under similar conditions of loading.

Reports have been received of good results in the treatment of high magnesium boiler water with silicates in order to avoid magnesium-phosphate deposits and to precipitate magnesium as silicate sludge. The boilers which are given this treatment are generally operated at low pressure and with high make-up. It seems risky to extend this treatment to boilers which operate in the pressure range which has been associated with silica deposits in the turbine. Silicate treatment was discontinued in the Somerset boiler because a small amount of silica scale had been found but an equally important consideration would be the internal condition of the turbine.

In spite of the fact that the three high-pressure units mentioned operate with less than 1 per cent make-up, at one time severe silica deposits had been found in the turbine. At present the silica deposits are light enough so that they can generally be disregarded in scheduling the regular outages of these machines, and this has been effected by reducing the boiler-water silica to as low a value as it is possible to obtain; it is usually of the order of 2 to 3 ppm of silica.

Authors' Closure

Mr. Finnegan's comments in regard to actions observed with phosphate and sulphite contribute to the information obtained from operating results.

The extent to which phosphate hide-out will occur, or be eliminated, when using potassium chemicals, will be affected by

[10] Chemical Engineer, Buffalo, Niagara Electric Corporation, Buffalo, N. Y. Mem. A.S.M.E.

the ratio of potassium to sodium present, and the Δt as illustrated by Figs. 2 and 5 of the reference paper.

The use of silica, properly controlled, has been of decided benefit in reducing the amount of adherent sludge in many boilers. Successful experiences in this respect include boilers operating at as high as 750-psi drum pressure, with no increase in the turbine deposition. The precipitation of residual magnesium entering the boiler as the silicate, instead of the hydroxide or phosphate, produces the most favorable conditions for the prevention of adherent sludges.

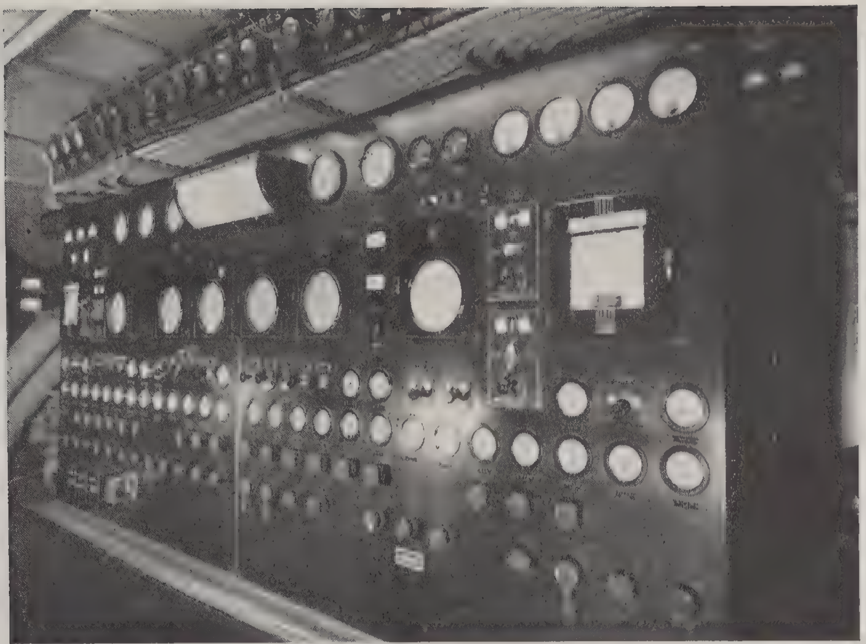

Fig. 1 Main Control Board 2000-Psi Boiler 6; Gr. 137

Experience With Instruments and Control Equipment for 2000-Psi Boiler at Somerset Station of Montaup Electric Company

By W. D. BISSELL[1] and E. B. POWELL[2]

This paper presents comprehensive descriptions and operating experiences with the instruments and control equipment as applied to the high-pressure high-capacity boiler of the Somerset Station. Details are given not only of the completely automatic operating and control equipment but also of the indicating and recording instruments which enable the operators to follow constantly the operating conditions of the boiler and thus permit immediate manual adjustment when the need is indicated.

The 2000-psi boiler at Somerset Station is numbered 6 from location and sequence of installation in the station and, for brevity, will be referred to as boiler 6 in this discussion. The instruments and automatic regulating devices which serve for operation control on the boiler are conveniently considered in three groups. In group (1) is classed equipment serving primarily for stabilization of output and efficiency of combustion; in group (2), equipment for maintenance of water conditions. For ready identification of equipment associated with the distinguishing characteristic of the boiler, a third grouping is employed for permanent instruments and controls installed to meet the special needs of forced circulation.

[1] Somerset Station, Montaup Electric Company, Fall River, Mass.

[2] Stone & Webster Engineering Corporation, Boston, Mass. Mem. A.S.M.E.

Contributed by the Power, Industrial Instruments and Regulators Divisions, and the Joint Research Committee on Boiler Feedwater Studies and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Control of Boiler Operation and Combustion Efficiency

The functions served by the instruments and control equipment of group (1), as classified in the preceding paragraph, are essentially those to be met in the operation of any modern boiler of equal capacity, i.e., pressure gages for steam, water, and air; flowmeters for steam, water, and fuel oil; feeder speed indicators for pulverized coal; instruments for evaluation of combustion efficiency; boiler-drum water-level recorder and drum water-level indicators; combustion and steam-pressure control; steam-temperature control; feedwater-pressure and flow-control devices. Interest in experience with this equipment is found chiefly in the instances where pioneering has been involved. Other equipment will be sketched merely for perspective.

Main Control Board. The main control board, mounting in-

struments and controls primarily of group (1), is shown in Fig. 1. The operating signal lights for the boiler circulating pumps at the extreme left near the top of the board are the only equipment of this figure not classed as of group (1). At right and left of the board are independently mounted primary-air pressure gages. Above the board are fuel-oil pressure gages and coal-stream temperature indicators. On the board proper the top row includes coal-feeder tachometers at the extreme left, and fan-motor ammeters at the extreme right, multipoint draft gages in the middle, with water- and steam-pressure gages. The row of recording instruments includes recorders for temperature of steam and boiler feedwater at the extreme left, and temperature of flue gases and air for combustion at the extreme right, with boiler-feedwater flow and pressure, and steam pressure, boiler-drum water level, main steam flow - air flow, reheater steam flow, fuel-oil pressure, and flue-gas oxygen in sequence in intermediate positions. At left and right of the feedwater flow recorder are mounted, respectively, controls for feedwater regulation and one of the drum water-level indicators. Similarly, at the sides of the flue-gas oxygen recorder are mounted, respectively, induced-draft-fan motor-exciter voltmeter and the steam-pressure reducing-valve controls. The lower half of the board is devoted almost exclusively to combustion and steam-pressure control, providing for manual control of the combustion system as a whole or for selective use of manual or automatic control as desired on the different elements of the system.

The flue gas-oxygen recorder is one of the highly informative instruments of the board. However, this instrument had barely passed the experimental development stage when purchased. As to be expected with novel equipment, a number of difficulties have been encountered from those frailties which it seems experience alone can fully bring to light. Experience has also proved the value of the instrument as a guide in control of combustion, giving more definite and precise indication of furnace conditions than the steam flow - air flow recorder which it serves to supplement. The flue-gas oxygen recorder is now to be extensively rebuilt.

Mounted on a hinged panel at the left of the recorder row are indicating pyrometers for main superheated-steam and reheated-steam temperatures. Mounted on a separate panel on the opposite side of the aisle, facing the main control board and not shown in the figure, is a group of instruments serving primarily to place control of the temperature of 375 psi superheated steam, delivered to the turbine throttle, in the hands of the high-pressure-boiler operator. This temperature control is effected by regulation of water supply to the desuperheater preceding the steam reheater normally, to the by-pass desuperheaters when the latter are in operation. The equipment of the panel includes two temperature indicators and recorders, one for each of the low-pressure-turbine throttles, an indicator for temperature of steam from the reheater, and water controls for the 4-in. by-pass desuperheater. Control valves for the desuperheating station ahead of the reheater, while not operated from the panel, are readily accessible at the rear of the boiler at the same grade. Controls for the 10-in. by-pass pressure-reducing and desuperheating station and for pressure reduction ahead of the 4-in. desuperheater are on the main board.

Variations in superheated-steam temperature on the low-pressure system are caused usually by variations in loading of low-pressure boilers which take up all load swings within the limits of their operating capacity. The throttle steam temperature on the low-pressure turbines must be held within close limits. Obviously, maintenance of mere constancy of reheated-steam temperature would not meet the requirements. Accordingly, manual adjustments are made in desuperheating the low-pressure steam prior to reheating, or in automatically controlled temperature of pressure-reduced steam, or in both, as required to give substantially constant temperature at the low-pressure-turbine throttles. The indicating pyrometers on reheated steam and the temperature indicators and recorders on steam at the throttles of the two low-pressure turbines give the operator clearly readable and closely concurrent values at these points at all times.

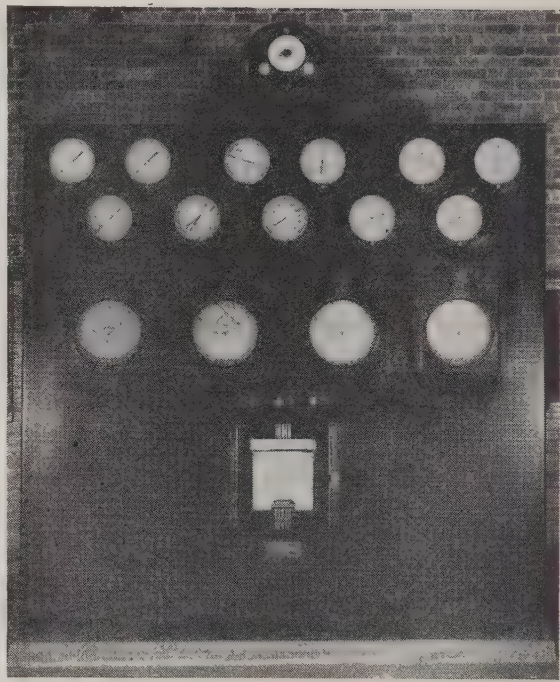

Fig. 2 Boiler-Feed-Pump Instrument Panel; Gr.137

Adjacent to the low-pressure-system steam-temperature control panel is a 12-point potentiometer indicator for checking recorders and for reading instantaneous temperatures. With multipoint recorders, several minutes may elapse between temperature measurements for a particular point. The indicators bridge the gap and give values instantly or continuously as desired.

Boiler-Feed-Pump Instruments. Fig. 2 shows the boiler-feed-pump instrument panel on the main operating floor, Gr. 137, on which are mounted a recorder for steam flow to turbine-driven feed pumps and for high-pressure-boiler feedwater flow and pressure, and a recording pyrometer for feedwater temperature at several stages of heating, as well as indicating gages for individual pump suction and discharge pressures, pump-turbine-drive exhaust and extraction pressures, and pressure differential between boiler drum and main feedwater header. Additional feedwater-system pressures and temperatures are indicated on the gages of the feedwater-heater instrument panel shown in Fig. 3. On this panel are also mounted controls for motor-operated feed-line valves at the high-pressure heaters and alarms for water levels in the shells of closed-type heaters. This panel is located on the ground floor, Gr. 118.

Regulation of Boiler-Feedwater Flow and Feed-Line Pressure. The feedwater flow recorder on the main board records the total of the separate flows in the two feedwater lines to the boiler economizer. The feedwater-regulator controls mounted adjacent to this recorder provide for adjustment of water-level position in the drum, for changes from automatic control from the corresponding drum-end thermostat to manual control, and

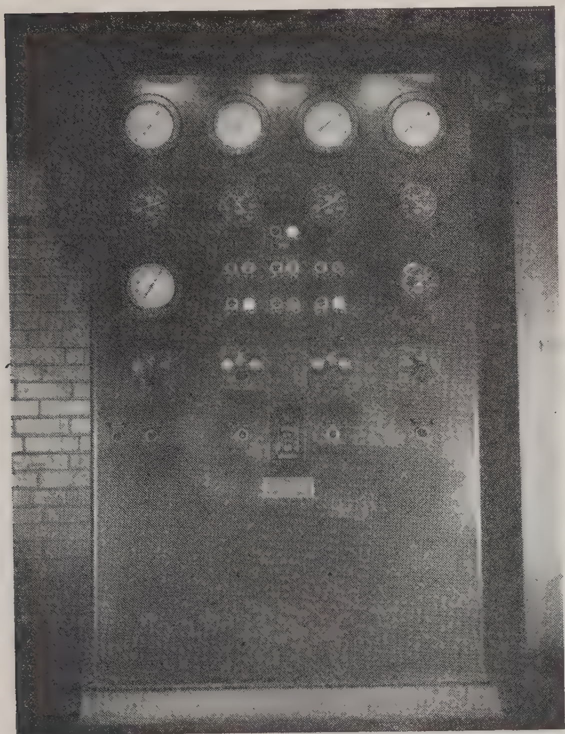

Fig. 3 Feedwater-Heater Instrument Panel; Gr. 118

vice versa on either or both feed-line flow regulators, and for the manual control of feed-line flow through either or both regulators. Valve position indicators immediately above the controls show at all times the degree of opening of each regulator. Separate line flow indicators, which are of value in equalizing regulator operation, are mounted adjacent to the respective individual metering orifices. The master controller, which functions to maintain the required head differential on the feedwater regulating valves and a definite pressure excess in the feedwater line for sealing the glands of the boiler circulating pumps, is mounted at the rear of the boiler and, as its adjustments are of relative permanence, is not arranged for control from the main board.

Transmission of the primary impulses from the drum end thermostats to the receiving diaphragms on the feedwater regulators, and from the feed-line excess-pressure master controller to the receiving diaphragms for adjusting the speed settings of the boiler feed pump turbine governors, has been accomplished eminently satisfactorily with compressed air. For the actual motivating fluid on the feedwater-regulator pistons, water was found during preliminary operation to give a smoother and more precise control and, accordingly, was adopted. On failure of the motivating water supply, the feed-regulating valve is automatically locked in position. Shifts back and forth between automatic and manual controls are effected easily and smoothly and, since the change in motivating fluid on the water-flow-regulator pistons, the functioning of the system has creditably met all demands, positively controlling the water level in the drum over the full capacity range of feed-line flow down to the condition of complete feedwater supply through circulating-pump gland leakage. Maintenance, aside from wear on the feed-pump low-load flow-recirculating valves, has called for little more than routine of inspection, cleaning, and lubrication of parts.

The feed-line pressure-control system provides not merely for high-pressure feed-pump speed adjustment but also for automatic spilling of the high-pressure feed-pump-drive turbine exhaust to the 15-psig exhaust system, and to atmosphere if required in the emergency of failure of one of the normally two such pumps in operation.[3] As there has been no high-pressure boiler-feed pump service failure, this last provision has not been called into operation.

The automatic control of recirculation of feedwater, required at low loads on the high-pressure boiler-feed pumps,[3] is also one of the functions of the high-pressure boiler-feed regulating system. The recirculating valve is operated by air impulse from the feed-pump-governor speed-setting control. As commented above, wear on the valve parts from the 2150-psi to 500-psi pressure reduction taken in the valve has been about the only source of significant replacement called for in maintenance of the high-pressure-feed regulating system. It is now believed that the life of these parts can be increased by change in design.

Combustion Control. Combustion control is of course initiated by change in steam pressure, with auxiliary adjustments from the accompanying steam flow and gas flow. The measurements of steam pressure and steam flow are usually direct and accomplished with adequate precision without regard to boiler design. On the other hand, for the modern boiler with heating surface almost exclusively of the radiant type, especially where the superheater is under gas by-pass control as in boiler 6, inherent sources of draft loss appropriate for the measurement of combustion-gas flow or combustion air flow become not so obvious. Control equipment and certain instruments used for giving a running check on control efficiency are dependent upon the precision of this measurement. The problem of providing the essentials for this measurement adequately and economically may well receive the careful attention of both boiler manufacturer and plant designer.

Before going into the experiences in measurement of combustion-gas flow on boiler 6, it should be pointed out that superheated-steam temperature is controlled by diverting through the upper economizer a part of the combustion gas flow which would otherwise make full travel of the convection bank of the superheater. The diversion of gas flow is accomplished by adjustment of main and by-pass dampers at the gas outlet of the economizer, shown diagrammatically in Fig. 4. Steam temperature is held very closely, swings in temperature rarely exceeding 5 F even during operation of soot blowers.

As first arranged, air flow for combustion was evaluated by summation of straight-line functions of gas pressure drops across the two parts of the economizer, utilizing connections at points about as indicated by C and D and E and F in Fig. 4. For recording steam flow - air flow relations, the combustion-gas flow initially was to be based upon the differential across the steam reheater between points indicated by A and B, also in Fig. 4. It early became evident, however, that the normal gas resistance of the reheater was not great enough to give consistent measurement of gas flow, and connections were made to provide for summation of straight-line functions of economizer-draft drops, paralleling the corresponding arrangements for combustion control. For metering purposes, however, an effort was made to compensate for variations in gas density by making the two pressure connections on the flue at the same level and extending the trailing pressure lead to the outlet of the economizer within the gas stream.

Attempts toward close co-ordination in combustion control en-

[3] "1825-Lb-Pressure Topping Unit With Special Reference to Forced-Circulation Boiler," by F. S. Clark, F. H. Rosencrants, and W. H. Armacost, Trans. A.S.M.E., vol. 65, 1943, pp. 461–477.

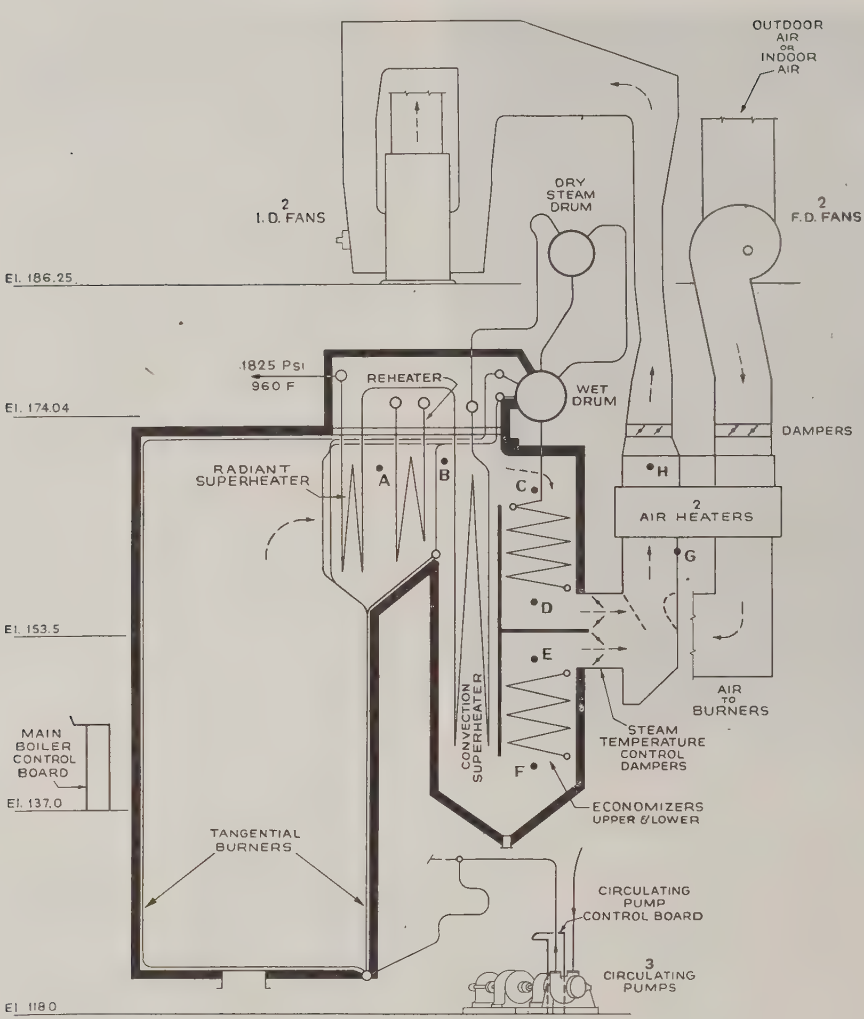

Fig. 4 Diagrammatic Section Boiler 6

countered the situation that pressures measured at the downstream connections in the economizer casings were seriously influenced by flow disturbances varying with the positions of main and by-pass dampers. In addition, during the first few months of operation, apparently the economizer gas-pressure differentials were affected by accumulating deposits on economizer surfaces and by flow disturbances from slag and ash accumulations in the superheater and reheater ahead of the economizer. These factors gave an annoying lack of consistency in gas-pressure differential in relation to actual rate of gas flow. After exhaustive efforts to improve performance, it was decided to abandon use of the steam-generating equipment proper as a source of gas-pressure differential and install restricting baffles to form an orifice in the short length of flue between the economizer-outlet dampers and the air heater, as indicated in Fig. 4. Observations with this arrangement, however, demonstrated that the pressure loss, as measured across the restricting baffles, was also influenced by the position of the dampers.

Air heaters of the rotating type are generally regarded as too unstable in draft characteristics for evaluating rates of gas flow, their variations in leakage and in resistance of path being thought too great to permit use of a device of that kind in the control of combustion and the metering of gas flow. However, use of the air-heater pressure differentials was adopted as a last resort. Connections for metering gas flow and the gas-flow factor in combustion control were shifted to G and H, Fig. 4. The results have been very promising. The arrangement has now been in operation more than 6 months, and combustion control and air-flow measurement have been found to respond satisfactorily not only to normal fluctuations but in meeting severe sudden changes in load.

Protective devices associated with the combustion control are arranged to give instant stoppage of fuel, coal or oil, on loss of primary air, on excessive back pressure in the furnace, or on failure of both induced-draft fans and, under this last condition, forced-draft and primary-air fans will be tripped out. The use of controlled circulation requires also that all fuel be instantly cut off on failure of circulation, which in this case is effected by the pressure differential across the circulating pumps on dropping to 20 psi.

Although not news to many associated with power-plant opera-

tion, it still cannot be too strongly stressed that successful operation of automatic controls is inescapably dependent upon their conscientious routine inspection and maintenance and, where compressed air is the control-impulse fluid as on boiler 6, absolute cleanliness and dryness of air and reliability of air supply are of vital importance. Control service compressors are provided in duplicate, each of capacity for full requirements, with the station compressed-air system as emergency reserve, each brought in automatically on predetermined fall in control-line pressure.

Control of Boiler Circulation

The control and instrument equipment of group (3), having to do primarily with control of the boiler circulation, are of the general type of group (1). Aside from the signal lights and emergency cutoff for fuel supply mentioned in discussing the equipment of group (1), and certain gages mounted directly on the circulating pumps, the present control equipment of group (3) is identified on the boiler-circulating-pump panel, Fig. 5, which is

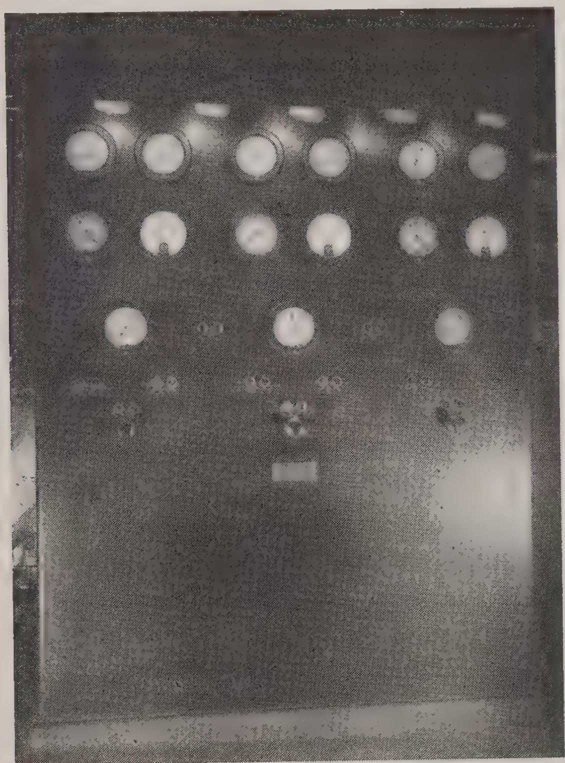

Fig. 5 Circulating-Pump Panel; Gr. 118

set up immediately beside the pumps as indicated in Fig. 4. There are three pumps, one motor-driven and two with combined motor-and-turbine drives. The gages of the two upper rows on the panel are arranged in three groups, showing for each pump labyrinth-seal injection and leak-off pressures and pump-drive-motor amperes.

Rise in labyrinth leak-off pressure, indicating an increase in sealing-water requirements, has served to give warning of wear in the labyrinth gland. The three gages of the next row below are for pump suction and discharge header pressures and steam pressure available at the pump-drive-turbine throttles. Operating signal lights for the two turbines are mounted in intermediate positions in this row. Below are, first a row of signal lights to indicate the position of the individual pump suction and discharge valves, whether open or closed and, below these, the motor-circuit-breaker controls and their open and closed position signal lights. For convenience of the pump operator for close observation while the boiler is being brought up to pressure or taken off pressure, injection pressure gages are mounted close to the injection control valves at the individual pumps.

In addition to the control facilities at present in use, three differential-pressure recorders have been provided and are now mounted in place, but not yet connected, on a separate panel near the main operating board of boiler 6 on Gr. 137. These recorders will give, for each of the three pumps, continuous record of pressure differentials between suction and discharge and between discharge and labyrinth-gland-seal injection. As pointed out in a paper by Parks, Patterson, and Ryan,[4] in the original sealing arrangement the breakdown from the operating pressure of the circulating pump to the final leak-off pressure of approximately 125 psi was effected by labyrinth exclusively. Water flow for sealing purposes and required boiler-feed line pressure, however, proved considerably higher than anticipated. With this condition corrected, the purpose of the new recorders is both to give continuous record of circulating-pump over-all performance and to permit taking full advantage of the gland-seal improvement in lowered feed-line pressure and feed-pump power consumption.

Control of Water Conditioning

As is to be inferred from the other papers of this series, experience with the instruments of group (2) has been of major interest at the station. Preliminary operation of the high-pressure unit early demonstrated that the simple routine of daily tests which had served for the older 375-psi plant was far from meeting the needs of close control of water conditions for boiler 6. More observations and more frequent observations were necessary to avoid too long continuance of contamination or too wide discrepancy between coincident chemical requirements and actual chemical concentrations. Specific factors determined as commonly contributing to upsets of water conditions and found calling for more prompt detection have included condenser leakage; evaporator carry-over; chemical-feed variations; change in continuous-blowdown recirculation; change in continuous blowdown to waste; contamination of stored condensate; subatmospheric pressure in deaerators; carry-over by steam from low-pressure boilers.

Hazards associated with condenser leakage in a seaboard plant hardly need comment. Evaporator carry-over at Somerset Station presented a greater than usual problem because the make-up feed is drawn from a surface supply subject to seasonal organic contamination of a sort which markedly affects the boiling properties of the water. The severe foaming which, prior to the installation of supplemental washing scrubbers on the evaporator vapor lines in the summer of 1944, was liable to occur at unexpectedly low evaporator concentration, was also very liable to pass undetected by any usual spot-sampling routine. The other factors mentioned have recognizable counterparts in most power stations. Factors less usual and less liable to occasional occurrence have also come to attention. All have important potentialities in any high-pressure-boiler plant.

Inspection following the overheating of a furnace-roof tube on January 13, 1943, had shown incipient pitting on tube interior surfaces with corrosion products constituting the major part of the

[4] "Operating History and Performance of 2000-Psi Forced-Circulation Boiler at Somerset Station of Montaup Electric Company," by G. U. Parks, W. S. Patterson, and W. F. Ryan, published on pages 411–427 of this issue of the Transactions.

deposited material within the tubes. It appeared that temperature rise in the metal from the insulating effect of a solid deposit on the interior surface of the tube might be promoting corrosion and so adding to the build-up of solid material with accelerating effects. Recording instruments to reveal and identify corrosive and other unfavorable conditions as they occurred seemed of pressing importance.

Hydrogen evolution at the steel surface was believed to give a reliable index of the rate of metal loss in the absence of free oxygen in the system. Consequently, among the first of the new instruments selected was a two-point hydrogen single-point oxygen recorder, its commercial development and possibilities having been brought to the attention of one of the authors by Sheppard T. Powell. Also, for the utmost promptness in detection and identification of departures from scheduled conditions, so as to minimize the continuance and effects of water variables, recorders were provided to cover continuously the highly informative test values of high-pressure feedwater and boiler-water pH and conductivity, and high-pressure steam and evaporator vapor conductivity. It was further recognized that instruments of this class should be under practically constant surveillance of operators who would themselves take any indicated corrective measures or would be in a position promptly to give proper instructions to others. In setting up the permanent installation, the hydrogen-and-oxygen recorder, the high-pressure feedwater and boiler-water continuous-blowdown pH recorder, the high-pressure feedwater and superheated-steam conductivity recorder, and the high-pressure boiler-water continuous-blowdown conductivity recorder were mounted on the high-pressure turbine-room main operating control board shown in Fig. 6, and the evaporator vapor-conductivity recorder was installed in the low-pressure station pump and evaporator room.

The high-pressure-turbine operator makes an hourly log of chemical-pump operation and continuous-blowdown valve settings; also, of recorded high-pressure feedwater oxygen and hydrogen concentrations, conductivity, and pH; boiler-water conductivity; and superheated-steam hydrogen concentration and conductivity. In logging these values entry is made of the temperature of the samples to the different recorders. Because of the importance of stability of water conditions in boiler 6, the operator makes hourly tests of the high-pressure continuous-blowdown water for alkalinity, phosphate concentration, and pH value as running checks on the recorders. With the close observation of water conditions so maintained in such comprehensive scope, even a slight change is quickly detected and usually its source rather promptly identified so that, in general, correction can be made before any serious effects have resulted. Nevertheless, much more in this direction can still be done. Meanwhile the broadened and heightened interest of the turbine operators, and their conscientious follow-up of irregularities observed in water conditions, are worthy of note and of emulation.

The more complete daily analysis of high-pressure-boiler water and boiler feedwater is made once daily by the plant laboratory and the results of these analyses are used as the over-all check on the hourly tests for basic control of water conditioning. The extremely low values to which contamination from hardness-producing salts is held in the condensate of this tidewater station make it practicable to recirculate boiler blowdown water to assist in stabilizing feedwater alkalinity and deliver the normal dosage of all water-conditioning chemicals continuously at the outlet of the deaerators to be carried through all high-temperature heat-exchange equipment to the boiler. A high-pressure chemical pump is available for delivery of chemicals directly to the boiler drum as special conditions may require.

Notwithstanding the generally small values of feedwater contamination, the recording instruments give very practical demonstration of the importance of continuity and uniformity in chemical feed. With intermittent delivery of chemicals, if there is serious condenser leakage or evaporator priming during an interval between periods of chemical introduction, there are prompt reflections in phosphate concentration and pH value, followed by increase in hydrogen evolution shown by the sample streams of both steam and feedwater. It is an almost invariable observation that serious upset in water conditions causes response in at least some discernible degree in the three feedwater test recorders. On this account, if one instrument shows an unusual in-

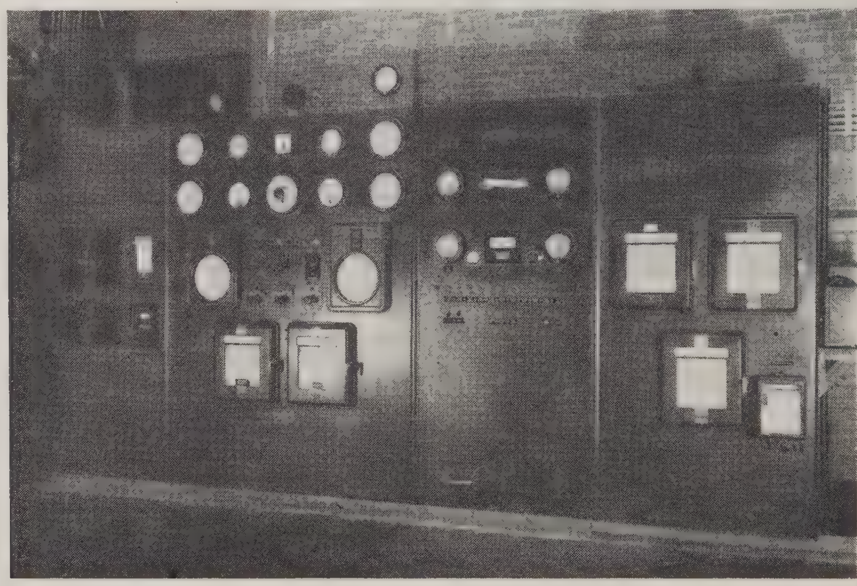

Fig. 6 Main Control Board High-Pressure Turbine; Gr. 137

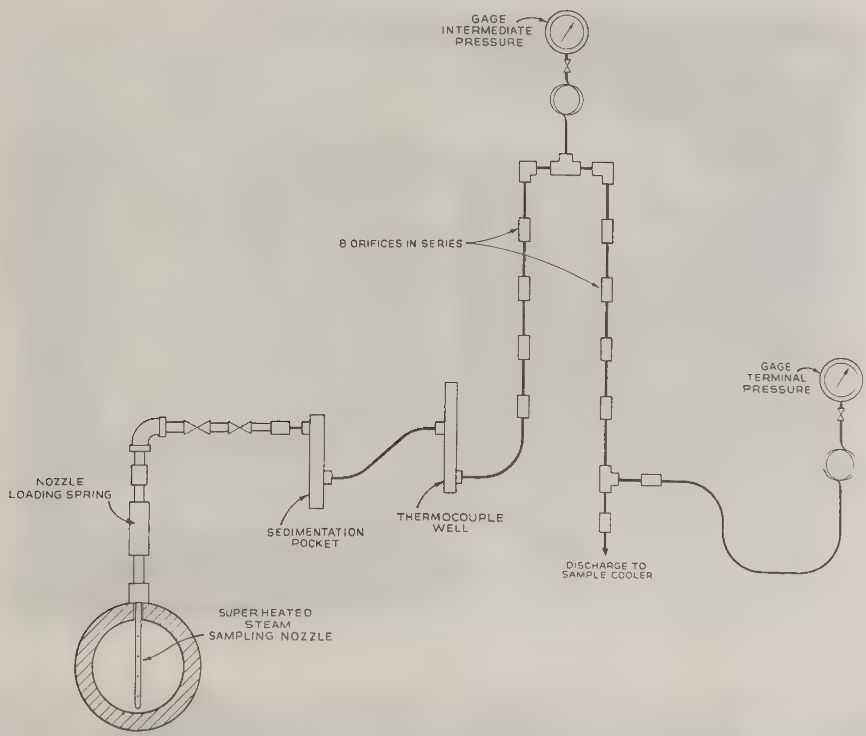

Fig. 7 Diagram of High-Pressure Superheated-Steam-Sampling Connection

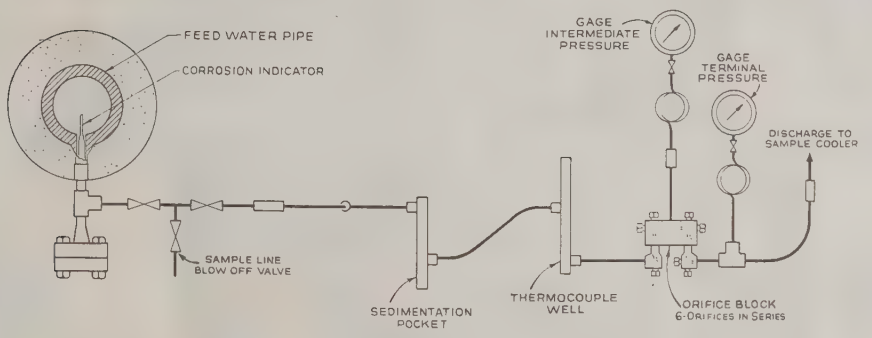

Fig. 8 Diagram of Boiler-Feedwater-Sampling Connection

dication without response of any sort in the records of the others, the condition and accuracy of that instrument are investigated.

Initial sampling connections and piping to the analyzing blocks of the hydrogen and oxygen recorder were temporary in anticipation of exploratory studies to be conducted. With the aid of degassing and sampling apparatus made available in March, 1944, through the courtesy of the American Gas and Electric Service Corporation and Beech Bottom Power Company, several weeks were spent in calibration of the hydrogen equipment and the potential accuracy and dependability of the analyzer and recorder demonstrated. After considerable preliminary investigation of sources of hydrogen liberation in the high-pressure system, permanent connections for sampling were established, and the installation was completed early in 1945. The oxygen-analyzing block of the recorder was connected to the high-pressure feedwater line immediately ahead of the boiler 6 economizer. One of the hydrogen-analyzing blocks receives its sample from the same point. The second hydrogen-analyzing-block connection was made to the high-pressure superheated-steam lead of the 1800-psi 950 F topping turbine.

Steam- and Water-Sampling Arrangements. General arrangements of the high-pressure steam- and feedwater-sampling connections are shown in Figs. 7 and 8. Fig. 9 is from a photograph of the feedwater-sampling connection immediately ahead of the economizer. The steam-sampling nozzle is stabilized stainless steel, and to avoid complexities associated with welding to the carbon-molybdenum steel of the adjacent outer fittings, this nozzle is held to seated position by a special alloy spring. The initial pressure of the steam sample is broken down through a series of eight orifices to about 100 psig. The sample is then led through stainless steel to cooling coils in which it is condensed and the temperature brought to approximately 120 F or some-

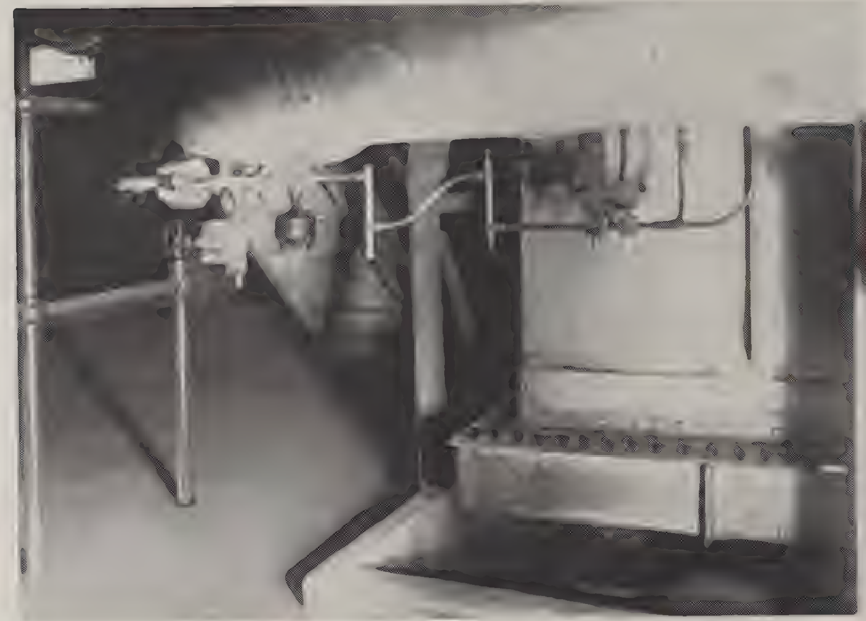

Fig. 9 View of Boiler-Feedwater-Sampling Connection

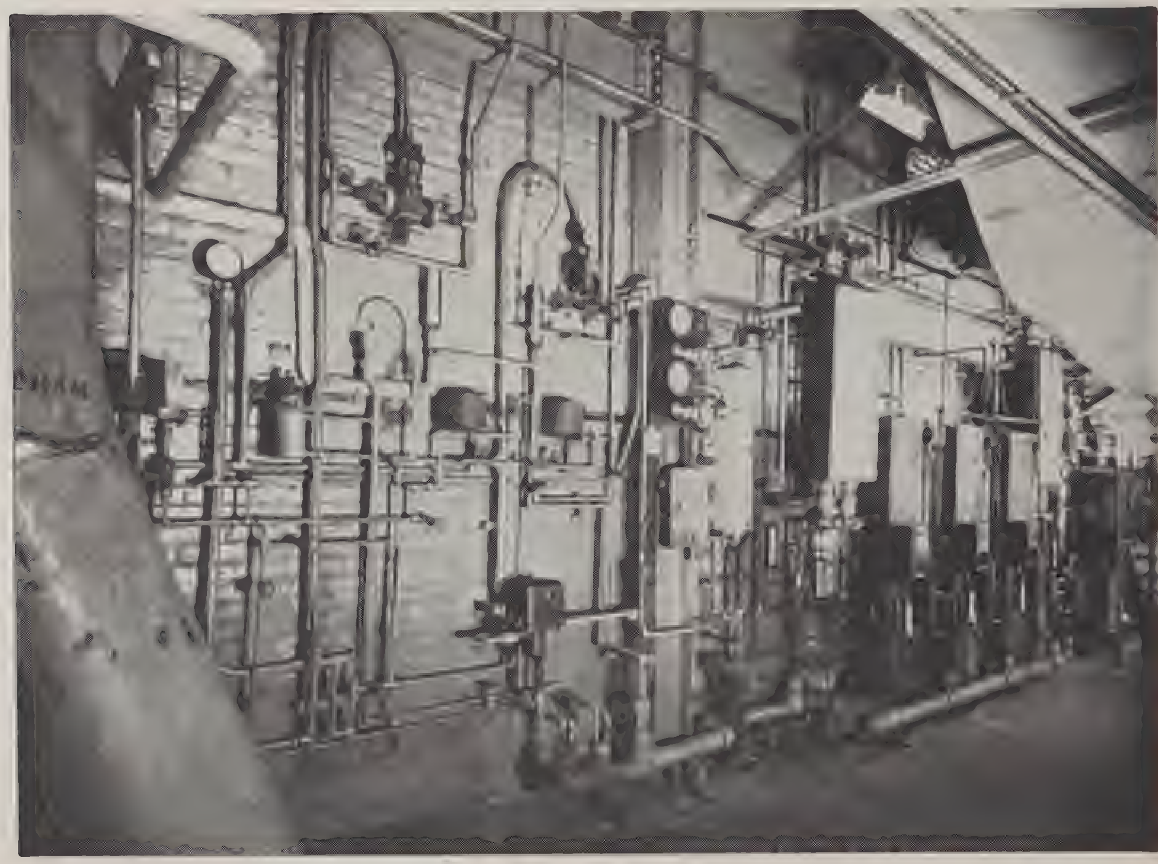

Fig. 10 Hydrogen-Analyzing Blocks, pH, and Conductivity Cells and Accessories

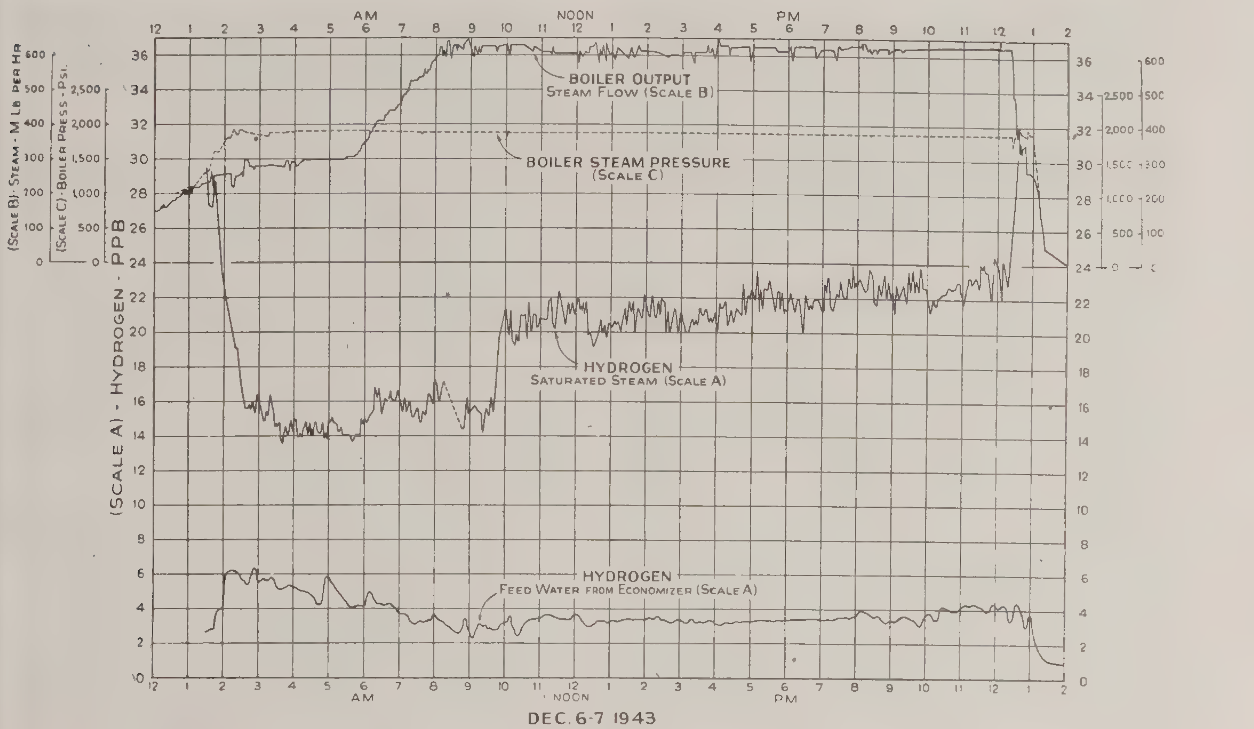

Fig. 11 Graphic Log: December 6 and 7, 1943

what higher for the hydrogen-analyzing block, and for degassing when a precise measure of steam conductivity is desired.

A part of the condensed sample is cooled further, to about 100 F, for purposes of the routine conductivity record. The high-pressure feedwater sample passes first over the corrosion-indicating specimen mentioned in a current paper by Bissell, Cross, and White,[5] then flows through a six-orifice stainless-steel pressure-reducing block in which the pressure is brought down to somewhat under 300 psig. The feedwater sample is also cooled to 120 F or somewhat higher, for the oxygen and hydrogen analyzing blocks, and parts are cooled further to 90 F, approximately, for conductivity and pH measurement. As in the case of the condensed sample of high-pressure steam, the tubing used to convey the feedwater sample is of stainless steel. Intermediate and terminal pressure connections are provided for the series of orifices on each sampling line.

Although used initially for checking the flow capacities of the orifices, the gages are part of the permanent installation to give warning of change in flow characteristics from accumulation of iron oxide in the orifices or from wear. The iron oxide present in the feedwater and steam, which is an important topic of another paper of this series,[5] deposits in the orifices of the feedwater-sampling system but apparently has had no effect on steam sampling. It has so far rather frequently proved necessary to clean the orifices of the feedwater system when returning the boiler to service after an outage. Similarly, at the end of approximately 1 year, the steam-sample flow rate through the orifice series was unchanged and the water-sample flow rate had been increased by roughly 40 per cent. The sample cooling coils and analyzing elements for the group (2) recorders mounted on the high-pressure-turbine operating board are shown in Fig. 10.

Records of Operation. Under current normal operating conditions, oxygen concentration recorded is "zero," expressed in parts per million (ppm), and the approximately full-load concentrations of hydrogen are close to 2 parts per billion (ppb) in the feedwater at the boiler 6 economizer entrance and below 3 ppb in the superheated steam as delivered to the high-pressure turbine. Other full-load characteristics recorded by the instruments of group (2) are conductivity of steam about 1.5 micromhos as sampled, varying with the seasons, and 0.6–0.7 micromho degassed; conductivity of chemically treated feedwater about 5 micromhos; conductivity of boiler water about 1000 micromhos; pH value of feedwater 8.3; and pH value of boiler water 10.3.

To illustrate the potential value of instruments of this general class it has seemed advantageous to select records of important departure from the characteristic performance. Figs. 11, 12, 13, and 14 have been prepared with this in view.

The hydrogen-and-oxygen recorder had been placed in operation November 7, 1943. The manufacturer's field calibration was undertaken within the next 10 days. On placing the instrument in service, the points of sampling at that time being at the economizer outlet and the dry drum of boiler 6, recorded hydrogen concentrations in the saturated steam seemed high, 10 ppb to 18 ppb at times, and showing progressive increase.

On November 27, 1943, the boiler was taken out of service for low-pressure blowdown, and on December 5, 1943, for complete change of water and cleaning of the main drum. Eleven pounds of solids, approximately 45 per cent magnetic oxide of iron from the analysis, were removed from the drum and the boiler was

[5] "Special Studies of the Feedwater-Steam System of the 2000-Psi Boiler at Somerset Station of Montaup Electric Company," by W. D. Bissell, B. J. Cross, and H. E. White, published on pages 429–442 of this issue of the Transactions.

returned to service, attaining full rated pressure and an output of 620,000 lb per hr shortly after 8 a.m. on Monday, December 6.

Values shown in Fig. 11 for hydrogen in feedwater and steam are probably not very significant prior to 12:30 or 1:00 a.m. when the boiler pressure had passed above 1000 psi. Full designed pressure was attained shortly after 2:00 a.m. The progressive fall in hydrogen concentration of the saturated steam shown prior to about 3:30 a.m. is probably due to dilution accompanying the increasing rate of steam generation which at 3:30 was approximately 375,000 lb per hr.

There had been a period of leakage in one of the condensers between midnight and 1:00 a.m. and this was probably, at least in part, responsible for the relatively high evolution of hydrogen in the feed system during the early morning hours. Feedwater make-up at the time was being drawn from the distilled-water storage tank and may have been not entirely free from solids contamination. It will be noted that instead of a fall in hydrogen concentration of the steam which is a usual accompaniment of an increase in rate of steam generation, the increase in boiler loading between 5:30 a.m. and 8:30 a.m. brought a small rise in hydrogen concentration of the steam.

Without much further change in loading, this was followed at 9:30 a.m. by rapid increase in rate of hydrogen evolution in the boiler by about 6 ppb, attaining a concentration in the saturated steam of 20–22 ppb at 10:00 a.m. With this substantially full-rated-load operation continuing the balance of the day, there was gradual progressive increase in rate of hydrogen evolution in the system until at 12:25 a.m., December 7, with the hydrogen concentration in the steam having reached 24 ppb, one of the roof tubes ruptured and the boiler was thereupon promptly taken off the line. The increases in hydrogen evolution in the feedwater system, raising the hydrogen concentration of the feedwater to 4 ppb at 8:00 to 8:20 p.m., and to a maximum of 4.4 ppb after 10:00 p.m., may have been due to leakage of one of the condensers. Records are not complete for the period.

In the light of subsequent experience it is of course perfectly obvious that on at least part of the boiler surface, temperatures had been excessive throughout the full period of operation prior to December 7, during which the hydrogen-and-oxygen recorder had been in service but at the time there was still much doubt regarding the interpretation to be placed upon the indications of the instrument.

As stated in the first paper of the series,[4] there have been two subsequent failures from overheating of furnace tubes in boiler 6. At the time of the first of these failures, discovered on taking the boiler off the line on July 16, 1944, the analyzing block for hydrogen concentration in steam had been out of order for several days, so depriving the operators of the possibility of a warning indication. The circumstances of the second failure October 26, 1945, are discussed later in the present paper under "Value and Limitations of Hydrogen Recorder."

Fig. 12 was prepared primarily to show by means of 6-hr averages immediate and cumulative effects of continued tube leakage in a service water heater. It will be noted, however, that the records show other features of interest. The steam condensed in the service heater was returned at intervals to the boiler-feed system by a trap. The water being heated was from the same surface supply from which water for the boiler-feed make-up evaporators is taken and is very low in dissolved inorganic solids but carries small and somewhat varying proportions of

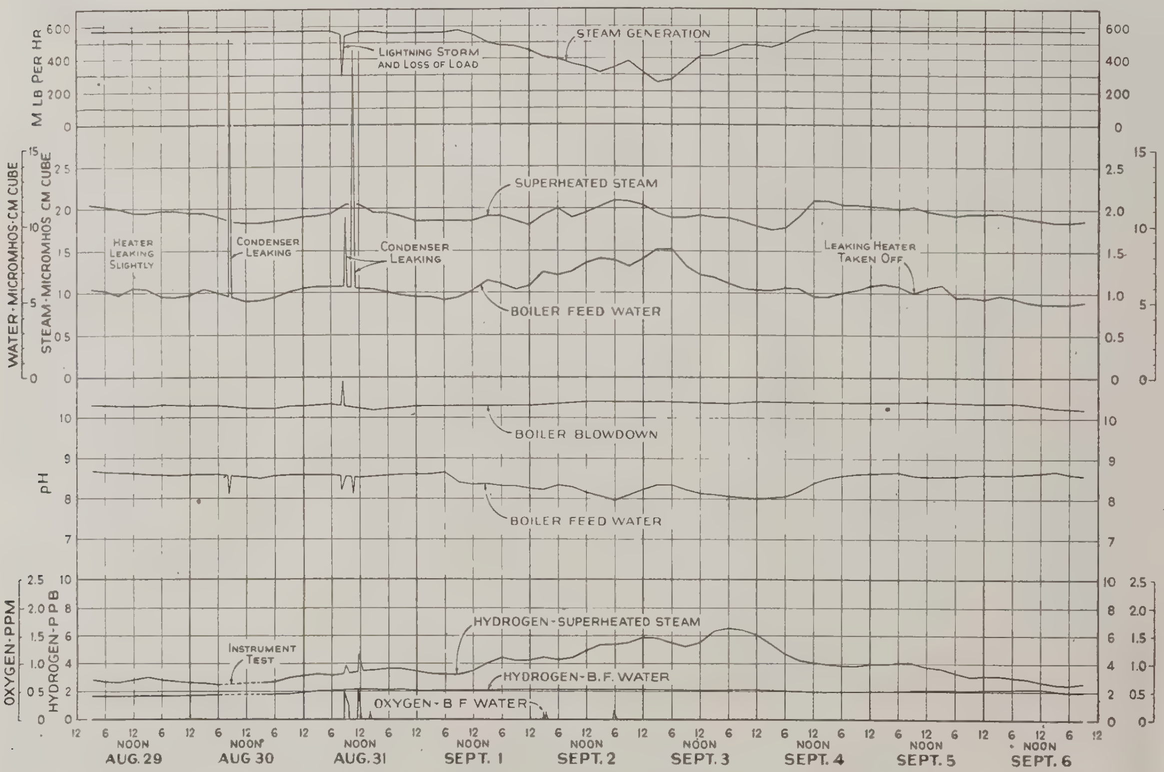

Fig. 12 Graphic Log: August 29 to September 6, 1945, Inclusive

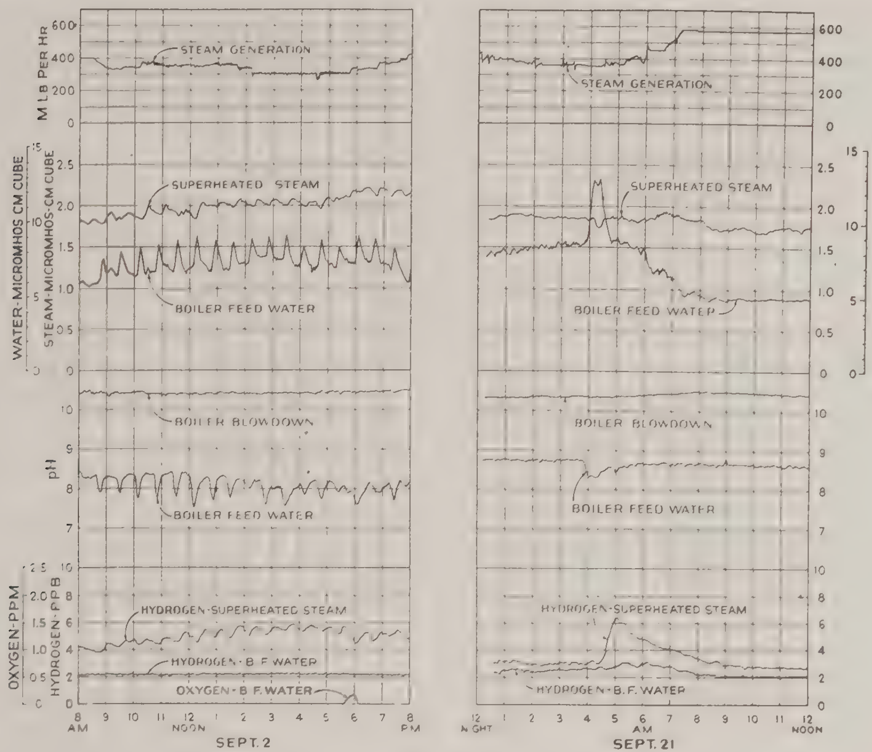

Fig. 13 Graphic Logs: September 2 and September 21, 1945

carbon dioxide and ammonia. Instantaneous effects of the operation of the trap during the height of the heater leakage are shown in some detail in Fig. 13.

Apparently the service-heater leakage on August 29, 1945, was comparatively slight and the irregularities in feedwater characteristics were not immediately sufficient to attract the attention of the operators. However, after the swings in feedwater conductivity became noticeable, it was some time before the source of contamination could be identified, on the evening of September 3, and then it was not found feasible to take the heater out of service for repair until September 5.

The rise in average feedwater conductivity over the period of September 1 to September 3, inclusive, is probably all to be accounted for as a reflection of the week-end and Labor Day holiday sag in load. The sag in feedwater pH average over the same period is believed similarly accountable as due to decrease and absence of low-pressure-boiler blow-down recirculation.

On the other hand, it seems that the high-pressure superheated-steam hydrogen concentration may be reflecting the effects of feedwater contamination to some extent and, on September 2 and 3, was also influenced by the low feedwater pH occurring on those days. As to be inferred from Fig. 13, the gas content of the raw water entering the system through the leaking heater causes increase in the conductivity of the superheated steam. The main cause of the higher-than-usual average conductivity of the steam was probably a high proportion of make-up coupled with seasonal increase in gas content of the raw make-up water.

On August 30 and on August 31 the effects of the service-heater tube leakage were augmented by main-condenser-tube leakage. Shortly after 8:00 a.m. on August 30 leakage of one of the surface condensers caused a brief rise in high-pressure-boiler feedwater conductivity from about 5 micromhos to 22 micromhos. The leakage was rather promptly corrected. The effect on pH value of feedwater was recorded by coincident drop from pH 8.6 to pH 8.1. Any effect on the rate of hydrogen evolution in either boiler-feed system or high-pressure boiler was masked by the test on the instrument begun shortly thereafter.

At about 8:20 a.m. on August 31 a severe lightning storm took out transmission lines which caused an abrupt drop in boiler load equivalent to about 275,000 lb steam generation per hr. The resulting temperature stresses in the condensers caused tube-joint leakage which was reflected in rise of feedwater conductivity from 5–6 micromhos to over 10 micromhos, drop in feedwater pH value from pH 8.6 to pH 8.2, and rise in hydrogen concentration in the superheated steam from 2.7 ppb to 3.5 ppb, the last mentioned being doubtless in part a simple reflection of loss of load. At the same time apparently the abrupt temperature change in the boiler-furnace tubes was sufficient to cause some release of "hide-out" alkalinity, as reflected by the abrupt rise in pH value of boiler water from around pH 10.35 to pH 10.9.

The initial stoppage of the condenser leakage was only temporary, and further leakage was registered about 10:30 a.m. which caused an abrupt and very brief increase in feedwater conductivity from about 6 micromhos to 20.5 micromhos with accompanying drop in pH value of boiler feedwater from pH 8.6 to pH 8.15 and somewhat marked increase in rate of hydrogen evolution in the high-pressure boiler, raising the concentration in the superheated steam from about 3 ppb to 4.4 ppb. The loss of load and leakage of condensers were both accompanied by oxygen contamination of the boiler feedwater. Whether this was due to unbalance of steam supply to the deaerators at the time is uncertain. There was further appearance of dissolved oxygen in

the feedwater on September 2 which is mentioned in the following discussion of Fig. 13.

Eight hours of chart records for Sunday loading on September 2 have been plotted at the left in Fig. 13 to show more clearly the response of the high-pressure-boiler feedwater steam system to the leakage in the service heater. As to be expected, the effects of the discharge of the heater drain trap are most acutely reflected in the conductivity and pH value of the boiler feedwater. The intermittent contamination, probably chiefly from the gaseous content, also appears in the conductivity of the superheated steam. The rate of hydrogen evolution in the boiler-feedwater system seems not to have been influenced, continuing practically constant at around 2.2 ppb as measured at the entrance to the boiler 6 economizer. The effect on hydrogen evolution in the boiler is, however, only slightly less noticeable than that on boiler-feedwater conductivity and pH value. It will be noted that in their general trend, the values for conductivity of superheated steam and boiler feedwater, the pH value of boiler feedwater, and the hydrogen evolution in the boiler are all responsive to the rate of steam generation.

The appearance of dissolved oxygen in the boiler feedwater beginning about 5:40 p.m. and reaching a value of about 0.2 ppm at 6:00 p.m., may have been due either to unbalance of steam supply to the deaerators resulting from the accompanying increase in boiler load or to admission of air-contaminated water in preparing one of the boiler feed pumps for return to service, actually put on the line about 6:50 p.m.

The graphs at the right in Fig. 13, for 12 hours of the morning of September 21, are intended primarily to show the recording of an instance of severe condenser leakage occurring shortly before 4 a.m., and so further illustrating the sensitivity of the feedwater line and boiler system to corrosive and otherwise adversely reactive constituents in the feedwater. The trends of the several values also reflect effects of change in rate of steam generation. Boiler 6 was carrying the entire station load up to 7:15 a.m.

The relatively high conductivity shown by the boiler feedwater prior to the development of the condenser leakage was probably the result of continuing the chemical-feed delivery at the average rate which had been adjusted for the higher load conditions of the previous day. The condenser leakage brought a rather abrupt rise in feedwater conductivity from about 8.5 micromhos to 12.5 micromhos and was effectively checked about one-half hour later, as will be noted from the rapid fall in feedwater conductivity to a value governed essentially by the rate of feedwater flow.

The conductivity of the superheated steam responded in a brief fall of about 0.1 micromhos, suggesting either a fixation of a part of the ammonia or carbon dioxide by dissolved solids of the contaminating harbor water or stabilizing effects of the chloride content of the harbor water. The pH value of the boiler feedwater dropped substantially simultaneously with the increase in conductivity. The recovery of pH value was more gradual. The trend of the dissolved-hydrogen values recorded for the boiler feedwater would suggest a somewhat delayed response. The response in rate of hydrogen evolution in the boiler, however, was quite marked and prolonged, the increase in concentration in the superheated steam being approximately 3.4 ppb, requiring a full 4 hours for recovery.

An item, possibly of interest in connection with the detection of condenser leakage, was met in the early investigation of potential sources of contamination which would affect the operation of boiler 6 when it was found that salt in considerable quantity could be added at the condenser hot well without effect upon the recorded condensate conductivity although appropriately increasing the feedwater conductivity. The hot-well conductivity cells drew their sample from the top of the condensate line ahead of the hot-well pump. Dispersion in the condensate line evidently was not sufficiently rapid.

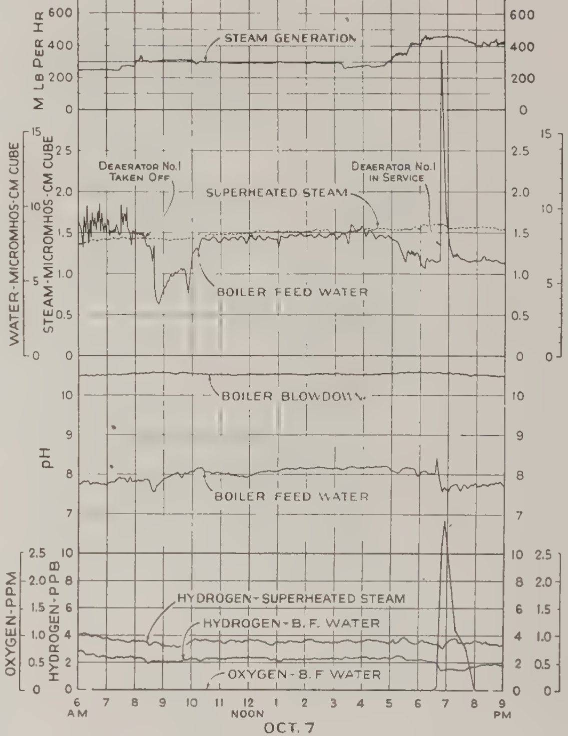

Fig. 14 Graphic Log: October 7, 1945

Fig. 14 shows recorded effects associated with taking one of the station deaerators out of service, and its return, on Sunday, October 7, 1945. The graphs give a 15-hr record. It seems probable that with the light loading of the early morning, there were sufficient differences in deaerator pressures to cause fluctuations in chemical-feed distribution between the two deaerators, resulting in the fluctuations in feedwater conductivity and pH noticeable between 6:00 a.m. and 8:00 a.m. Around 8:30 a.m. steps were taken to cut deaerator No. 1 out of service. The chemical feed to that deaerator was not immediately shut off. With the falling pressure in the first deaerator, chemical feed to the other deaerator was automatically discontinued.

The absence of chemical feed caused immediate drop in conductivity of feedwater from about 8.0 to 3.5 micromhos, accompanied by a drop in pH value of feedwater of about 0.3. On returning the deaerator to service shortly after 6:30 p.m., the abrupt delivery to the feed line of the chemical solution earlier accumulated in the deaerator brought a brief increase in feedwater conductivity from a value of approximately 6 micromhos to somewhat over 20 micromhos, with an accompanying increase in feedwater pH value from about pH 7.8 to pH 8.25. The water first delivered to the feed system from the deaerator returned to service also contained sufficient dissolved oxygen to cause the high-pressure-boiler feedwater to register concentration in excess of 3 ppm, and dissolved oxygen in considerable concentration was not entirely out of the high-pressure-boiler-feed system until more than 1 hour later.

Apparently a part of the oxygen in the feed system was consumed in reaction with the hydrogen evolved in that system. The rate of hydrogen evolution in the boiler shows increase as

an effect of the sudden introduction of the chemical charge but the hydrogen in the feedwater fell appreciably. A corresponding effect on the hydrogen concentration in the boiler feedwater, but of lesser magnitude, is noticeable when the deaerator was taken out of service, although the recorded rise in dissolved-oxygen content at that time was exceedingly brief and reached a maximum of only about 2 ppm. It would seem likely that in this latter instance the effect on rate of hydrogen evolution recorded for the feedwater system was due more to the marked decrease in feedwater conductivity without great loss in pH value than to reaction of evolved hydrogen with oxygen.

Value and Limitations of Hydrogen Recorder. It had of course been recognized that hydrogen evolved in the feedwater system or boiler might combine to some extent with oxygen where found dissolved in the water in contact with the metal surface. A contingency overlooked was that an oxide of copper might function in the system as a source of oxygen. The practical importance of chemical action of that sort was brought strongly to attention by the October 26, 1945, furnace-tube failures taking place without marked increase in recorded rate of hydrogen evolution in the boiler.

One of the low-pressure-turbine stage heaters, temporarily operated as an evaporator condenser with feedwater entering at 215–218 F and condensing vapor of about 16 psig pressure, corresponding to about 250 F, had been returned to this vapor-condensing service October 10, in the midst of the season of maximum organic contamination of the evaporator water supply and immediately following installation of 468 new tubes of Admiralty alloy. Apparently the copper carried into the feedwater system in oxidized form from the corrosion of the new Admiralty alloy tubes was sufficient to cause practically complete disposal of the hydrogen evolved from the overheating of the boiler tubes. Although the boiler had been in operation less than 6 weeks following an inspection when the drum had been cleaned, considerably in excess of the quantity of metallic sludge to be expected from 6 months of operation was found to have accumulated in this brief period, and the sludge was approximately two-thirds copper, which is one and one-half times the maximum proportion of that metal previously found.

A factor which will be recognized as commonly responsible for hydrogen concentrations out of step with temperature or other corrosive influences normally to be associated with concurrent operation is the abnormally high chemical activity of the metal surface following an acid-cleaning of the boiler. The rate of hydrogen evolution which occurs during the first few weeks of operation of an acid-cleaned boiler may be so high as to mask any increase in rate of hydrogen evolution which might result in the boiler from other than extreme contamination or actual overheating.

Notwithstanding such interfering factors as mentioned, hydrogen evolution in a boiler gives the best known indication of corrosion or solution of the boiler metal and the measurement of the rate of this evolution affords the only feasible basis for judging the extent of corrosion taking place while the boiler is in operation. Means for minimizing the effects of interfering factors are being studied.

Conclusions

It will be obvious from the experience in operation of boiler 6 that on modern steam-generating equipment, instruments appropriate to the requirements, especially recording instruments, can be not merely informative but of tremendous instructive value and sources of heightened interest in operating personnel. The importance and the potentialities of such instruments doubtless increase with the degree of operating efficiency which it is desired to maintain, and for a base-load unit of the capacity and type of boiler 6, become essential. To fulfill their potentialities, in fact to avoid real hazard from their presence, the instruments must be rugged by the station environment standard rather than by that of the laboratory and must be followed with definitely scheduled routine inspection, calibration, and maintenance of frequency and thoroughness that will assure precision of indication within acceptable limits at all times.

Comment on the importance of control-equipment maintenance is given earlier in the paper, and should not need repetition. High standard of upkeep in instruments is hardly less vital and, from the point of view of subsequent analysis of operating events, dependable instrument charts can contribute invaluably to the continuity of informative records.

Acknowledgments

The authors desire to record their appreciation of valuable suggestions on the text of the paper made by Mr. G. U. Parks, and of his permission for use of the station data, and of the assistance of Mr. H. E. White in arrangement of these data. They would also record their appreciation of the very constructive criticism of the text by Mr. W. S. Patterson.

The authors take this occasion to express appreciation of the very generous co-operation afforded them by Mr. Philip Sporn, and Mr. W. L. Webb of the American Gas and Electric Service Corporation, and Mr. J. R. McDermet of the Elliott Corporation, which aided greatly the studies conducted in preliminary exploration of hydrogen evolution in the high-pressure system at Somerset Station. Appreciation is also expressed of the very exceptional patience shown by Mr. G. U. Parks and his operating organization and their co-operation in adapting their plant operation to the needs of this investigation.

Discussion

W. W. Cerna.[6] The authors have gone to considerable pains to present complete descriptions and operating experiences of useful instrumentation and control equipment applied to the operation of a high-pressure high-capacity boiler. These descriptions and operating experiences cover not only completely automatically operating and control equipment, but also indicating and recording equipment to enable the operators to have constant knowledge of conditions, and thus provide for ready manual adjustments when the need is indicated.

The authors refer to the log of operations maintained and hourly simple control tests conducted by the high-pressure-turbine operator, in connection with chemical feed, blowdown, etc., for the No. 6 boiler. It has been of particular interest and satisfaction to note that a few minutes spent each hour on these readings and control, plus the daily complete check and establishment of the expected chemical-feed requirements for the next 24 hours by the chemical laboratory, have resulted in boiler-water conditions of unusual uniformity. This is of particular significance when one considers the fact that the Somerset Station is a tide-water plant, using salt water for condenser cooling. The significance of "close control" of boiler-water conditions at this plant will be particularly apparent by referring to the control ranges of various chemicals and salts maintained in the No. 6 boiler water, as given in another paper of this Montaup series by Scott and the writer.[7]

In discussing both the value and limitation of the hydrogen recorder, the authors also point out that copper oxide in the water

[6] Hall Laboratories, Inc., Pittsburgh, Pa.

[7] "Water Conditioning for the 2000-Psi Boiler at the Somerset Station of Montaup Electric Company," by W. W. Cerna and R. K. Scott, published on pages 443–451 of this issue of the Transactions.

system might serve as a means of combining with hydrogen to prevent the indication of hydrogen formation. That this mechanism can and does occur has been confirmed by some laboratory work carried out by Messrs. Kaufman and Trautman of our research department. The tests were conducted in laboratory bombs. One test, using a synthetic but highly concentrated boiler water with considerable alkali present to insure the formation of hydrogen at the test temperature, showed, as expected, considerable hydrogen formation. In another test synthetic boiler water of the same composition was used to which some solid copper oxide was added. The test conditions were repeated, with the result that very little free hydrogen was found and considerable of the copper oxide had been reduced to metallic copper.

Under similar boiler conditions copper salts or compounds would be effective in preventing hydrogen indication and recording by the recording instrument, as illustrated by the following reactions:

$$CuO + H_2 \rightarrow Cu + H_2O$$

$$Cu_2Cl_2 + H_2 + 2KOH \rightarrow 2Cu + 2KCl + 2H_2O$$

R. R. Donaldson.[8] Automatic combustion control, which started out as a convenient accessory for holding constant steam pressure, today has virtually the status of an integral part of the boiler plant. It is depended upon to perform the twin functions of regulating the operation of the boiler so as to satisfy the load demands in a desired manner and of automatically maintaining optimum controlled fuel-air ratios.

In accomplishing the latter function, that of production of proper fuel-air ratios, the ability to measure fuel input and air input is paramount in the present state of the art. It is fair to state that for the present, the production of desired fuel-air ratios by flow proportioning of fuel Btu and air represents the most advanced engineering solution. While someday we may reasonably expect to have gas-analyzing apparatus which is rapid and dependable enough to be used for automatic control of combustion, we have not yet reached that point.

The authors found that it was necessary to do considerable experimental work on their boiler before a satisfactory location for measurement of products of combustion or air flow could be obtained. Their final use of the Ljungström preheater drop gave them a satisfactory solution for this installation.

The difficulties encountered in measurement of air flow for boiler control are not unique with Montaup. There are many installations where perfection of results is limited chiefly by ability to measure air flow satisfactorily.

Admittedly, the problem of metering the products of combustion or the air supply to a large steam-generating unit is far more difficult than the ordinary metering of fluid flows. If, however, in the inception of every new steam-generating installation, the purchaser and the manufacturer of the equipment would spotlight this feature of the boiler design, there is no doubt that better operating installations from the standpoint of maintained furnace efficiencies would result.

The present paper will serve as a stimulus and spearhead to a concerted program for investigating and improving the metering characteristics of modern steam-generating equipment.

[8] Hagan Corporation, Pittsburgh, Pa. Mem. A.S.M.E.

Authors' Closure

Mr. Cerna has contributed valuably in presenting results from bomb tests conducted by Hall Laboratories, Incorporated, in studying the possibilities of reduction of copper oxide in high-pressure boiler water by hydrogen evolved from the boiler metal. Through the courtesy of the Laboratories we learn that similar tests more recently conducted in continuing their investigation have demonstrated that, by increasing the proportion of copper oxide added to the bomb solution, actual evolution of hydrogen may be practically prevented and that the same general effects are observed to occur in the bomb at the temperature with which we are concerned whether or not alkali other than from the boiler metal is present.

Mr. Finnegan's pressing emphasis[9] on the need for more fundamental work on the pure chemistry of the water-metal systems, especially throughout the high-temperature range, is very gratifying. Lack of fundamental information on the reactions of both iron and copper in the range of temperature and other environmental conditions met in the feedwater-boiler water-steam circuits of the high-pressure unit has been a serious handicap throughout the operation of boiler 6 to date.

Mr. Donaldson underscores very aptly the present necessity for dependable, instantaneous, mechanical evaluation of combustion air-flow from the standpoint of the control-equipment designer and manufacturer. The authors concur fully that provisions for evaluation for air-flow rate should be made one of the elementary considerations in design of the modern fuel-burning steam-generating unit.

The improvement work on the flue-gas oxygen-recording equipment and on the boiler-feed-pump flow recirculating valves, mentioned in the paper as to be undertaken, is going ahead promisingly but has not in either case progressed sufficiently to warrant further comment now. The installation of the boiler-circulating-pump pressure differential recorder also is not yet complete. In regard to the last-mentioned equipment however, such brief preliminary observations as have been made have brought to attention valuable data on pressure conditions during periods of boiler start-up and shutdown which would not otherwise have been noted and present impressions are that the information made available by the graphic record will prove even more useful than initially contemplated.

[9] "Special Studies of the Feedwater-Steam System of the 2000-Psi Boiler at Somerset Station of Montaup Electric Company," by W. D. Bissell, B. J. Cross, and H. E. White. See discussion by T. J. Finnegan, pages 441 and 442 of this issue of Transactions.

The Supply of Air to Coal-Fired Steam Locomotives

By R. A. SHERMAN,[1] W. H. BROWNE,[2] and R. B. ENGDAHL[2]

As part of the program of research for Bituminous Coal Research, Inc., means by which the performance of coal-fired steam locomotives can be improved are being investigated. Because of the importance of the method and of the rate of supply of air in any combustion process, an analysis has been made of the factors that govern the supply of air to the locomotive by its unique method of discharge of the exhaust steam to the stack through the front-end nozzle. The analysis shows that for any given front-end arrangement and locomotive, there is a definite upper limit of output of the locomotive at which the weight of air supplied will equal that required for the fuel burned. A similar but lower limit applies on the basis of fuel fired. Below these limits the percentage of excess air will increase with decreasing output of the locomotive; above the limits there will be a deficiency of air. The position of the limits is determined by the performance characteristics of the engine, of the boiler, and of the front-end arrangement. A program of research is outlined which is aimed to supply information (*a*) on the design of front ends for maximum combustion efficiency through proper excess air and minimum carry-over of cinders, and for maximum power output through reduction in back pressure on the cylinders, and (*b*) on the merits of overfire air in improving combustion efficiency through reduction in carry-over of cinders.

The method of supplying air to steam locomotives is unique. Whereas the stationary boiler plant has a high stack or, more commonly, fans to provide the pressure difference necessary to force the air through the fuel bed and the combustion gases over the heat-absorbing surfaces of the boiler, the steam locomotive has only a short stack that barely projects above the boiler shell. The pressure difference, usually called draft, is provided by discharging the steam exhausted from the cylinders through a nozzle at the base of the stack.

Furthermore, whereas the stationary boiler is equipped with dampers or other means by which the rate of air supply can be closely adjusted to the requirements of the load or of the fuel, the locomotive has, practically without exception, no independent control of the air supply. As the output of the engine increases, the amount of steam admitted to and exhausted from the engine increases, and the rate of supply of air for combustion of the fuel increases.

This simple method of supplying the air also serves to a considerable degree to decrease the objectionable noise of the exhaust steam. It was for this purpose that Trevithick and Stephenson, to whom first use of the method is variously credited, originally turned the exhaust steam into the stack. However, although the steam must be exhausted from the engines, the movement of air in this way is, obviously, not without expenditure of energy. The increase in back pressure on the cylinders, necessitated by the restriction of the nozzle, materially decreases the horsepower output of the locomotive.

This paper analyzes the available data on the performance of the air-supply system of locomotives to determine its relation to the efficiency of combustion and to that of the engine at various power outputs. The particular purpose is to determine what information on the problem of air supply is lacking that might be obtained by research, that the performance of coal-fired locomotives may be improved.

Previous Work

Few parts of the locomotive lend themselves to such ease of trials of changes as the exhaust nozzle and its arrangement with relation to the stack. Hence much cut-and-try work has gone on and has been reported in the literature. The International Railway Fuel Association and its successor, the Railway Fuel and Traveling Engineers Association, have long had a Standing Committee on Front Ends, Grates and Ash Pans, and the committee has, with few exceptions, had an annual report. No attempt will be made here to review completely these many reports.

One of the most thorough discussions of exhaust arrangements is that of Chapelon (1)[3] in his book on the steam locomotive, but this has not been generally available in this country. MacFarland (2) presented a discussion principally directed to a demonstration of the power loss through high back pressures because of restricted nozzles and made a plea for induced-draft fans. Jackson (3) recently gave a review of the problem of drafting locomotives with a discussion of the attempts at variable-exhaust nozzles.

Young (4) has presented a complete analysis of the early work on the drafting problem as an introduction to the report of his exhaustive experimental investigation of the performance of various types of nozzles. His work was done on a one-fourth-scale model of a locomotive.

Fry (25) has recently shown that the performance of the front end affects the efficiency of combustion and sets the maximum output of the locomotive.

Methods for Evaluation of Exhaust Arrangements

Despite the long use of the steam nozzle to move air and gases in the locomotive and its extended use in other applications as the boiler-feedwater injector, the steam-jet ejector for air-conditioning systems, and the steam jet for blowing air in overfire-air applications, the action of the nozzle is not wholly understood. The ability of the jet to entrain and to move air is apparently a function not only of the kinetic energy of the steam, but is also related to the exposed surface of the steam. Because of the importance of the latter factor the steam is often divided into multiple streams as in the pepperbox nozzle, or the surface is in-

[1] Supervisor, Fuels Division, Battelle Memorial Institute, Columbus, Ohio. Mem. A.S.M.E.

[2] Research Engineer, Fuels Division, Battelle Memorial Institute. Jun. A.S.M.E.

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

creased by a star-shaped section, or by bridges. It is interesting to note, however, that Young (4) found, as a result of his tests, a difference from the worst to the best no greater than 12 per cent in the performance of various nozzles as expressed by the ratio of the weight of air moved to the weight of steam exhausted through the nozzle.

Because of the obvious necessity of high differences of pressure, or draft to move the gases through the restricted area of the long fire tubes of the locomotive boiler, and also the obvious undesirability of high back pressures on the engine cylinders, the number of inches of water draft in the front end per pound of back pressure has long been used as the criterion of the performance of front-end arrangements. Although this may apply for a given locomotive of fixed resistance to flow, an attempt to apply such a criterion to locomotives of different design and size is obviously in error. It is as incorrect as it would be to rate the capacity of fans by the static pressure that they develop, or to express the capacity of boilers by the pressure at which they operate.

Young (4, 5) clearly recognized that the value of a front-end arrangement should be measured by its ability to move air; this could not be measured by draft alone. In a discussion (6) of the paper by Jackson, he has made a good statement of the problem as follows:

"The actual problem of front-end design is to produce an arrangement which will pull an optimum weight of air through the fire for each pound of steam generated and discharged, maintaining at the same time, three mechanical conditions: 1 Minimum back pressure, 2 clearing the smokebox, and 3 lifting the smoke above the train. The greatest difficulties lie in the mutual incompatibility of these mechanical requirements."

Factors Affecting the Performance of Front Ends

Engdahl and Holton (7) in an investigation of steam-air jets for supplying overfire air have shown that with air tubes whose area is large relative to the area of the steam nozzle, ratios of the order of 200 to 500, as much as 15 to 20 lb of air could be moved per lb of steam used. They found, as others have shown, that the entrainment ratio increases as the steam pressure decreases and as the ratio of the area of the air tube to that of the steam nozzle increases. As they were interested in the jets as blowers to move air at high velocities against small differences in pressure, they did not investigate the effect of resistance to flow, but this should, obviously, affect the performance of jets. The temperature of the gases moved should also be important.

Effect of Steam Pressure at Nozzle. As the output of a locomotive increases, the weight of steam used and exhausted from the cylinders through the nozzle increases. The back pressure at the nozzle increases approximately as the square of the weight of steam discharged. Hence the range of back pressures at the nozzle is large.

Fig. 1 shows the variation of the entrainment ratio, the ratio of weight of gases to weight of steam, with back pressure for a typical locomotive. As expected, the air-steam ratio decreases with increase in back pressure. The ratio is much lower than those quoted from the work of Engdahl and Holton (7) on overfire-air jets because, as will be shown later, of the difference in area ratios of stack to nozzle, and because of the resistance to the flow of gases.

Effect of Ratio of Areas of Stack and Nozzle. As previously noted, almost every steam locomotive has been the subject of experiments on the exhaust nozzle and front-end arrangements and thus has had the ratio of the area of the stack to the nozzle changed. In these experiments, however, facilities have not been available to measure the air flow as well as the steam flow.

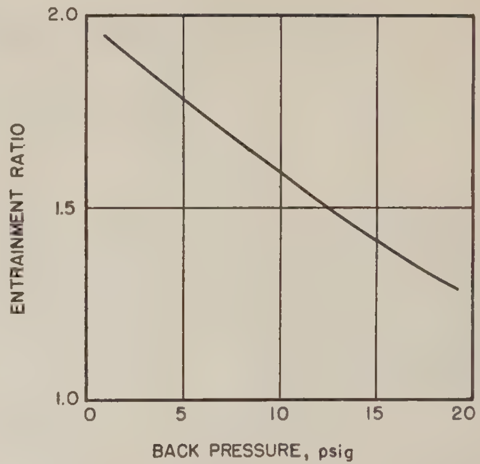

Fig. 1 Relation of Entrainment Ratio to Back Pressure on Nozzle for Typical Locomotive
(Ratio, area of stack to nozzle, 6.2. Resistance ratio, 4.5×10^{-5}.)

For data on the effect of this variable the work of Young (4) is used.

Fig. 2 shows the relation derived from Young's data of the entrainment ratio to the ratio of areas of stack and nozzle. As his data were obtained on a quarter-scale model, the original data have been corrected by the factors that he derived from principles of similarity. The pressures and dimensions are multiplied by 4, the reciprocal of the scale of the model, and weights of air and steam by $4^{5/2}$ or 32.

The curves show the increase of the entrainment ratio as the ratio of the area of stack to that of the nozzle increases. Curves for two steam pressures, 8 and 32 psig, are included. Again, the reduction of the entrainment ratio with increase in pressure is shown.

Effect of Resistance Against Which Gas Is Moved. A third factor which governs the performance of a locomotive front end is the resistance against which the gases must be moved. The resistance is determined by the path of the gases in the front end, the area and length of the tubes and flues of the boiler, the area of the throat back of the arch, the thickness of the fuel bed, the area of the air openings in the grate, and of the air openings into the ashpan.

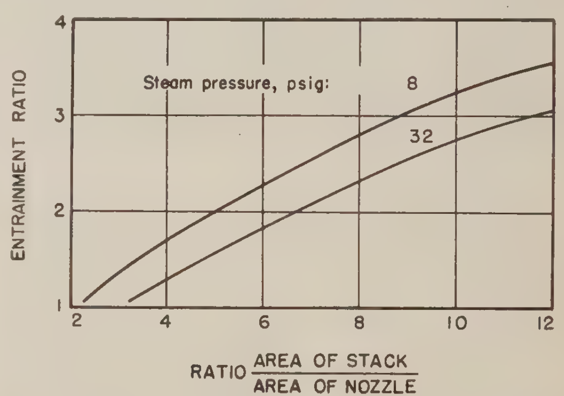

Fig. 2 Relation of Entrainment Ratio to Ratio of Area of Stack to Nozzle
(Resistance ratio, 2.8 to 3.0×10^{-5}. Data from Young (4) corrected for size ratio.)

The data in Fig. 2, as were most of the data presented by Young (4, 5), were taken with a fixed resistance made up of that of the locomotive model and an orifice of fixed area. The resistance was so chosen by Young as to simulate that of a full-scale locomotive.

Fig. 3 presents data obtained by Young in a series of tests in which the area of the orifice in the system was changed to vary the resistance. The solid portion of the curves covers the range of data given by Young; the dashed portion was drawn from data taken at Battelle on a $^1/_{16}$-scale model. Each curve represents one constant steam pressure on the nozzle.

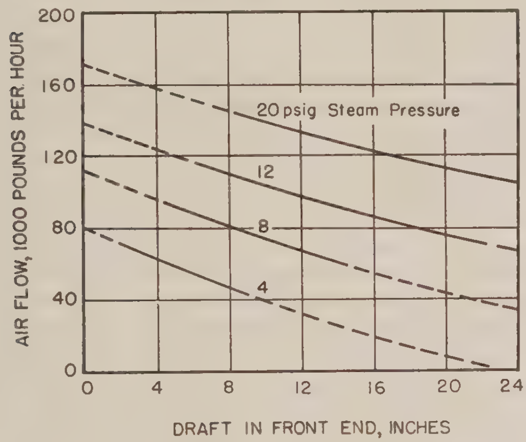

Fig. 3 Relation of Air Flow to Draft in Front End and to Steam Pressure

(Ratio of area of stack to nozzle, 11.1. Data from Young (4) corrected for size ratio.)

The curves in Fig. 3 show the way in which the rate of air flow decreases as the resistance, as indicated by the draft, increases. This figure demonstrates clearly the inadequacy of the draft as a criterion of the performance of front ends. By increase of resistance the draft obtained with a given nozzle and steam pressure can be increased, but the air supplied, which is the true measure of the performance, decreases.

Concept of Resistance to Gas Flow. In consideration of the flow of electricity, we have a concept of three definite measures; current, voltage, and resistance, and the familiar relation that

$$\text{Current} = \frac{\text{Voltage drop}}{\text{Resistance}}$$

In the flow of fluids we have, corresponding to current, the volume or weight of the fluid, and to voltage, the pressure drop, but no term for resistance is in common use. As a result the pressure against which the pump, fan, or ejector moves the fluid is often used as a measure of the resistance. In locomotives the pressure is less than atmospheric and is termed draft.

Unfortunately, also, the term draft is often confusedly used not only as a measure of pressure drop and resistance, but also of rate of flow of gas as well; that is, a high draft is often assumed to be synonymous with a high rate of flow of gas. As just shown in Fig. 3, this is far from necessarily true.

The general relation of the rate of flow of gases to the pressure difference in turbulent flow is

$$Wa = K\sqrt{d}$$

where

Wa = rate of flow in pounds per hour

K = a constant coefficient involving a number of physical factors in the system of flow

d = pressure difference or draft in inches of water

Legein (8) introduced the conception of a factor of temperament, T, in his studies of locomotive drafting, and Chapelon (1) and Young (5) have also used it, but otherwise it has not been used in this country. The temperament is equivalent to the constant coefficient K in the foregoing equation. Thus we have

$$Wa = T\sqrt{d}$$

$$T = \frac{Wa}{\sqrt{d}}$$

Compared with the flow of electricity, the temperament T corresponds to the conductivity.

The reciprocal of the conductivity can, however, be as conveniently used, and we have

$$Wa = \frac{\sqrt{d}}{R}$$

$$R = \frac{\sqrt{d}}{Wa}$$

This expression considers both the pressure difference or draft and the rate of flow of gas and thus permits proper comparison of different locomotives or parts of the gas-flow system of a given locomotive.

Fig. 4 presents Young's data used for Fig. 3 in which the ratios of weights of air and steam are plotted against the resistances to flow. The curves show a sharp decrease in the entrainment ratio with an increase in the resistance. The curves emphasize again the impossibility of accurate comparison of the performance of front ends of locomotives without consideration of the resistance of the gas-flow system.

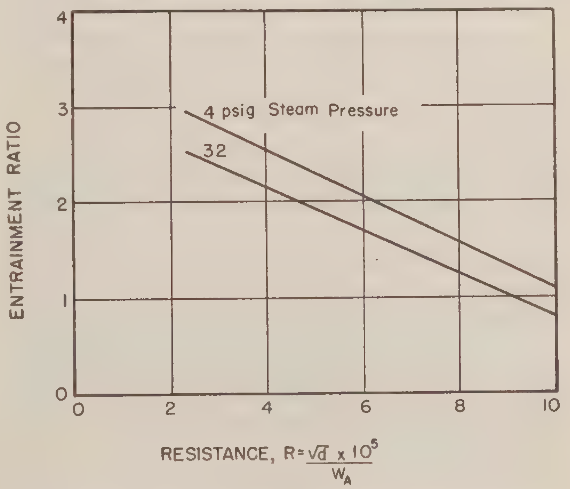

Fig. 4 Relation of Entrainment Ratio to Resistance to Air Flow

(Ratio of area of stack to area of nozzle, 11.1. Data from Young (4) corrected for size ratio.)

Effect of Temperature of Gas. The data of Young that have been used are for the movement of air at atmospheric temperatures, for which most of his tests were run. He did, however, investigate the effect of temperature, and Fig. 5 shows the relation that he found. The reduction in the entrainment ratio was as

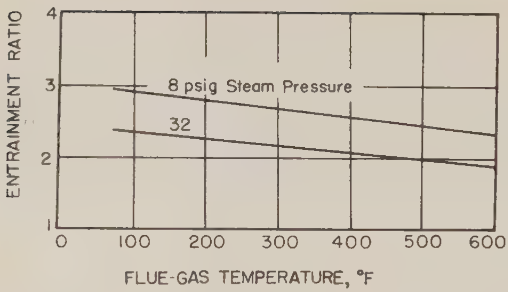

Fig. 5 Relation of Entrainment Ratio to Flue-Gas Temperature

(Ratio of area of stack to area of nozzle, 8.5; resistance ratio, 2.8 to 3.0 × 10^{-5}. Data from Young (4) corrected for size ratio.)

great as 25 per cent for gas at 600 F as compared to gas at room temperature.

The percentage reduction in entrainment ratio by increase in the flue-gas temperature increased as the rate of flow of gases increased. Hence it may be concluded that the effect of temperature in full-scale locomotives would probably be greater than found by Young. His model did not include a boiler, although it did have flues, and thus the temperature gradient in the gases was obviously lower than in a locomotive.

Summary of Factors Affecting Front-End Performance. The data that have been presented show that all four factors, steam pressure, ratio of area of stack to nozzle, resistance of the gas-flow system, and the flue-gas temperature are of major importance in determining the performance of the locomotive front end. Other factors, as the distance between the nozzle and base of stack, and length and taper of stack, also affect the performance but are of less importance. The data also show that it should be possible to predict the performance of a front end and to eliminate the cut-and-try methods that have been so largely used.

That Young's data are applicable to full-scale locomotives may not be apparent at once. Fig. 1 showed entrainment ratios of the order of 1.25 to 2, while the figures prepared from the model tests showed entrainment ratios of the order of 2 to 3. However, we note in Fig. 2 that the entrainment ratio for a steam pressure of 32 psig and a stack- to nozzle-area ratio of 6.2, which is that of the locomotive in Fig. 1, is slightly less than 2. If we apply that point to Fig. 4 at the resistance ratio of about 3 × 10^{-5}, the resistance for the locomotive of Fig. 1, an entrainment ratio of the order of 1.5, or within the range of Fig. 1 will result, even without correction for temperature.

Hence Young's data provide an excellent basis for the development of basic design data for front ends. They require supplementary and confirming tests on full-scale locomotives where care is taken to obtain accurately the weight of gases moved. More data are also required on the resistance of the gas-flow system in locomotives. This, again, requires accurate information on the weight of gas moved.

Relation of Front-End Performance to Air-Fuel Ratio

Now that we have established the general effect of the several pertinent factors on the amount of gas moved by the front end, let us examine how it satisfies the air requirements of a locomotive at various power outputs.

Fig. 6 presents six plots that develop the relation for a typical locomotive between the requirements and the supply of air. Plot (A) shows the relation between the hourly steam output of the boiler and the locomotive output in drawbar horsepower. This is a straight-line relation because the cylinder efficiency and the mechanical efficiency were practically constant over the range of outputs.

Plot (B) shows the relation between the heat-liberation requirements expressed in pounds of coal per hour and the steam output. This is essentially a straight line as the efficiency of heat absorption is practically constant. Note that this is the heat liberated from the coal burned and not the coal fired. As is well known and as will be discussed later, there is a large loss of heating value of the coal fired at high rates because of the solid combustible matter that passes out of the firebox unburned. Combustible matter that does not burn does not use air and is thus not here considered.

Plot (C) shows the relation of the weight of air required per hour to the coal burned. As a definite weight of air is required per pound of coal, this relation, too, is obviously a straight line. Plot (D) shows the relation between the entrainment ratio, weight of air per pound of steam, and the output of steam. Because the back pressure on the cylinders increases as the rate of steam output increases, and because the entrainment ratio decreases with increase of pressure, we have the relation shown.

Plot (E) shows the relation of the air requirements and the air supplied to the drawbar-horsepower output of the locomotive. Combining relations shown in plots (A), (B), and (C) gives us the curve of air required, which is a straight line. Combining (A), (B), and (D) gives us the curve of air supplied. The nozzle in this particular locomotive gave such entrainment ratios that the rate of air supply first increased at a faster rate than the rate of

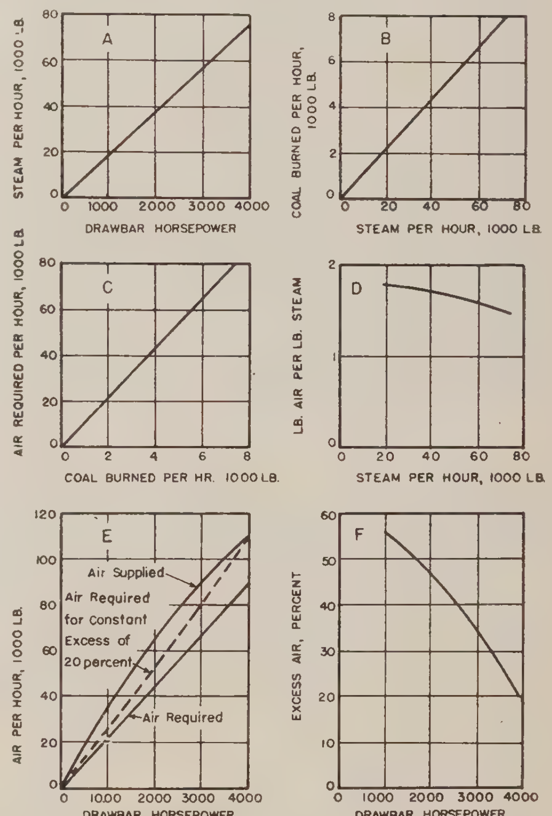

Fig. 6 Development of Relation Between Air Requirements of Locomotive at Different Rates of Power Output and Air Supplied by Front End

air requirements but, because of the decrease of the entrainment rate with increase in back pressure, the rate of increase dropped off at high power outputs.

Combustion of any fuel requires a certain amount of excess air, either to avoid losses in unburned gaseous combustible matter, or to avoid excessively high temperatures. This excess, for good efficiency in heat transfer, should be at a minimum. Whether the excess should be constant over the range of rates of operation, or whether it should increase or decrease with the rate cannot be definitely and generally answered; it depends upon the design of the combustion device.

However, it can be stated with assurance that the amount of air supplied should never be less than required. If it were, although by firing more fuel, the output of the boiler and engine would continue to increase, the unburned combustible losses would become excessive, and the over-all efficiency would be low.

Plot (F) in Fig. 6 shows the relation of the excess air as calculated from the curves of air required and air supplied in plot (E). It can be seen that the percentage excess was rapidly decreasing; if the locomotive had been carried to higher outputs there would have been an actual deficiency of air.

If the front end of the locomotive had been so designed as to follow exactly the air requirements at low outputs the drooping characteristics of the air-supplied curve would have resulted in a deficiency of air at a much lower power output.

The dashed curve in plot (E) shows the correct characteristic for a constant excess of 20 per cent for this locomotive.

Calculations of Conditions for Equilibrium of Requirements and Supply of Air. The relations established by curves as in Fig. 6 can also be worked out by simple mathematical relations. The following expressions give the relations between the weights of air supplied and required:

$$\left[\frac{\textit{Locomotive}}{\textit{Performance}}\right]\left[\frac{\textit{Boiler}}{\textit{Performance}}\right]$$

$$\frac{\text{Lb air req'd}}{\text{Dhp}} = \frac{\text{Lb steam}}{\text{Dhp}} \times \frac{\text{Lb coal}}{\text{Lb steam}} \times \frac{\text{Lb air req'd}}{\text{Lb coal}} \ldots [1]$$

$$\left[\frac{\textit{Locomotive}}{\textit{Performance}}\right]\left[\frac{\textit{Nozzle}}{\textit{Performance}}\right]$$

$$\frac{\text{Lb air supplied}}{\text{Dhp}} = \frac{\text{Lb steam}}{\text{Dhp}} \times \frac{\text{Lb air entrained}}{\text{Lb steam}} \ldots . [2]$$

But in Equation [1]

$$\frac{\text{Lb coal}}{\text{Lb steam}} = \frac{\dfrac{\text{Btu}}{\text{Lb steam}}}{\dfrac{\text{Btu absorbed}}{\text{Btu liberated}} \times \dfrac{\text{Btu}}{\text{Lb coal}}}$$

$$= \frac{\text{Btu}}{\text{Lb steam}} \times \frac{\text{Btu liberated}}{\text{Btu absorbed}} \times \frac{\text{Lb coal}}{\text{Btu}} \ldots\ldots\ldots [3]$$

and

$$\frac{\text{Lb air req'd}}{\text{Lb coal}} = \frac{\text{Lb air req'd}}{\text{Btu}} \times \frac{\text{Btu}}{\text{Lb coal}}$$

It is well established, although not so widely recognized as it should be, that the weight of air required per Btu is practically constant for solid, liquid, and most gaseous fuels. The value is 7.56 lb of air per 10,000 Btu or 7.56×10^{-4} per Btu. Hence we have

$$\frac{\text{Lb air req'd}}{\text{Lb coal}} = 7.56 \times 10^{-4} \times \frac{\text{Btu}}{\text{Lb coal}} \ldots\ldots\ldots [4]$$

Substituting values in Equations [3] and [4] in [1],

$$\frac{\text{Lb air req'd}}{\text{Dhp}} = \frac{\text{Lb steam}}{\text{Dhp}} \times \frac{\text{Btu}}{\text{Lb steam}} \times \frac{\text{Btu liberated}}{\text{Btu absorbed}} \times \frac{\text{Lb coal}}{\text{Btu}} \times 7.56 \times 10^{-4} \times \frac{\text{Btu}}{\text{Lb coal}}$$

which becomes

$$\frac{\text{Lb air req'd}}{\text{Dhp}} = \frac{\text{Lb steam}}{\text{Dhp}} \times \frac{\text{Btu}}{\text{Lb steam}} \times \frac{\text{Btu liberated}}{\text{Btu absorbed}} \times 7.56 \times 10^{-4} \ldots\ldots [5]$$

The factor of calorific value of the coal cancels out and the weight of air per drawbar horsepower is dependent only on the efficiency of the locomotive in the use of steam, the enthalpy of the steam, and the absorption efficiency of the boiler, and is independent of the calorific value of the coal.

This proves that if the not infrequent statement that a locomotive was correctly drafted for one coal, but not for another, is correct, it is not so because of differing air requirements based upon calorific value. It may, however, have a basis in the fact that one coal may require a higher excess of air, either to avoid losses in unburned gaseous combustible, or to avoid clinkering and slagging difficulties because of the fusion characteristics of its ash.

Returning to Equations [2] and [5] for the air supplied and required, it can be seen that in equating them the factor of locomotive performance is eliminated and we have

$$\frac{\text{Lb air entrained}}{\text{Lb steam}} = \frac{\text{Btu}}{\text{Lb steam}} \times \frac{\text{Btu liberated}}{\text{Btu absorbed}} \times 7.56 \times 10^{-4} \ldots\ldots [6]$$

Using Equation [6], we can calculate the entrainment ratio we must have under given conditions. For example, if the Btu per pound of steam, its enthalpy, is 1300, and if the boiler absorption efficiency is 80 per cent

$$\frac{\text{Lb air entrained}}{\text{Lb steam}} = 1300 \times \frac{100}{80} \times 7.56 \times 10^{-4}$$

$$= 1.23$$

The entrainment ratio of the front end must be 1.23 if the theoretical air requirements of the coal burned are to be met. If an excess of 20 per cent is required, the entrainment ratio must be $1.20 \times 1.23 = 1.48$.

Thus as has also been shown by Fry (25) the calculation of the required entrainment ratio for the front end reduces to an extremely simple basis. If we knew accurately the values of the several variables that fix the entrainment ratio, the design of a front end could be reduced to an exact engineering operation.

Relation of Excess Air to Power Output. Relations similar to those in Fig. 6 have been worked out for several locomotives from data from testing plant tests of the Pennsylvania Railroad and the University of Illinois. They all show similar relations to those presented and it is needless to burden this report with them.

The percentage excess air can, of course, be calculated from the composition of the flue gases and it is usually a standard item in the report of a locomotive test. The excess air calculated from the flue-gas composition gives, it must be recalled, the excess on the basis of the fuel burned and not on the basis of the fuel fired. Hence the values obtained are comparable with those presented in Fig. 6.

Fig. 7 presents the relation of the percentage excess air to the output for four locomotives: a Mikado 2-8-2 tested at the Uni-

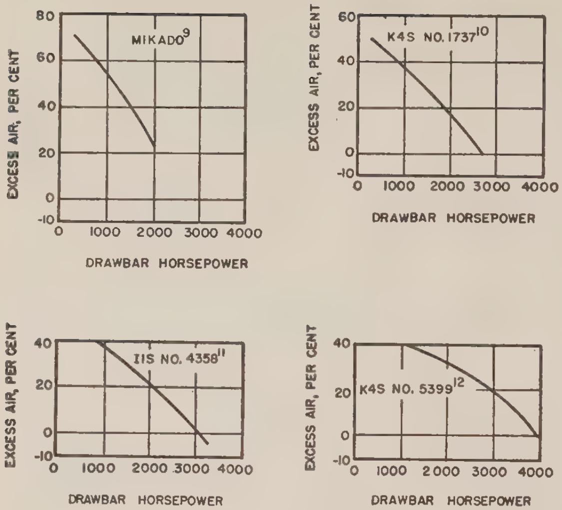

Fig. 7 Relation of Excess Air to Power Output for Four Locomotives

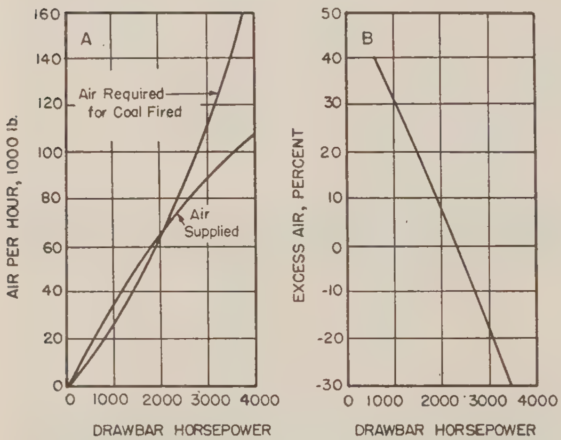

Fig. 8 Relation of Weight of Air Required and Supplied, and of Percentage Excess Air to Power Output of Locomotive on Basis of Coal Fired

versity of Illinois (9); a K-4-S No. 1737 (10); an I-1-S No. 4358 (11); and a K-4-S No. 5399 (12); all tested at the Altoona testing plant of the Pennsylvania Railroad.

The four curves have the same characteristic shape. Two of them reached zero excess air, and one showed an actual deficiency, at the maximum output at which they were tested.

Relation of Air Supplied by Front End to Air Required for Coal Fired. The calculations and considerations up to this point have been on the basis of the coal burned or heat liberated. It is more customary, however, in locomotive practice, to consider the over-all performance on the basis of the coal fired to the grate.

Fig. 8 shows on this basis the relation of the air required and supplied for the same locomotive used as an example in Fig. 6. The method of development was the same as for the former figure, but only the curves for the air required, and the air supplied, and the curve of percentage excess air are shown in plots (A) and (B), respectively.

Above an output of 2200 drawbar horsepower (dhp), the amount of air supplied by the front end was less than that required for the coal fired, and the deficiency amounted to 30 per cent at an output of 3500 dhp. On the basis of the heat liberated from the coal burned, plot (F) in Fig. 6 showed that there was still an excess of 28 per cent at that power output.

Fig. 8 shows that at an output of 3500 dhp the air required for the coal fired was 142,000 lb per hr. At this output, as shown in Fig. 6, the steam generated and exhausted through the nozzle was 65,000 lb per hr. For the front end to have supplied the air required, each pound of steam would have had to have entrained 2.18 lb of air. Actually, the entrainment ratio at this output was 1.50.

The data on locomotive performance which have been examined in this study have shown no front ends that had entrainment ratios of over 2 at high outputs, although conceivably, one might be designed for such performance.

If, however, the front end did supply enough air for the coal fired at an output of 3200 dhp and if, as it must, the performance curve was the same as shown in Fig. 8, plot (A), then it is obvious that the percentage excess air at all lower outputs would have been very great.

General Law of Front-End Performance. From the foregoing considerations one can draw the following generalization or law on the performance of front ends relative to their function of moving the air and combustion gases:

For any given front-end arrangement and locomotive there is a definite upper limit of output of the locomotive at which the weight of air supplied will equal that required for the fuel burned. A similar but lower limit applies on the basis of fuel fired. Below these limits the percentage of excess air will increase with decreasing output of the locomotive; above the limits there will be a deficiency of air. The position of the limits is determined by the performance characteristics of the engine, of the boiler, and of the front-end arrangements.

To use terms common in locomotive practice, this law means that if a locomotive is correctly drafted at a given output it will be overdrafted at all lower outputs and underdrafted at higher outputs. It does not follow, necessarily, that all locomotives are correctly drafted at the maximum output at which they are designed to operate. They may be either underdrafted or overdrafted at that point, but it follows that the trend will be toward overdrafting at lower rates and underdrafting at higher rates. The possibility of manual or automatic controls to maintain a constant excess of air will be discussed later.

Relation of Performance of Front End to Loss of Heat in Cinders

Fig. 9 shows the relation of the percentage heat loss in cinders, together with that of the excess air, on the basis of the coal fired as in plot (B) Fig. 8, to the power output of a locomotive. The rather rapid rise of the loss in cinders at the power outputs in which the air supplied drops below the air required would seem to indicate that the cause of the rise in cinder loss might be the deficiency of air.

This can best be analyzed, perhaps, by comparison of the cinder loss in locomotives where the rate of air supply is a variable, dependent upon the rate of steam generation, and in stationary boilers where the rate of air supply is a variable, independent of the rate of steaming or rate of firing, and is controlled by dampers or fans.

Fig. 10 shows a comparison of the relation of the cinder loss to the rate of firing in pounds of coal per square foot for a locomotive fired by a stoker, and for a stationary boiler fired by underfeed stokers. Data on a stationary boiler fired by spreader stokers would have been preferred, but they were not found. The locomotive data are from the tests by Collins (13) of a J-1-b; the

stationary-boiler data are those given by Driscoll and Sperr (14).

Two facts stand out in these data, as follows: (*a*) At low rates of firing of the locomotive when there is an ample excess of air, the cinder loss is higher than for the stationary boiler; (*b*) at a rate of approximately 75 lb of coal per sq ft per hr, the loss of cinders from the stationary boiler equals that of the locomotive and the direction of the curve is steeply upward.

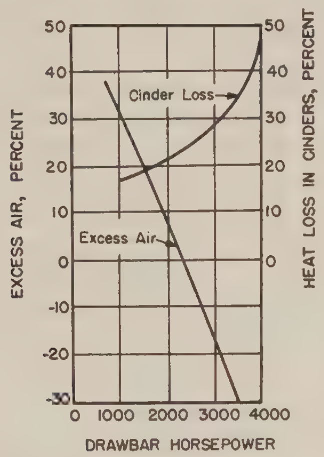

Fig. 9 Relation of Excess Air on Basis of Coal Fired and Cinder Loss to Power Output of Locomotive

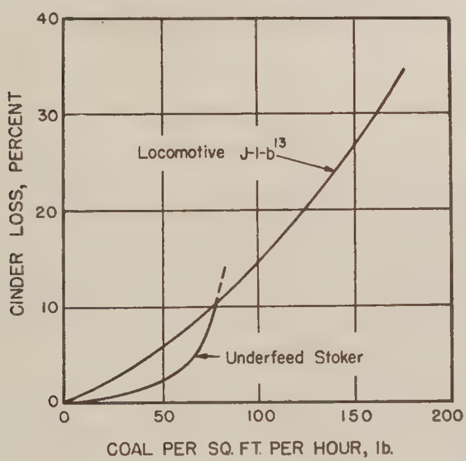

Fig. 10 Comparison of Relation of Cinder Loss to Rate of Firing in Locomotives and Stationary Boilers Fired With Underfeed Stokers

As there was no question of deficiency of air supply in the stationary boiler, the conclusion to be drawn is that the deficiency in air supply, on the basis of the coal fired, is not the major cause of the high cinder loss at high rates of firing in the locomotive. Rather, the high cinder loss is the result of the high carrying power of the gases for solids at the high velocity with which they move through the combustion chamber at high rates of operation. Coal is both carried out of the bed and is swept out of the stream of coal being fired before it reaches the fuel bed.

The probability is that the loss might be somewhat lower if an excess of air were present because the rate of burning in suspension is affected by the amount of oxygen present in the gas. But because the time is so short only the smaller particles would have an opportunity to burn.

Effect of Back Pressure on Power Output

As mentioned in the introduction of the paper, the back pressure on the cylinders which is required to discharge the steam through the nozzle, both increases the amount of steam required for a given output and reduces the maximum power developed. This has always been well recognized, and MacFarland (2) presented extensive data on the back-pressure power loss as calculated from indicator diagrams. Hence but one example will be required to demonstrate the magnitude of the loss of power.

Fig. 11 shows the relation of the horsepower loss by back pressure for locomotive No. 4358 of Class I-1-S (11) to the output. The back-pressure horsepower was calculated as the difference

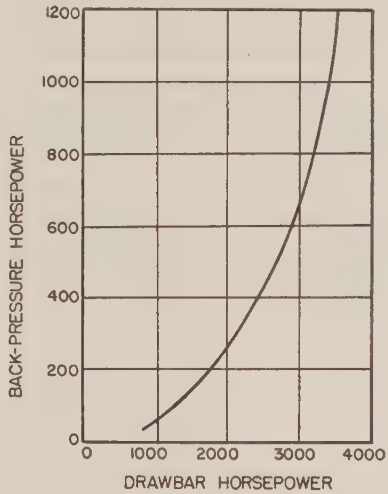

Fig. 11 Horsepower Loss by Back Pressure on Locomotive I-1-S, No. 4358 (Reference 11)

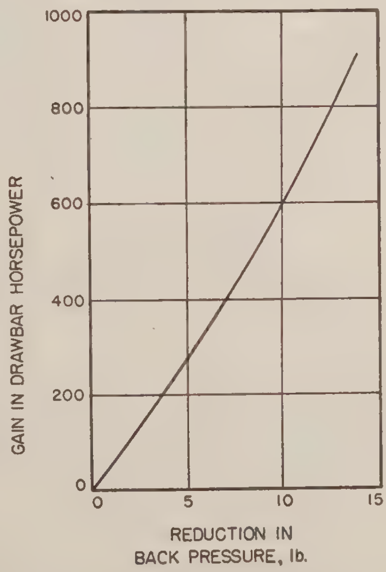

Fig. 12 Gain in Power Output of Locomotive I-1-S, No. 4358, by Reduction in Back Pressure From 14 Psig

in the horsepower actually developed and that which would have been developed if the steam had been expanded to one tenth of the gage back pressure. Although this degree of reduction in pressure was arbitrarily selected, it is satisfactory to illustrate the order of the power loss.

The loss in power increases rapidly with the increase in the net power output and amounts to about 1200 hp or one third of the net maximum output of the locomotive. The desirability of reduction of this loss is obvious.

Fig. 12 presents one further demonstration of the gain in power output by decrease in back pressure. This shows, for the same locomotive as Fig. 11, the gain in output for the same steam consumption by reduction in the back pressure at an output of 2930 dhp, and a back pressure of 14 psig. The gain is slightly more than directly proportional to the reduction in pressure, but for this locomotive each 1-lb reduction means a gain of approximately 60 dhp.

Efficiency of Front End

Fig. 13 shows the power required to supply the air and move the combustion gases through the boiler for the same locomotive used in the example, Fig. 11. The air horsepower was calculated in the usual manner from the rate of flow of the gases, with correction for temperature, and the pressure against which the gases were moved.

Also shown in Fig. 13 is the ratio of the power required to move the gases to the power lost by the back pressure. Above an output of 1000 dhp the ratio is about constant at $^{1}/_{10}$, that is,

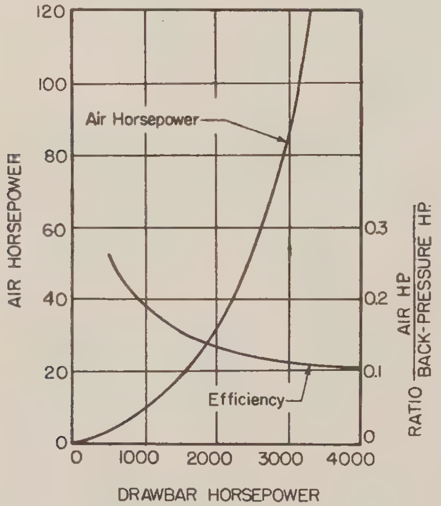

Fig. 13 Relation of Air Horsepower Required to Move Air and Gases and of Ratio of Air Horsepower to Back-Pressure Horsepower to Output of Locomotive

the efficiency of the front end, by this method of calculation, is 10 to 20 per cent.

Young (4) has shown that on the basis of the kinetic energies in the gases and in the steam at the nozzle, the efficiency of the front end is of the order of 5 per cent, but it is considered that the foregoing calculation is of greater significance.

Possibilities of Improvement in Design and Performance of Locomotive Air-Supply Systems

The lines along which efforts to improve the combustion efficiency and over-all performance of the coal-fired locomotive should proceed become clear from the analysis that has been presented. They are as follows: (*a*) To control the drafting appliance that the percentage excess of air for the coal burned will be essentially constant at all rates of operation; and (*b*) to move the air and combustion gases with the minimum back pressure on the cylinders.

With these objectives in mind we shall review the various methods of improvement that have been tried in the past. These include the following: (*a*) Change in area or shape of the nozzle; (*b*) change in area of stack; (*c*) change in resistance to air flow; (*d*) manually and automatically variable nozzles; and (*e*) induced- and forced-draft fans.

Change in Area and Shape of Nozzle. Probably the most common method of changing the locomotive to improve its "drafting" is to change the diameter of the nozzle or to change to one of

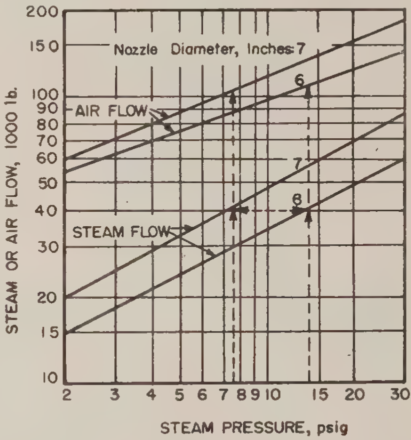

Fig. 14 Relation of Steam Pressure at Nozzle to Rate of Flow of Air and Steam
(Data from Young, reference 4, corrected for size ratio.)

special shape. If the engine does not "steam" properly the usual step is to insert a small nozzle. If it seems wasteful of fuel or emits excessive cinders, a larger nozzle is put in.

Fig. 14 presents curves from Young's work (4) which permit an analysis of what happens when the size of the nozzle is changed. The relations between the steam pressure on the nozzle and the rate of steam and of air flow are plotted on logarithmic paper on which they assume straight lines.

Let us assume first that the locomotive has a 7-in-diam nozzle and is found not to steam properly. A 6-in. nozzle is installed. The dashed lines indicate, as an example, that when the pressure at the nozzle was 7.6 psi, the rate of flow of steam was 40,000 lb per hr, and the rate of flow of air was 105,000 lb per hr. If the flow of steam remains the same, the back pressure with the 6-in. nozzle will be 14 psi; the air flow will be 109,000 lb per hr. The increase is only 4 per cent and is caused, it is to be noted, not because of the increased steam pressure, as we have seen that this decreases the entrainment ratio; instead, it is because of the increase in the ratio of area of stack to nozzle because of decrease in the area of the latter.

Actually, to maintain the same power output from the locomotive with the smaller nozzle would necessitate an increase in steam flow because of the increased loss in power by increase in back pressure. The cutoff would have to be increased, the rate of flow of steam and the back pressure would increase, and the flow of air would increase. A greater net output will be obtained from the engine but at a lower engine efficiency.

Conversely, it can be seen that if a change was made from a 6-

in. to a 7-in. nozzle, the cutoff could be decreased to obtain the same output and the steam flow and air flow would decrease more than shown by the dashed lines.

The size of the nozzle must be of course that which will give the design output of the locomotive, but data should be made available which would make unnecessary a trial-and-error method to establish the correct size. As for special designs and shapes, some have merit, but Young's findings have already been recalled; namely, that among all that he tried the best was only 12 per cent better than the poorest.

Although the size of the nozzle may be varied, we have seen that the laws of flow are such that the percentage excess air supplied will decrease with increase in rate of output of the locomotive, and a fixed nozzle cannot change this fundamental relation.

Change of Area of Stack. The dimensions of the stack of a locomotive once built are not susceptible of simple change and less experimenting has been done with the stack.

For maximum performance of any given combination of stack and nozzle, the distance from the nozzle to the top of the stack must be such that the steam will fill or "seal" the stack. Because the height of the stack is limited by the clearances of the right of way, there is a limit to the stack diameter that can be used.

Furthermore, as the stack diameter is increased, the velocity of discharge of the gases is decreased and trailing smoke that obscures vision may result. European locomotives have often used "wings" to lift the gases above the train. They have also been used to some extent in this country.

Change of Resistance to Air Flow. The analysis in the earlier part of the paper showed that the resistance against which the air and gases are moved is one of the principal factors that determine the entrainment ratio of an exhaust nozzle. The expression for resistance

$$R = \frac{\sqrt{d}}{Wa}$$

was suggested.

Table 1 gives some of the data on five locomotives, the average over-all resistance calculated from the draft in the front end, the resistance of the tubes from the pressure drop across them, and the ratio of the two resistances.

Resistance values from individual tests of each locomotive showed no general trend with locomotive output but were, with few exceptions, nearly constant at the average value shown. When it is seen that the resistance of the tubes is 67 to 75 per cent as large as the total resistance, it is not surprising that the over-all resistance should not change with a change in the thickness of the fuel bed or other variables in the course of tests at various rates of output.

Table 2 gives the pressure drop across various parts of the air- and gas-flow system of a locomotive, the resistance of each section, and the ratio of each to the over-all resistance.

The last column of Table 2 gives the ratio of the resistance of each of the several parts of the system to the over-all resistance. It must be remembered that the sum of the resistances is not equal to the total resistance; instead, the total resistance equals the square root of the sum of the squares of the resistances of the separate parts..

Resistance of Diaphragm and Tubes. The two largest single resistances in the system are those of the diaphragm and tubes. They are, respectively, 59 and 51 per cent of the over-all resistance or together

$$\frac{100\sqrt{2.3^2 + 2.0^2}}{3.9} = 78 \text{ per cent of the total}$$

Hence it would seem that here lies the greatest opportunity for reduction in the resistance and thus the improvement in the performance of the exhaust system.

The function of the diaphragm is to break up and extinguish cinders that they may be discharged freely and harmlessly from the stack. Many variations, such as so-called cyclone front ends, have been tried. A review of all suggested designs and an experimental program on new designs would now be in order.

The resistance of the tubes is high because of the high velocity with which the gases are passed through them. This high velocity results in excellent heat transfer but, as previously pointed out, at the expense of a loss of efficiency of the engine because of the high back pressure.

The over-all efficiency of the locomotive in its use of the heat liberated from the coal is the product of the efficiency of the engine and that of the boiler. As the efficiency of the engine decreases and that of the boiler increases, as the area of the tubes is decreased or their length is increased, there is a value for area and length at which the over-all efficiency is at an optimum.

An analysis to establish whether the values now used in locomotive design are the optimum should be made. If not, experimental research to establish the optimum should be carried out.

Resistance of the Arch. The resistance to the flow of gases backward to the throat of the arch and then by a hairpin turn forward is an important part of the resistance, but the gain in combustion efficiency by this turbulence and in absorption efficiency by the surface of the tubes is large and cannot be sacrificed. It is possible, however, that in the locomotive used as an example, and others, the throat is too small and should be increased.

Resistance of Fuel Bed and Grate. What part of the resistance of the fuel bed and grate is caused by the fuel bed and what by the grate separately is not known. They must be considered together. The total is the smallest of the four resistances so far considered. Even if the resistance could be removed entirely, the flow through the system would not be greatly increased.

For example, the over-all pressure drop would be reduced from

TABLE 1 AVERAGE RESISTANCE TO FLOW OF GAS FOR FIVE LOCOMOTIVES

Locomotive	Max dhp	Grate air openings, per cent	Areas, sq in. Nozzle	Areas, sq in. Stack	Ratio, (Area of stack)/(Area of nozzle)	Resistance Total	Resistance Tubes	Ratio, R(tubes)/R(total)
I-1-S 4358(11)	3334	32	35.9	314	8.8	5.0	3.7	0.74
H-8-SB 387(15)	1588	27	30.9	227	7.3	5.7	4.3	0.75
Mikado 1742 (9)	1833	37	28.3	265	9.4	4.5	3.1	0.69
K-4-S 1737(10)	2876	28	38.2	214	5.6	5.2	3.5	0.67
K-4-S 5399(12)	3934	15	51.2	310	6.1	4.0	2.7	0.68

TABLE 2 DRAFT, PRESSURE DROPS, AND RESISTANCE TO GAS FLOW IN LOCOMOTIVE[a]

Position	Draft, in. water	Pressure drop, in. water	R, $\frac{\sqrt{d}}{Wa} \times 10^5$	Ratio to over-all resistance
Front end	20.1	...	3.9	1.00
Across diaphragm	..	6.9	2.3	0.59
Back of diaphragm	13.2	...	...	..
Across tubes	..	5.3	2.0	0.51
At flue sheet	7.9	...	...	..
Across arch	..	4.2	1.8	0.46
Firebox	3.7	...	...	..
Across fuel bed	..	3.0	1.5	0.38
Ashpan	0.7	...	...	..
To atmosphere	..	0.7	0.7	0.18

[a] Drawbar horsepower = 3839. Air flow = 114,600 lb per hr.

20.1 to 17.1 in. and the resistance would be as 20.1 to 17.1 or as 4.5 to 4.1. The rate of flow for a given draft varies inversely as the resistances. Hence for the same draft in the front end the rate of flow without the resistance of the fuel bed and grates would be $\frac{4.5}{4.1} = 110$ or 10 per cent greater.

This shows that the effect of reductions in the percentage air openings in the grates, which is often now as low as 7 to 10 per cent compared to 40 to 50 in earlier locomotives, is not that of so increasing the resistance that the ratio of air to fuel is decreased. Instead, the advantage of reduced air openings is that they improve the distribution of the air over the grate and prevent the passage of a large amount of air through a hole that may occur in the fuel bed.

Resistance of Ashpan. The resistance offered to the flow of air by the inlets to the ashpan is much the smallest of the several resistances, but the fact that there should be any resistance, as evidenced by a draft in the ashpan, has often been questioned by engineers.

Even if it were removed, the change in the over-all resistance would be small, of the order of

$$\frac{\sqrt{20.1 - 0.7}}{\sqrt{20.1}} = \frac{4.4}{4.5} = 0.98$$

that is, a 2 per cent decrease. This is small but a reduction would undoubtedly be desirable. The seemingly prevalent rule that the area of the inlets to the ashpan should be 15 per cent of the area of the grate may well be questioned.

Manually and Automatically Controlled Front Ends. Many suggestions have been made of variable nozzles, some controlled manually from the cab and some automatically. The history of the manually controlled nozzles has been that they were difficult to maintain in working order at the temperatures of the front end or that they were closed to their minimum at all times. Their acceptance in this country has not been great.

Lewis Nozzle. Of the several automatically controlled nozzles, the Lewis, described by Jackson (3), attained some use. It was a slot-type nozzle in which two vanes were mounted on shafts which were geared together and connected to a weighted loading arm damped by a dashpot. The vanes "pumped" at low engine speeds as the back pressure varied, but at high speeds they rode at balance, opening further as the pressure increased. The desired functioning was never clearly described in terms of rate of air flow at various outputs. One statement was made that a constant draft was obtained. This is obviously impossible and undesirable as the draft must increase as the rate of output increases.

The action must have been that by opening with increased pressure, the back pressure was held down but because the ratio of area of stack to nozzle was decreased, the rate of air supply was less than it would have been with a fixed nozzle. The reverse has been shown to be the desirable action.

Bertram Automatic Drafting. A device for automatic control of the draft on a locomotive is covered by two patents (16, 17) to Bertram, and some results have been described. It consists of several spring-opposed (18) valves in a line leading from the exhaust pipe to the atmosphere. The valves start to open at a pressure of 5 to 6 lb, and by by-passing part of the steam from the exhaust nozzle the back pressure is reduced. Also, it must reduce the amount of air moved by a nozzle of the same size; whereas we have seen that the need is for a greater amount of air at high rates and a reduction at low rates.

If a smaller nozzle than normal for a nonautomatic nozzle were used and the spring-loaded valves opened early to by-pass a part of the steam at low rates, some measure of the desired action might be attained, but neither the patents nor descriptive literature clarify the exact action. The article states that 59 locomotives of the Lehigh Valley were equipped with the device from 1938 to 1941, but it is understood that all have now been removed.

The Clemac Front End. Another automatically controlled front end has been recently developed by W. H. Clegg, general superintendent, Motive Power and Car Equipment, and R. R. McIntosh, mechanical engineer, Grand Trunk Western Railroad. Only a short discussion has been published (19), but Messrs. Clegg and McIntosh have kindly made all details available to the authors. The development is also well described in two patents (20, 21).

The essential features are as follows:

1 Back-pressure relief valves of the weighted poppet type to govern back pressure on the cylinders by exhaust of steam to the atmosphere.

2 A draft-relief valve that opens to admit air to the front end and thus limits the draft. The draft at which the valve opens increases as the output of the locomotive increases because the valve is loaded by the pressure of the steam at the nozzle through a piston on the upper end of the stem of the valve.

3 A cinder ejector which consists of a pipe that extends from the point of maximum draft inside the skirt of the stack to the bottom of the front end.

4 The elimination of the diaphragm, table plate, draft sheet, and netting from the front end.

The action of the back-pressure relief valve is not essentially different from that of the Bertram, and the same objections apply.

The draft-relief valve may have merit in that it should be possible to maintain the air-fuel ratio constant or to vary it as desired over the range of outputs.

The elimination of the diaphragm, table plate, draft sheet, and netting should decrease the over-all resistance to the flow of gases, permit a larger nozzle, and thus reduce the back pressure on the cylinders. Why these supposedly essential features can be eliminated by the inclusion of the back-pressure and draft-relief valves is not clear. However, the two locomotives equipped with the device have been approved from the standpoint of danger of forest fires for operation in Michigan by the Conservation Department, and in Canada for operation on the associated lines of the Canadian National Railways.

Admittedly, there is much that is not known about the factors that control the pickup and carry-over of cinders. If the carry-over varies as some power, greater than 1, of the rate of flow of gases, the carry-over will undoubtedly be high at the instant the exhaust valve on the engine cylinder opens, when the rate of discharge of steam is at a maximum. Thus it is conceivable that the carry-over would be considerably less for the same total flow if the rate of flow of gases were continuously at the average value than if the rate varies widely, as it must with the intermittent exhaust of steam at widely varying pressure.

Correct Principles of Automatic Control of Air Supply. The automatic devices described appear to have been designed in the belief that a locomotive is overdrafted at high rates and underdrafted at low rates, whereas we have seen that the reverse is true. The correct principle is to install a nozzle of the correct size to give the required output of the locomotive and the desired excess air at that output and to open the nozzle, to by-pass steam, or to relieve the front-end draft at lower outputs to maintain a constant excess of air.

If this were manually controlled an indicator in the cab would be required as a guide to the proper setting. A dual gage, one pointer indicating flow of steam by pressure drop across a nozzle, the other indicating flow of gas by pressure drop across the tubes

with such adjustment of the linkages that when the pointers were together the rate of air supply was correct, would be suitable. This is the principle of the Bailey boiler meter, widely used on stationary boilers and adaptable to automatic control. The difficulties of application to locomotives with widely varying pressures, particularly at low speeds, are apparent.

Induced- and Forced-Draft Fans. The possibility of the application of either forced- or induced-draft fans to the locomotive has occurred to many. MacFarland (2) as the result of his studies of the power losses because of back pressure, designed an induced-draft fan operated by high-pressure steam. The design was applied to one switcher and three road locomotives of the Atcheson, Topeka and Santa Fe Railroad from 1912 to 1914. From 1920 to 1927 Dr. W. F. M. Goss experimented with a turbine-driven induced-draft fan using exhaust steam.

Jackson (3) reports that both of these failed because of the impossibility of maintenance of the blades against the erosion by slag particles and cinders.

The Committee on Front Ends, Grates, and Ash Pans of the International Railway Fuel Association (22) has described a forced-draft installation on a locomotive of the Texas and Pacific, burning both Texas lignite and bituminous coal. Although the first results were promising, no further application seems to have been made. The fan was operated by a turbine, presumably on exhaust steam, although this was not definitely stated. The exhaust of the turbine passed through a nozzle to the stack. Both the nozzle and stack were considerably enlarged as compared to normal practice.

The difficulties of forced draft are obvious. The firebox will be under pressure greater than atmospheric, and the firing door cannot be opened unless the fan is shut off.

Overfire Air. The use of overfire-air jets to decrease the loss by unburned combustible gases and to decrease the smoke has long been practiced and is now coming into greater favor because of the work of Bituminous Coal Research, Inc. (7, 23, 24). This work has given the correct principles of design of steam and fan-driven jets and has shown how the noise which so often prevented their use, can largely be eliminated by a simple silencer. More than 600 locomotives on 24 railroads have now been equipped with jets based on the BCR design.

If 10 or 20 per cent of overfire air is desirable, the question arises whether the supply of 30 to 50 per cent of the air would not be more desirable.

The introduction of the air over the fire would do little to relieve the work on the front-end nozzle and thus reduce the back pressure because the gases would still have to be moved around the arch, through the tubes, and around the diaphragm. We have seen that these constitute the bulk of the resistance of the system.

Whether the use of large proportions of overfire air would reduce the large losses in cinder at high rates depends upon the amount of this carry-over that comes from the fuel bed and the amount that is carried from the coal thrown from the stoker and thus never reaches the bed. Only experimental research will determine the value of overfire air, but as the possible gains are great, a thorough research is warranted.

Conclusions and Recommendations for a Program of Research

This analysis of the problem of the supply of air to coal-fired locomotives by means of the "front-end" system of exhausting the steam from the cylinders into the stack through a fixed nozzle shows the following:

1 That the design of the front end definitely establishes at all rates the efficiency of combustion of the coal, as measured by the excess of air, and the efficiency of the cylinders except for some modification possible by the engineer by position of the throttle and cutoff.

2 That the front end does not increase the rate of supply of air proportionately to the rate of output of the locomotive but leads to a decreasing excess or even a deficiency of air at high rates.

3 That the loss of heat in cinders carried out of the combustion chamber at high rates is not necessarily the result of the deficiency of air supply by the front end.

4 That the high back pressures on the cylinders at high rates lead to large losses of power output.

5 That one of the principal reasons why high back pressures are required is the high resistance offered to gas flow by the diaphragm and the tubes.

6 That most attempts at manually or automatically controlled front ends have been based on the assumption that locomotives are overdrafted at high rates of output and underdrafted at low rates, whereas the reverse is true.

A program of research to improve the performance of the coal-fired steam locomotive by improvement of the air-supply system should include the following:

1 A program of tests on a full-scale locomotive to fix definitely the value of the factors, ratio of area of stack to nozzle, steam pressure, temperature of the gas, and resistance to gas flow that determine the entrainment ratio of a nozzle.

This information can be obtained on standing tests using throttled desuperheated steam. Of particular importance is that the weight of gases moved be accurately measured. This is preferably obtained by movement of the gases with a fan and measurement by means of a nozzle or a Pitot tube.

With this information and with knowledge of the resistance to gas flow and of the consumption of steam of a given locomotive, the proper size of nozzle for a given performance could be established by calculation, and cut-and-try methods of adjustment of the front end would be eliminated.

2 Review of design of front ends and experimental research to develop a design with the minimum resistance to the flow of gas consistent with proper self-cleaning characteristics.

3 Review of the problem of heat transfer in the boiler tubes to fix the area and length to make the over-all efficiency of the locomotive an optimum.

4 Conduct a thorough program of experimental research to determine the value of overfire air in elimination of smoke, reduction of losses in gaseous combustible, and of losses in cinders.

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Discussion

E. D. Benton.[4] While investigations and testing of various combinations and arrangement of front-end parts have been carried on for many years, no one, heretofore, has clearly set forth the principles so that the ability of the front end to move air could be rationalized. Had this been done previously many of the misconceptions of its inherent characteristics would have been known, and much time and effort saved in attempts to design and patent ideas which were doomed to disappointment. The apparently simple fact that steam-air entrainment ratios decrease with either an increase in steam flow at the nozzle or increase in draft, and therefore at maximum output there is no surplus steam exhausted, has not been appreciated except by a few people. This has resulted in many attempts to improve the front end by the application of principles which, in effect, did just the reverse.

The development of a simple formula for calculating entrainment ratios, based upon the fact that both the air rate per Btu in the coal and boiler-absorption efficiency are essentially constant thus leaving only one variable, that of total heat or enthalpy of the steam, is particularly valuable. It, of course, assumes that all of the steam generated is exhausted through the nozzle or absorbed by the boiler feedwater. However, a sizable percentage of the steam generated by the boiler does not pass through the exhaust nozzle, such as the heat lost from continuous boiler blowdown, the steam-air pump, cab and headlight generator, stoker engine and jets, train heating, etc. Obviously, a factor representing this should be included in Equation [6].

The paper, by inference, brings out an interesting but seldom appreciated point, that exhaust pressures must be higher for modern high-pressure high-superheat engines than for older or saturated-steam engines. Assume two engines with the best possible front-end arrangement, but one engine producing steam having an enthalpy of 1200 Btu per lb, and the other engines with steam conditions giving 1395 Btu per lb. Then the engine with the greater enthalpy must entrain $\frac{1395 - 1200}{1200} \times 100$ or 16 per cent more air per lb of steam exhausted. If Young, in his investigations at the University of Illinois, was able to measure a difference of only 12 per cent between the best and the worst front-end nozzle-stack arrangement, it becomes apparent that the engine having the highest enthalpy must have a well-designed front end, and particular attention must be given the resistance offered to the flow of gases from the ashpan to the front end. It would appear that Young's statement, "the mutual incompatability of the mechanical requirements of the front end," has another difficulty, i.e., that of enthalpy.

The concept of resistance and resistance ratios as presented in the paper should prove helpful to anyone experiencing drafting difficulties, particularly when attempting to compare front-end proportions between engines having substantial differences in enthalpy of the steam generated. That grate and fuel-bed resistance is only about 15 per cent of the total will surprise many stationary-power-plant men who usually consider the fuel bed as the major resistance to the flow of air. It is for this reason that engines appear to steam equally well with a fuel and ash bed free of clinkers of 14 in. in thickness as with one only 6 in. thick.

If the steam turbine is to enjoy the success it justly deserves, considerable attention must be given the optimum "resistance train" in order to lower back pressure and improve water rates, since an increase in enthalpy of the steam is handicapped by an increase in exhaust pressure.

J. R. Jackson.[5] In the main, the writer follows the analysis and subscribes to the points advanced by the authors, particularly that "draft," as expressed in pressure differential, is not the proper basis for the study of the flow of heat through the locomotive boiler, and that the composition, weight, and temperature of the gases moved, at various locations in progression through the boiler, should be determined for different rates of power development. It is agreed with that in this respect there is a lack of information and a field for investigation which should yield a better understanding of the factors in the burning of solid fuels on grates throughout the normal range of power development of a steam locomotive boiler and possibly lead to an extension of the economic range over that heretofore obtainable.

However, there are two points made by the authors which the writer finds it difficult to subscribe to:

1 The evaluation of the resistances to flow through the boiler, particularly to the relatively small value attributed to the fuel bed.

2 The statement of "the correct principles of automatic control of air supply."

As to the first point, fire-bed resistance; the front-end plates and spark-arrester appliances, the tubes and flues, the arch, the grates, and the ashpan openings are the fixed resistances; the fire bed is the variable resistance and consequently the only means the locomotive fireman has to regulate and control air supply. While fire-bed resistance as the authors show, is of a relatively small magnitude as compared with the fixed resistances, it is a most important factor in the control of steam generation for any given rate of power development in any existing coal-

[4] Fuel Engineer, Louisville and Nashville Railroad Company, Louisville, Ky. Mem. A.S.M.E.

[5] Engineer of Tests, Missouri Pacific Railroad, St. Louis, Mo. Mem. A.S.M.E.

fired steam locomotive. The difference between a good fireman and a poor one is essentially the manner in which he regulates fire-bed resistance to control air supply.

As to point 2, i.e., automatic control of air supply. If by "correct size to give the required output of the locomotive" the authors mean maximum output, a nozzle of relatively large area is essential if the cylinder back-pressure horsepower loss is to be kept at the desired minimum for those conditions. However, in practice, with the relatively large nozzle desired for maximum output, it is too large for the lower rates of power development and the locomotive will "steam hard" or "fail for steam" as the rate of flow of exhaust steam through the nozzle is decreased by throttle or cutoff control. The universal remedy followed to improve steaming at the lighter loads under the conditions as just set forth is to reduce the nozzle area. This is directly the opposite procedure to the authors' statement of the correct principles of automatic control of the air supply and wherein the writer finds it difficult to agree. The case of a relatively large-nozzled locomotive in effect approaches the authors' concept for the automatic control of air supply, and yet it does not work out in practice.

It has been realized for many years that the back-pressure horsepower loss in the reciprocating steam locomotive is relatively high at the higher rates of power development. The writer has had considerable experience in attempts to bring about improvement in this direction and still believes that this angle requires further study and experimentation in any program of research looking to the improvement of the conventional reciprocating steam locomotive.

The control of air flow through the grates or above the fire in a coal-fired steam locomotive, throughout the relatively wide range of power development, is not a simple problem but it is believed that the approach, as suggested by the authors in this paper, should yield long-needed supplementary information and gives promise of realizing some worth-while improvements in the efficiency of the steam locomotive within the relatively narrow thermal-efficiency limitations attainable in this type of motive power.

In conclusion, the writer is thoroughly in sympathy with the authors' recommendations for a program of research employing a full-sized locomotive and hopes that it will be carried out along the general line as set forth in this paper.

R. R. McIntosh[6] and W. H. Clegg.[7] We note in the section, "Relation of Air Supplied by Front End to Air Required for Coal Fired," the authors refer to a locomotive developing 3500 dhp, at which output there was a deficiency of 30 per cent in excess air required for the coal fired, while actually for the coal burned, there was still an excess of 28 per cent at that power output. This fact seems to offer the most lucrative field for research in correcting the factors leading to such a tremendous loss in unburned fuel.

It is also noted in "Relation of Performance of Front End to Loss of Heat in Cinders," that the authors conclude that the high cinder loss is the result of the high carrying power of the gases for solids at the high velocity with which they move through the combustion chamber at high rates of operation, and that coal is both carried out of the bed and swept out of the stream of coal being fired before it reaches the fuel bed.

It is apparent that these undesirable conditions exist at high rates of output where they are influenced by the higher back pressure developed, which rapidly increases the ratio of the loss of back-pressure horsepower to the drawbar-horsepower output, as illustrated in Fig. 11. The small percentage of increase of air flow obtained by large increases in back pressure is obtained only at a very high price and is entirely out of proportion. It is therefore reasonable to assume that much may be accomplished in this direction where both high values of back pressure and air flow are automatically controlled, especially where such permits the locomotive to develop its boiler and tractive-effort capacity by reducing materially the ratio of back pressure-horsepower loss to drawbar-horsepower output and maintaining a more satisfactory range of entrainment ratio in the movement of gases.

We heartily concur in the authors' conclusions and recommendations for a program of research which should embrace a careful study of past and present designs of air-flow control, and the effect of unburned fuel loss by the control of back pressure, especially in view of the high ratio of back pressure - horsepower loss to drawbar-horsepower output. It is obvious from the authors' conclusions that they regard high back pressure as a very important factor which should be minimized by reduction of resistance to air flow, and we believe this to be well within the bounds of possibility with automatic control.

[6] Chief Draftsman, Grand Trunk Western Railroad Company, Battle Creek, Mich.

[7] General Supt. of Motive Power and Car Equipment, Grand Trunk Western Railroad Company, Battle Creek, Mich. Mem. A.S.M.E.

Authors' Closure

Mr. Benton has correctly pointed out that an increase in enthalpy of steam, by increase in pressure or in superheat, or both, results in an increase in the required entrainment ratio of the front end. His conclusion that this means that an engine having higher enthalpy must have a well-designed front end is also sound as there is little purpose in increasing the energy in the steam at the high level if one must decrease the size of the nozzle and thus throw the energy away in increased back pressure.

Mr. Jackson questions the conclusion that the resistance of the fuel bed plays a small part in control of the flow of air through the system. His belief that the fireman can control the amount of air by the thickness of the bed is natural and it is true that if the pressure drop through the fuel bed is increased, the supply of air will decrease, but it is not true that it will decrease greatly for normally expected changes in resistance.

If we take, for an example, the conditions shown in Table 2, when the pressure drop across the entire system was 20.1 inches of water and that across the fuel bed was 3 inches, and assume that the drop across the bed is doubled, then the increase in resistance will be

$$\frac{\sqrt{23.1}}{\sqrt{20.1}} = 1.07$$

or seven per cent. That is, the air flow would be decreased by seven per cent, certainly not a large amount, for a doubling of the pressure drop across the fuel bed. The authors will certainly agree with Mr. Jackson that the difference between a good fireman and a poor one is the manner in which he maintains the uniformity of his fuel bed. For example, if the fireman fires too much coal and greatly increases the thickness of the bed, the result may be heavy smoke and loss due to carbon monoxide and unburned volatiles, not because the increased thickness of bed reduced the amount of air appreciably but because it changed the air-coal ratio by too much coal.

Conversely, a decreased rate of firing will lead to a thin fuel bed, low CO_2, and falling steam pressure, not because the air has increased a great deal but because the coal has decreased.

Mr. Jackson questions whether the authors mean maximum output in the statement, "correct size to give the required output

of the locomotive". The answer is that they mean the maximum output desired for the given locomotive, although this is usually the maximum that can be obtained. The authors recognize that as the size of the nozzle is decreased to give a higher entrainment ratio and more air, the cylinder output decreases because of the increased back pressure and the efficiency of the engine will decrease. Hence there should be an optimum size of nozzle for maximum output, but often a higher output can be obtained even at lower efficiency and this sacrifice is made.

The authors confess to lack of practical locomotive experience, but fail to see why, if a given nozzle gives satisfactory operation at high outputs, there should be "hard steaming" at lower outputs. The mathematical relations, fully confirmed by test data, show that the entrainment ratio of a nozzle increases as the flow of steam and thus the back pressure decreases. The "hard steaming" cannot be for lack of air. Could it be from too great an excess?

Messrs. McIntosh and Clegg appear to be in full accord with the conclusions of the paper and suggest that it is possible to obtain automatic control to avoid back pressures. The authors agree as to desirability, but point out again that the method of by-passing steam is wrong in principle because, when more air is needed, it can definitely not be obtained by not passing part of the steam through the nozzle.

Analysis of Special Electronic-Tube Tests

By J. H. Campbell[1] and C. G. Donsbach,[2] Schenectady, N. Y.

The complexity of manufacturing electronic tubes is such that professional groups normally associated with design and laboratory activities are called upon to work in the factory. These groups contribute considerably to the control of material assembly, test, and the outgoing quality of the product. In this industry many variables exist and because none of the tube parts can be tested electrically until the tube is completed, a system of "statistical quality control" is in order. This paper illustrates improvements which may be accomplished through special tests on tubes in the course of production, followed by statistical analysis of the results.

The complexity of the manufacture of many articles has been greatly increased in the past few years due to at least one or more of the following conditions:

1 The large number of new devices.
2 The relatively short period of time allowed for development.
3 The large quantities on order.
4 The precise quality and limitations imposed by critical wartime requirements.

There are probably few manufacturing processes more complex than the manufacture of electronic tubes. In order to cope with the situation many professional groups, normally associated with design and laboratory activities, must be called upon to work in the factory. The regular complement of production engineers in an electronic-tube factory must have the assistance of chemical and metallurgical engineers as well as glass technologists. This technical help in manufacturing contributes considerably to the control of material assembly, test, and the outgoing quality of the product.

Another important tool which is gaining in popularity and use in most industries is statistical quality control. In the electronic-tube industry the need for control of the many variables is especially acute since none of the parts can be tested for electrical characteristics until the tube is completely made. These variables start from the material composition of the parts and continue through all processing stations to the final test for electrical characteristics. When properly applied to manufacturing, statistical analysis becomes a library of information to help engineers, foremen, and operators make the right decision at the right time.

This paper is intended to illustrate what can be accomplished through the use of special tests in the production of electronic tubes, with particular emphasis on the need for statistical analysis of the results.

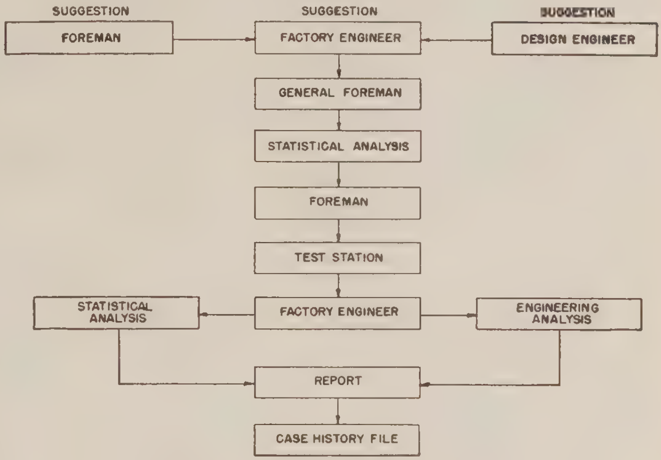

Fig. 1 Start and Finish of Special Test

"Special Tests" for Electronic Tubes

In order to avoid confusion in the term "special test," which will be referred to quite often, the following explanation is given:

The making of electronic tubes like many other manufacturing processes is accomplished by means of standard instructions. These instructions are the best known method of producing a particular tube type at the time the tube is considered developed and ready for manufacture. A special test usually has one variable which is not within the standard instructions. The engineer or person who suggests the test believes the one variable introduced in the manufacture of the tube will improve the quality and/or reduce manufacturing losses. If the test accomplishes either or both of the improvements, the corresponding change is made in the standard instructions.

It is therefore obvious that the special test is one of the most important tools for the factory to improve the quality of its product or reduce the losses in manufacture.

In order to utilize the test method to its fullest extent, complete co-ordination of all the groups in the factory must be obtained. Fig. 1 shows how this is accomplished. In general, the three groups who initiate a test are the engineers in the factory, the foremen, and the design engineers. The factory engineer's office serves as a clearing house for all technical information and the engineer accepts or rejects a special test based on the value of the information he expects may be gained by conducting it.

When a test is approved from a technical viewpoint, the general foreman is consulted regarding the priority the test should have in the production schedule. The written test form is then studied by the statistician to determine whether the method to be used will properly isolate the one variable being tested. The foreman then receives complete instructions and the tubes are assembled and tested. It then becomes the joint responsibility of the engineer and the statistician to analyze the data to determine whether the results are significant enough to require a change in standard instructions. A case history is then kept for future reference. The data are arranged by tube type and subject and made available to all who may have occasion to use them. When

[1] Factory Engineer, General Electric Company, Schenectady Tube Works.
[2] General Electric Company, Schenectady Tube Works.

Presented at the Spring Meeting of the Metropolitan Section, New York, N. Y., May 8, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

this file is properly kept it serves to avoid duplication of effort as well as to provide information on past experience to help solve difficult problems in the future.

Production Flow Chart

To give an idea as to the number of variables encountered in the making of a tube, a typical flow chart of production is shown

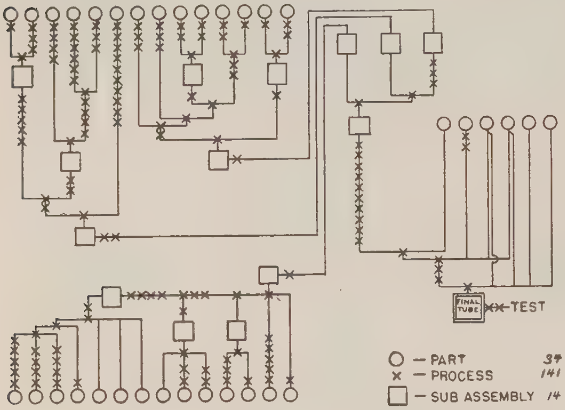

Fig. 2 Flow Chart Showing Possible Variables

in Fig. 2. It will be noted that this chart contains 34 parts, 141 manufacturing processes, and 14 subassemblies. In the manufacturing processes we have not included on the chart any subprocesses such as cleaning schedules of parts, exhaust schedules, etc. Likewise, Fig. 2 does not indicate the number of different machines, or operators, or test sets that may be in use at any one time. For ease of illustration these additional variables were omitted from the flow chart. It is important to note that practically any number of these variables or combinations of them may affect one or more of the electrical characteristics of the tube. The decision as to the total possible number of variables which can be encountered will be left to the reader's imagination.

Control over these variables is maintained statistically in many ways, a few of which may be listed as follows:

(*a*) Control over incoming material by sampling procedures.

(*b*) Control over processing stations by per cent defective, and variables control charts based upon 100 per cent and/or sampling inspection.

(*c*) Control at the test station by means of per cent defective, and variables control charts based upon 100 per cent and/or sampling tests. This may mean control over previous processing procedures since the results at test are, in many cases, directly or indirectly related to processing.

(*d*) Control by means of special tests.

In the remainder of the paper we will point out the method employed in analyzing special tests and some of the results obtained by the use of this system in an electronic-tube plant.

Procedures Followed in Special Tests

Fig. 3 is a flow chart which will give the general basis for the instigation, analysis, and disposition of such tests. An explanation of this chart follows:

The various reasons for running a test may arise from four general sources: 1 It may be the suggestion of a production worker, a foreman, or an engineer to change a certain processing procedure in the hope of improving the quality of the tube. 2 If a rise occurs in the per cent of defective tubes, it is the engineer's responsibility to isolate and correct the cause. This is often done by means of a special test where the suspected cause is the only variable. These troubles may in some cases be outstanding, as when the tubes suddenly become defective, or when per cent defective and/or variables control charts indicate trouble. 3 Another cause for which a special test may be run is to determine whether any differences exist among processing procedures such as among operators or machines. This source also uses special tests to compare test sets. 4 A special test may result from a necessity of control over certain incoming materials such as base metals, filaments, grids, etc., or from necessity of material control over certain processing materials, such as, emission-coating mixes, etc.

In general, regardless of the reason for conducting a special test, the procedure is to manufacture a small sample of tubes under prescribed conditions. Because of the large number of variables encountered it is absolutely necessary to send a control test along with the special test so that conclusions can be studied

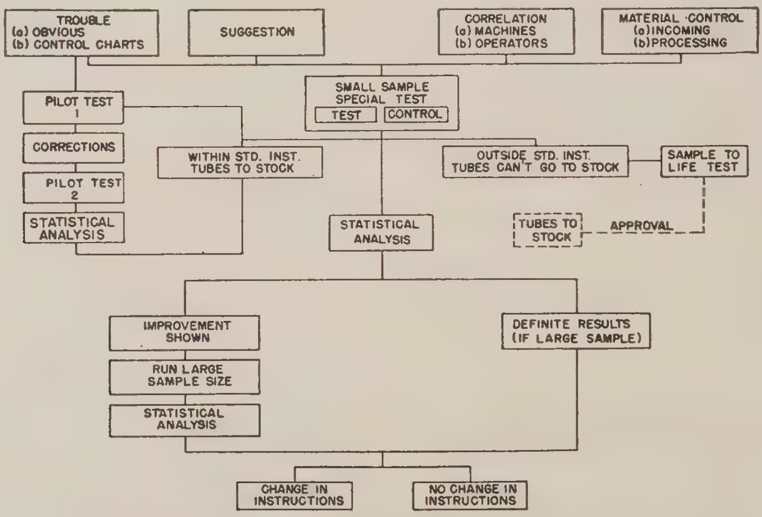

Fig 3 Flow Chart Showing Instigation and Procedures Involved in Special Tests

properly. This is the means used for reducing experimental error by holding constant all factors except the one under investigation. It means that the control test must be processed by exactly the same machines and same conditions (except for the one variable being tested) as the special test. Unless this is done, the results of the special test are not indicative.

These special tests may be further classified into two catagories, i.e., those in which the procedures are within standing instructions, and those which are outside standing instructions. In the case of the former the tubes may go directly to stock, but where the test is for tubes made under procedures which are outside standing instructions, the tubes cannot be stocked until the quality is determined by means of a life test or equivalent on a reasonable sample of the lot. If the tubes pass the additional prescribed test they may then go to stock.

As soon as the tubes are tested the results of the special test are compared with those of the control test. It is in this comparison that the use of the statistical analysis is applied. Fig. 4 shows the various statistical tests applied to determine whether the special-test results are different from normal production (control test). A brief explanation of each of these tests will be given, but for the basic theory the reader is referred to the Bibliography of this paper.

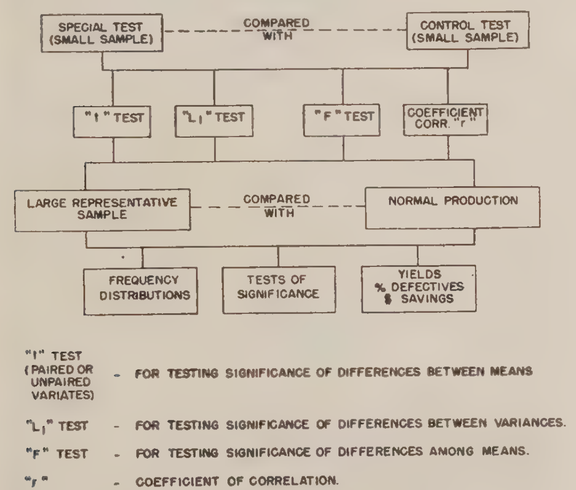

Fig. 4 Flow Chart Showing Methods of Statistical Analysis

Statistical Comparison of Test Results

In the statistical comparison of test results the usual procedure is to figure the mean and standard deviation of the individual electrical characteristics for both the special and control tests, and then be sure that a controlled process exists. The question which then arises is, "Are these means and variations statistically different between the special and the control tests?" In order to determine this, various statistical tests of significance are applied. One of these is the t test. This determines on the basis of the sample size whether the "averages" are significantly different from each other. The t test may be subdivided into two categories, i.e., that of paired, and that of unpaired variates. In all cases where control tests are run along with special tests, it is possible only to use the t test for unpaired variates. In other cases where partial special tests are run without control tests, such as in checking test equipment, the t test for paired variates is used. This is illustrated in Example 4.

The L_1 test is used to determine the significance of differences in "variance" between the special and control tests. The methods of computing the values of the t and L_1 are shown in examples 1 and 4, and the significance is determined from tables prepared for the purpose.[3]

Where there are more than two means and/or variances to be compared, the F test is generally used. Example 3 is an illustration of the use of this F test. Where correlation studies are indicated, the coefficient of correlation and line of regression are computed.

If the results of the special tests show fairly significant differences or are borderline cases, it is usually the procedure to send through a larger representative sample; perhaps a day's production or more. The control for this test is actually normal production before and after the large sample is run. In running this larger test care is taken to see that all parts and processes are randomized as is normal production. If three machines and three operators normally work on one of the tube processes, then this test is run so that the tubes are randomized among these machines and operators as they are in normal production.

After the tubes are tested for electrical characteristics, the results are statistically compared to normal production. The method of best presenting the results of this comparison has been found to be by cumulative frequency distributions, sometimes plotted on probability paper. This not only gives the shape of each distribution but enables one to estimate the average and standard deviation as well as per cent defective. This type of comparison is illustrated in examples 1 and 2. Where it is thought necessary, statistical tests of significance are also applied to these large sample sizes. However, in most cases a study of the curves together with the small-sample analysis are sufficient to determine whether or not to change normal production over to the new procedure.

Pilot Tests

As mentioned previously, there are instances where special tests are run in about the same manner as normal production. The cause for running this test (which is called a "pilot test") is to determine if the yield of percentage of good tubes can be increased. The principle used is first to observe every process in the production of the tube. During this observation period referred to as pilot test 1, no corrections are made but are merely noted. The results of the yield of this are tabulated with all possible discrepancies. Then pilot test 2 is run with all corrections applied. The corrections are decided upon by engineers who consult with foremen and test observers.

Fig. 5 shows the yield of a certain tube (not yet out of the developmental stage) plotted over a period of time. Note the yield of pilot test 1 and then note the upward trend resulting as corrections were applied. Pilot test 2 showed almost a 71 per cent increase in yield over pilot test 1. The yield on this tube has continued along the yield obtained by pilot test 2.

The examples given in the appendix are taken from case histories of recent special tests. They have been selected as representative cases of actual production and are believed to be self-explanatory. The number of these tests being run and the results obtained from them can be illustrated by the following figures for April, 1945:

Number of tests started	50
Number finished	34
Number of those completed showing improvements	16

The program, as explained in this paper, is being followed actively at the present time. It is helping to produce the highest-quality product. In the postwar period the ultimate consumer will derive benefits in the form of increasingly good quality.

[3] Tables are available in various texts on the subject. (Refer to Bibliography following.)

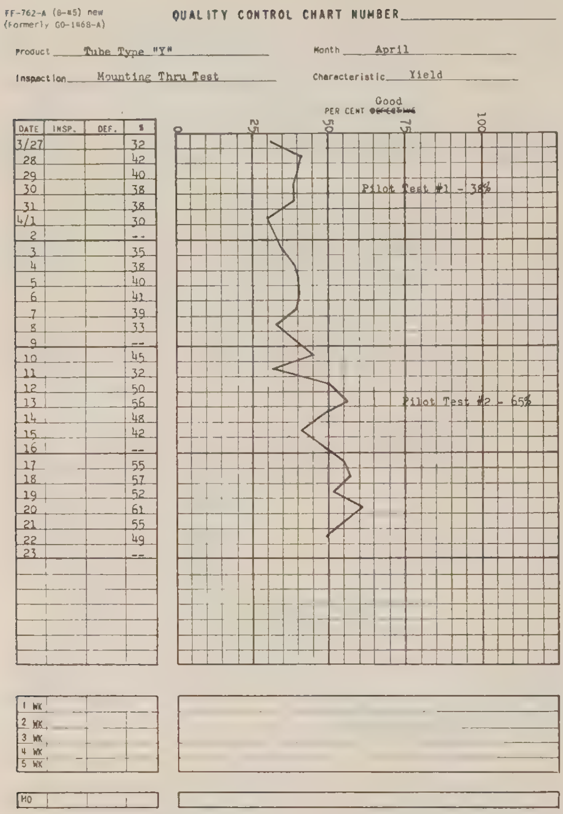

FIG. 5 SAMPLE QUALITY-CONTROL CHART

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Appendix

EXAMPLE 1 SPECIAL TEST RUN AS RESULT OF SUGGESTION TO IMPROVE QUALITY

On one tube type the electrical characteristic "grid current" was the cause for a fairly large percentage of loss. The engineer decided that a change in the method of assembly would narrow the tube-to-tube variation and eliminate a considerable number of electrical defects.

After considering the daily production schedule and the number of possible variables present, it was decided to send through a special test of 50 tubes together with a control test of 50 tubes. For the control test, every part used came from the same lot of material and all processing operations were made by the same operators and on the same machines, under the same conditions as the special test, except for the one variable being tested.

The following results were tabulated for the grid characteristic:

	Test	Control
Sample size (n)	50	50
Average ($\bar{X}$), ma	37.4	42.0
Standard deviation (σ), ma	4.2	7.1

Were these results statistically different? The statistical t test was applied to determine the significance of the difference between the averages, and the L_1 test was applied to determine whether this difference in averages was due to inequality in variances.

For the t test, applying the formula

$$t = \frac{d}{\sqrt{\dfrac{\Sigma(X_t - \bar{X}_t)^2 + \Sigma(X_c - \bar{X}_c)^2}{n_t + n_c - 2}\left(\dfrac{1}{n_t} + \dfrac{1}{n_c}\right)}}$$

we have

$$t = \frac{4.6}{\sqrt{\dfrac{890 + 2518}{50+50-2}\left(\dfrac{1}{50} + \dfrac{1}{50}\right)}} = 3.9$$

which for 98 deg of freedom is significant.[4]

Applying the formula for the L_1 test

$$L_1 = \frac{s_t^2 s_c^2}{s_a^2 s_a^2}$$

where

$$s_a^2 = {}^1/_2(s_t^2 + s_c^2)$$

we have

$$L_1 = \frac{(17.8)(50.4)}{(34.1)(34.1)} = 0.78$$

$$+ s = {}^1/_2(17.8 + 50.4)$$

The value of L_1 for sample sizes of 50 is 0.9612, beyond which lie 5 per cent of all values of L_1. The value 0.78+ is far below this and indicates that the variances of the two assumed normal populations from which these samples were drawn differ significantly.

Because of the results in the special test of 50 tubes, it was decided to obtain a larger representative sample by running at least a day's production with the new assembly technique. In doing this the day's production was run with all shifts, all machines, and other variables that would be encountered in normal production. The distribution curves plotted on probability paper, Fig. 6, will give approximately the whole comparison as follows:

	Test	Normal production
n	309	500
$\bar{X}$	36.8	44.1
σ	4.5	8.0
Per cent defective	1.0	14.0

The actual yield (per cent good from starting 100 assemblies) showed an increase from 74 to 83. This increase in yield represents a monthly saving of $2500 to $3000, based upon the current production schedules.

[4] Degrees of freedom represent the number of independent values in the comparison.

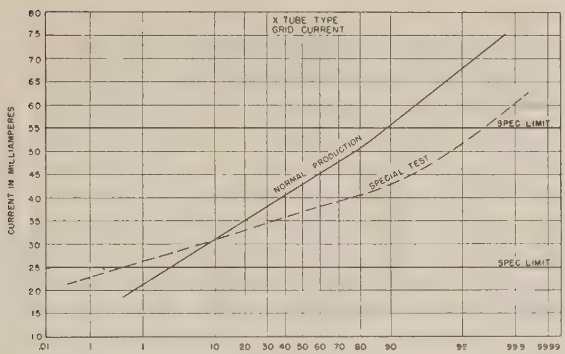


FIG. 6 X-TUBE TYPE GRID CURRENT

The following nomenclature is used in this section:

d = difference between means
X_t = individual value of each special-test reading
X_c = individual value of each control-test reading
$\overline{X}_t$ = average of special-test readings
$\overline{X}_c$ = average of control-test readings
n_t = sample size of special test
n_c = sample size of control test
s_t^2 = variance of special test
s_c^2 = variance of control test
σ = standard deviation

EXAMPLE 2 SPECIAL TEST RUN TO REDUCE HIGH PER CENT DEFECTIVE AS SHOWN BY STATISTICAL CHARTS

On a very critical tube type the per cent defective chart, Fig. 7, indicated a large loss at test due to a plate-voltage characteristic. This characteristic is primarily determined by the relative spacing of certain elements of the tube, which had previously been done by mechanical means. The engineer decided an improvement could be obtained by spacing the tube electrically with a capacitance bridge. This equipment was built for the test and arranged so that a predetermined reading of capacitance could be obtained while the tube was being assembled (the correlation of capacitance and spacing is shown in Fig. 8).

In using this fixture a small lot of tubes was made and the analyzed results indicated so much improvement that a special test was run on 420 tubes. These tubes were made on a completely randomized basis so that a correct comparison could be made with normal production.

Cumulative frequency distributions were plotted and are shown in Fig. 9. The results in the form of per cent defective are tabulated below:

	Normal production	Special test
Plate voltage	15.0	0.5
Capacitance	10.0	0.8
Gassy	3.7	1.4
Shorts	6.6	1.7

The reduction in gassy and shorted tubes was due to the more uniform spacing.

Statistical tests confirmed the differences in plate voltage and capacitance to be significant.

The device was put into use and the control chart in Fig. 7 shows the results in improvement in losses and quality. The over-all yield on this tube increased 49 per cent due to this factor, and the dollar savings amounted to over $7500 per 1000 tubes to stock.

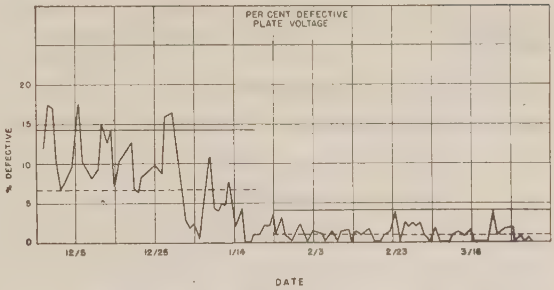


FIG. 7 PER CENT DEFECTIVE PLATE VOLTAGE

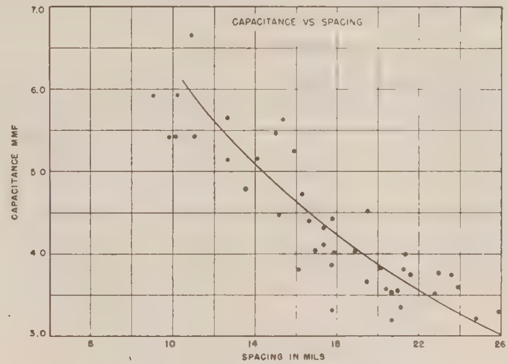


FIG. 8 CAPACITANCE VERSUS SPACING

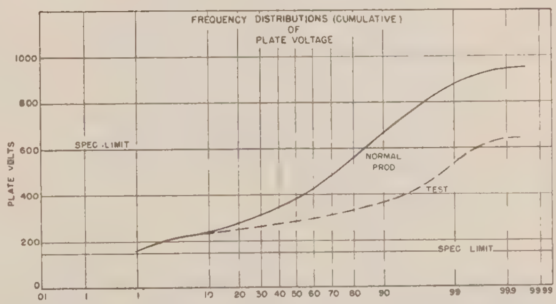


FIG. 9 FREQUENCY DISTRIBUTIONS, CUMULATIVE, OF PLATE VOLTAGE

EXAMPLE 3 COMPARISON OF EXHAUST SCHEDULES THROUGH USE OF L_1 AND F TESTS

Considerable difficulty was encountered in obtaining proper emission on a particular tube type. One of the possible variables which could affect this was the schedule by which the tubes were exhausted.

The engineer ran a special test using four different exhaust schedules. The following results were tabulated:

Exhaust schedule	A	B	C	D
n	16	24	28	19
$\overline{X}$	28	20.2	27	21

From the averages given in the table, schedules B and D look much better than schedules A and C. However, do the averages of these schedules differ significantly among themselves, or do they most likely come from the same homogenous population?

In order to determine this the F test was applied (after applying the L_1 test which showed that the variances of the assumed normal populations from which these samples were drawn were not significantly different).

In the F test we simply segregate the total variances into the variances among schedule means, and the variances within schedule means. We compute the ratio and compare it to the known distribution of F ratio for random samples. In other words, "we compute the ratio of an estimate associated with a suspected 'cause' to the estimate which best defines the error of the experiment. If the probability is small, say, less than 0.05, that this ratio could have occurred by chance in sampling from normal populations of identical means and variances, the hypothesis that such were the populations is rejected."

The following shows the computation of F for the comparison of exhaust schedules:

Source of variation	Sum of squares	Degrees of freedom	Mean square
Among schedules..	$\Sigma(\bar{X}_c - \bar{X})^2$	$K - 1$	$\sigma_1^2 = \dfrac{\Sigma(\bar{X}_c - \bar{X})^2}{K-1}$
	873.7	3	291.2
Within schedules...	$\Sigma(X - \bar{X}_c)^2$	$n - K$	$\sigma_2^2 = \dfrac{\Sigma(X - \bar{X}_c)^2}{n-K}$
	9631.0	83	116.0
Total............	$\Sigma(X - \bar{X})^2$ 10504.7	$n - 1$ 86	

where

$$\Sigma(\bar{X} - \bar{X})^2 = \Sigma\bar{X}_c^2 - \frac{(\Sigma X)^2}{n}$$

$$51080.55 - 50206.9 = 873.7$$

$$\Sigma(X - \bar{X}_c)^2 = \Sigma X^2 - \Sigma\bar{X}_c^2$$

$$60711.50 - 51080.55 - 9631.0$$

$$\Sigma(X - \bar{X})^2 = \Sigma X^2 - \frac{(\Sigma X^2)}{n}$$

$$60711.50 - 50206.9 = 10504.7$$

$$\text{where } \Sigma\bar{X}_c^2 = \frac{(\Sigma X_1)^2}{n_1} + \frac{(\Sigma X_2)^2}{n_2} + \frac{(\Sigma X_3)^2}{n_3} + \frac{(\Sigma X_4)^2}{n_4}$$

$$F = \frac{291.2}{116.0} = 2.501$$

The table of values of F for 3 and 83 deg of freedom produces the ratio of 2.72. Since the value of 2.501 is less than this, we must take it as an indication that the exhaust schedules all come from the same homogenous population and therefore they had no significant effect on the electrical characteristic.

The meaning of terms is as follows:

X = individual value of electrical characteristic
n = sample size
K = number of samples

Example 4 Showing Control Over Test Equipment by Means of Correlation Between Test Sets

One of the methods of maintaining adequate control over test equipment is to check factory sets with standard laboratory sets for all electrical characteristics. Although this does not strictly adhere to the full procedure of running complete special tests in order to make these correlations, the actual testing procedure is considered in the category of a special test.

The usual method is to take a small sample of tubes, test them on the one set, and then test them on the standard set (using the same operator). The results are then analyzed for statistical significance between averages by means of the t test for paired variates. The following is a good example of what can be done with statistical tests of significance in such correlations on a certain tube type:

Electrical characteristic	Factory average	Laboratory average	Difference	Statistical significance
A	183.9	177.6	6.3	Significant
B	4.44	4.35	0.09	Significant
C	4.70	4.62	0.08	Significant
D	0.255	0.268	0.013	Not significant
E	4.28	4.31	0.03	Significant
F	2.11	2.00	0.11	Significant
G	0.322	0.307	0.015	Not significant
H	0.311	0.317	0.006	Not significant
I	0.573	0.423	0.114	Significant
J	0.533	0.428	0.105	Significant
K	21.1	21.5	0.4	Significant

Although many of these differences show statistical significance, only three of them were "practically" significant. The factory sets were recalibrated until these differences were not significant.

The t test of significance is illustrated for the "A" characteristic as follows:

$$t = \frac{\bar{d}}{\sigma/\sqrt{n}}$$

where

$$\sigma^2 = \frac{\Sigma d^2 - \dfrac{(\Sigma d)^2}{n}}{n-1}$$

where

$\bar{d}$ = difference between means
n = sample size
Σd^2 = sum of individual differences squared
$(\Sigma d)^2$ = square of sum of individual differences

$$t = \frac{6.3}{\sigma/\sqrt{21}} = 3.45$$

$$\sigma^2 = \frac{2237 - \dfrac{(133)^2}{21}}{20}$$

$$\sigma = 8.35$$

The Effect of Heat Loss on the Performance of Exchangers With Interconnected Walls

By P. R. TRUMPLER,[1] NEW YORK, N. Y.

Equations are derived for the calculation of exchangers in which the passage walls are bonded to each other, providing substantially uniform wall temperature over any cross section normal to the flow. Results are presented for both countercurrent and cocurrent exchangers and an example illustrating the solution is given.

Nomenclature

The following nomenclature is used in the paper:

h_s = thermal conductance, based on surface A_s, between metal wall and the surroundings at temperature θ_s, Btu hr^{-1} ft^{-2} deg F^{-1}
A_s = reference surface for h_s, sq ft per ft of exchanger length
Q_s = heat flow from metal wall to surroundings, positive for heat flow to surroundings, Btu hr^{-1}
W = fluid-flow rate, lb hr^{-1}
c = specific heat at constant pressure, Btu lb^{-1} deg F^{-1}
h = fluid film conductance, Btu hr^{-1} ft^{-2} deg F^{-1}
A = surface washed by fluid, sq ft per ft of exchanger length
θ_s = temperature of surroundings, deg F
θ = temperature of warm fluid, deg F
T = temperature of cold fluid, deg F
η = temperature of metal wall, deg F
z = length co-ordinate, parallel to fluid-flow directions, positive in direction of flow of cold fluid, ft
L = exchanger length, ft

Subscripts T and θ refer to the fluids having those temperatures
Subscript 0 refers to value at $z = 0$
Subscript 1 refers to value at $z = L$

Introduction

Under certain conditions heat transfer between an exchanger and its surroundings can have an important effect upon the performance of the exchanger. Even a small heat leak may appreciably alter the temperature difference between the two fluids as calculated for adiabatic performance. Heat-loss effects are often appreciable where a small temperature difference between the fluids is required by the necessity for high thermal efficiency in the process, or where the exchanger operates at a temperature far from the ambient. Examples are found in plants for the separation of gases by deep refrigeration, and at the other end of the temperature scale, in the gas-turbine recuperator.

For the case of a double-pipe exchanger the effect of heat loss has been investigated by Nesselmann.[2] The recent development of extended-surface exchangers, particularly for deep-refrigeration processes, has brought practical importance to another case, namely, exchangers of the general type shown in Fig. 1. For purposes of the present analysis this type is characterized by passages bonded together in such a manner that the wall-temperature variation over any cross section normal to the flow is small compared to the temperature difference between the wall and the surroundings.

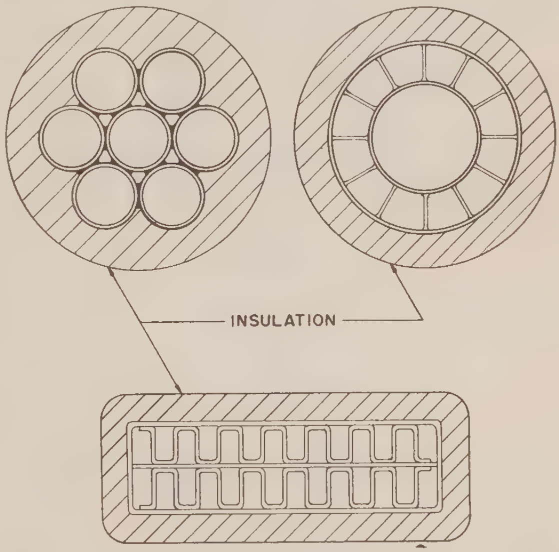


Fig. 1 Typical Heat Exchangers With Interconnected Walls

It will be seen that the results are applicable to exchangers in which the passages are not bonded together, provided the temperature difference between the various walls at a cross section is small. This is generally true of double-pipe or shell-and-tube exchangers in which either (1) the film coefficient of the shell fluid is very much higher than that of the tube fluid or (2) if the two fluids are exchanging heat at a small temperature difference.

Under what conditions should the effect of heat loss be investigated? The equations appear to yield no simple criterion. If, however, a rough estimate of the heat loss shows that the mean temperature difference based on terminal values is appreciably changed, a check for the effect of heat loss should be made.

The performance equations derived in the following section are limited only by the usual assumptions, which require that conductances and specific heats remain constant for the length of the exchanger, and that heat conduction in the metal or the insulation along the length of the exchanger is not important. The effect of heat conduction through fins, wall, bonding material, and fouling films may be taken into account by correcting the individual film coefficients.

Countercurrent Flow

The three basic equations are:

[1] Development Engineer, M. W. Kellogg Company. Jun. A.S.M.E.
[2] "The Influence of Heat Loss From Double-Tube Heat Exchangers," by K. Nesselmann, *Zeitschrift für die Gesamte Kälte-Industrie*, vol. 35, 1928, pp. 62–67.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

$$W_\theta c_\theta d\theta = W_T c_T dT + h_s(\eta - \theta_s)A_s dz \quad [1]$$

$$W_T c_T dT = h_T A_T(\eta - T)dz \quad [2]$$

$$W_\theta c_\theta d\theta = h_\theta A_\theta(\theta - \eta)dz \quad [3]$$

Eliminating η from Equations [2] and [3]

$$\frac{W_T c_T dT}{h_T A_T dz} + \frac{W_\theta c_\theta d\theta}{h_\theta A_\theta dz} = \theta - T \quad [4]$$

Eliminating η and z by Equations [3] and [4], Equation [1] becomes

$$\frac{W_\theta c_\theta}{W_T c_T}\left[1 + \frac{h_s A_s}{h_\theta A_\theta}\left(1 - \frac{\theta - \theta_s}{\theta - T}\right)\right] d\theta = \left[1 + \frac{h_s A_s}{h_T A_T}\left(\frac{\theta - \theta_s}{\theta - T}\right)\right] dT \quad [5]$$

The variables may be separated if we make the transformation

$$x \equiv \frac{\theta - \theta_s}{\theta - T} \qquad y \equiv \theta - T \quad [6]$$

$$xy = \theta - \theta_s$$

$$xdy + ydx = d\theta \qquad dT = (x - 1)dy + ydx \quad [7]$$

Substituting also

$$D \equiv \frac{W_\theta c_\theta}{W_T c_T} \quad [8]$$

Equation [5] becomes

$$\frac{dy}{y} = \frac{1 - D - D\dfrac{h_s A_s}{h_\theta A_\theta} + \left(\dfrac{h_s A_s}{h_T A_T} + D\dfrac{h_s A_s}{h_\theta A_\theta}\right)x}{1 + \left(D + D\dfrac{h_s A_s}{h_\theta A_\theta} + \dfrac{h_s A_s}{h_T A_T} - 1\right)x - \left(\dfrac{h_s A_s}{h_T A_T} + D\dfrac{h_s A_s}{h_\theta A_\theta}\right)x^2}\, dx \quad [9]$$

Using Equation [9] to eliminate y from Equation [4] we determine x as a function of z

$$\int_{x_0}^{x} \frac{dx}{1 + (a + D - 1)x - ax^2} = \int_0^z \frac{dz}{K} \quad [10]$$

where

$$\left.\begin{aligned} a &\equiv \frac{W_\theta c_\theta}{W_T c_T}\frac{h_s A_s}{h_\theta A_\theta} + \frac{h_s A_s}{h_T A_T} \\ K &\equiv W_\theta c_\theta\left[\frac{1}{h_\theta A_\theta} + \frac{1}{h_T A_T}\left(1 + \frac{h_s A_s}{h_\theta A_\theta}\right)\right] \end{aligned}\right\} \quad [11]$$

Integration yields

$$x = \frac{1}{2\left(\dfrac{D}{h_\theta A_\theta} + \dfrac{1}{h_T A_T}\right)} \left\{\frac{1}{h_T A_T} + \frac{D}{h_\theta A_\theta} + \frac{1}{h_s A_s}\left(D - 1 + r\frac{R_0 e^{(r/k)z} + 1}{R_0 e^{(r/k)z} - 1}\right)\right\} \quad [12]$$

where

$$\left.\begin{aligned} r &\equiv \sqrt{(D - 1 + a)^2 + 4a} \\ R_0 &\equiv \frac{-2ax_0 + (a + D - 1) - r}{-2ax_0 + (a + D - 1) + r} \end{aligned}\right\} \quad [13]$$

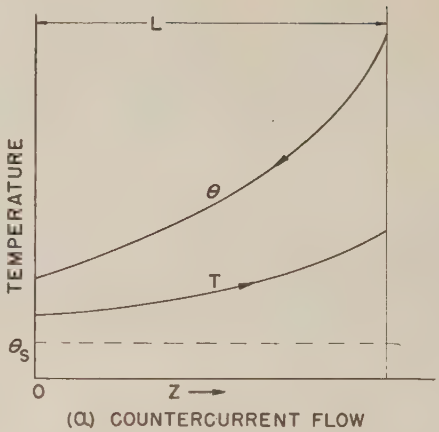


FIG. 2

Integrating Equation [9] and eliminating x by Equation [12] we obtain y as a function of z

$$\frac{y}{y_0} = me^{pz} + ne^{qz} \quad [14]$$

where

$$m \equiv \frac{1}{2r}\left\{r - D + 1 - h_s A_s(1 - 2x_0)\left(\frac{D}{h_\theta A_\theta} + \frac{1}{h_T A_T}\right)\right\}$$

$$n \equiv \frac{1}{2r}\left\{r + D - 1 + h_s A_s(1 - 2x_0)\left(\frac{D}{h_\theta A_\theta} + \frac{1}{h_T A_T}\right)\right\}$$

$$p \equiv \frac{U_\theta A_\theta}{2W_\theta c_\theta}\left\{r - D + 1 - h_s A_s\left(\frac{D}{h_\theta A_\theta} - \frac{1}{h_T A_T}\right)\right\}$$

$$q \equiv \frac{U_\theta A_\theta}{2W_\theta c_\theta}\left\{-r - D + 1 - h_s A_s\left(\frac{D}{h_\theta A_\theta} - \frac{1}{h_T A_T}\right)\right\}$$

$$\frac{1}{U_\theta A_\theta} \equiv \frac{1}{h_\theta A_\theta} + \frac{1}{h_T A_T}\left(1 + \frac{h_s A_s}{h_\theta A_\theta}\right)$$

$$r \equiv \sqrt{(D - 1 + a)^2 + 4a}$$

$$a \equiv h_s A_s\left(\frac{D}{h_\theta A_\theta} + \frac{1}{h_T A_T}\right)$$

$$D \equiv \frac{W_\theta c_\theta}{W_T c_T}$$

Equations [12] and [14] completely define the exchanger performance. Two special cases will be noted.

$h_s A_s = 0$

Then

$$a = 0 \qquad r = D - 1 \qquad n = 1$$

$$m = 0 \qquad p = 0$$

$$q = \frac{U_\theta A_\theta}{W_\theta c_\theta}\left(1 - \frac{W_\theta c_\theta}{W_T c_T}\right)$$

and

$$y = y_0 exp\left[\frac{U_\theta A_\theta}{W_\theta c_\theta}\left(1 - \frac{W_\theta c_\theta}{W_T c_T}\right)z\right] \quad [15]$$

which is the equation of the adiabatic exchanger.

$h_sA_s \to \infty$ with other film coefficients finite

$$a \to \infty \qquad r \to \infty$$

$$\frac{r}{h_sA_s} = \frac{D}{h_\theta A_\theta} + \frac{1}{h_TA_T}$$

and Equation [14] becomes

$$y = (\theta_0 - \theta_s)exp\left(\frac{h_\theta A_\theta}{W_\theta c_\theta}z\right) - (T_0 - \theta_s)exp\left(-\frac{h_TA_T}{W_Tc_T}z\right) \ldots [16]$$

Exchanger Length. The calculation of exchanger length is of prime importance to the heat-transfer engineer. The length may be determined by the following procedure:

From Equation [4]

$$\frac{W_Tc_T}{h_TA_T}\int_0^1 dT + \frac{W_\theta c_\theta}{h_\theta A_\theta}\int_0^1 d\theta = \int_0^L (\theta - T)dz \ldots [17]$$

or

$$\frac{W_Tc_T}{h_TA_T}(T_1 - T_0) + \frac{W_\theta c_\theta}{h_\theta A_\theta}(\theta_1 - \theta_0) = \int_0^L (\theta - T)dz \ldots [18]$$

Also

$$\int_0^L (\theta - T)dz = \int_0^L ydz = y_0\frac{m}{p}(e^{pL} - 1) + y_0\frac{n}{q}(e^{qL} - 1) \ldots [19]$$

Three of the terminal temperatures must be given in the statement of the problem. There are four possible combinations of the three temperatures, and the solution for each case will be given.

Case 1. Given θ_0, θ_1, and T_0:

From Equation [14]

$$\theta_1 - T_1 = y_0me^{pL} + y_0ne^{qL} \ldots\ldots\ldots\ldots [20]$$

Substituting Equations [19] and [20] into Equation [18]

$$\frac{W_Tc_T}{h_TA_T}\left(\frac{\theta_1 - T_0}{\theta_0 - T_0}\right) + \frac{W_\theta c_\theta}{h_\theta A_\theta}\left(\frac{\theta_1 - \theta_0}{\theta_0 - T_0}\right) + \frac{m}{p} + \frac{n}{q} = m\left(\frac{1}{p} + \frac{W_Tc_T}{h_TA_T}\right)e^{pL} + n\left(\frac{1}{q} + \frac{W_Tc_T}{h_TA_T}\right)e^{qL} \ldots\ldots [21]$$

Equation [21] fixes the value of L.

Case 2. Given θ_0, T_1, and T_0:

Substituting θ_1 from Equation [20], the solution is

$$\frac{W_Tc_T}{h_TA_T}\left(\frac{T_1 - T_0}{\theta_0 - T_0}\right) + \frac{W_\theta c_\theta}{h_\theta A_\theta}\left(\frac{T_1 - \theta_0}{\theta_0 - T_0}\right) + \frac{m}{p} + \frac{n}{q} = m\left(\frac{1}{p} - \frac{W_\theta c_\theta}{h_\theta A_\theta}\right)e^{pL} + n\left(\frac{1}{q} - \frac{W_\theta c_\theta}{h_\theta A_\theta}\right)e^{qL} \ldots\ldots [22]$$

Case 3. Given θ_0, θ_1, and T_1:

The solution may be readily obtained by changing the direction of the z-axis and interchanging subscripts 0 and 1 of the terminal temperatures. Note that this requires changing x_0 to x_1 in the equations defining m and n. The solution is

$$\frac{W_Tc_T}{h_TA_T}\left(\frac{\theta_0 - T_1}{\theta_1 - T_1}\right) + \frac{W_\theta c_\theta}{h_\theta A_\theta}\left(\frac{\theta_0 - \theta_1}{\theta_1 - T_1}\right) + \frac{m}{p} + \frac{n}{q} = m\left(+\frac{1}{p} + \frac{W_Tc_T}{h_TA_T}\right)e^{-pL} + n\left(+\frac{1}{q} + \frac{W_Tc_T}{h_TA_T}\right)e^{-qL} \ldots [23]$$

Case 4. Given θ_1, T_1, and T_0:

The solution is found as in Case 3

$$\frac{W_Tc_T}{h_TA_T}\left(\frac{T_0 - T_1}{\theta_1 - T_1}\right) + \frac{W_\theta c_\theta}{h_\theta A_\theta}\left(\frac{T_0 - \theta_1}{\theta_1 - T_1}\right) + \frac{m}{p} + \frac{n}{q} = m\left(+\frac{1}{p} - \frac{W_\theta c_\theta}{h_\theta A_\theta}\right)e^{-pL} + n\left(+\frac{1}{q} - \frac{W_\theta c_\theta}{h_\theta A_\theta}\right)e^{-qL} \ldots [24]$$

Calculation of Heat Loss:

Case 1. Given θ_0, θ_1, and T_0, or θ_0, T_1, and T_0:

T_1 or θ_1 may be calculated from y_1 where

$$y_1 = y_0me^{pL} + y_0ne^{qL}$$

Knowing the terminal temperatures, the heat loss may be calculated

Case 2. Given θ_0, T_1, θ_1, or T_0, T_1, θ_1

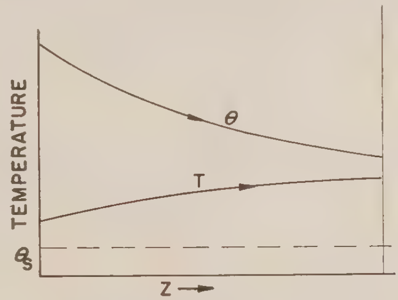
T_0 or θ_0 may be calculated from

$$\frac{\theta_0 - T_0}{\theta_1 - T_1} = me^{-pL} + ne^{-qL} \ldots\ldots\ldots\ldots [25]$$

Note that x_0 must be changed to x_1 in the equations defining m and n.

Parallel Flow

The three basic equations for parallel flow are



(b) PARALLEL FLOW

FIG. 3

$$-W_\theta c_\theta d\theta = W_Tc_TdT + h_s(\eta - \theta_s)A_sdz \ldots\ldots\ldots [26]$$

$$W_Tc_TdT = h_TA_T(\eta - T)dz \ldots\ldots\ldots\ldots\ldots\ldots [27]$$

$$-W_\theta c_\theta d\theta = h_\theta A_\theta(\theta - \eta)dz \ldots\ldots\ldots\ldots\ldots\ldots [28]$$

By a treatment similar to that for countercurrent flow the solution is found to be

$$x = \frac{1}{2\left(\frac{1}{h_TA_T} - \frac{D}{h_\theta A_\theta}\right)}\left\{\frac{1}{h_TA_T} - \frac{D}{h_\theta A_\theta} - \frac{1}{h_sA_s}\left(D + 1 + r_c\frac{R_{c0}e^{(r_c/k)z} + 1}{R_{c0}e^{(r_c/k)z} - 1}\right)\right\} \ldots [29]$$

where

$$r_c \equiv \sqrt{(D + 1 - a_c)^2 + 4a_c}$$

$$R_{c0} = \frac{2a_cx_0 + (D + 1 - a_c) - r_c}{2a_cx_0 + (D + 1 - a_c) + r_c}$$

$$a_c \equiv h_sA_s\left(\frac{1}{h_TA_T} - \frac{D}{h_\theta A_\theta}\right)$$

and

$$\frac{y}{y_0} = m_c e^{p_c z} + n_c e^{q_c z} \quad \text{................} [30]$$

where

$$m_c \equiv \frac{1}{2r_c} \left\{ r_c - D - 1 - h_s A_s (1 - 2x_0) \left(\frac{D}{h_\theta A_\theta} - \frac{1}{h_T A_T} \right) \right\}$$

$$n_c \equiv \frac{1}{2r_c} \left\{ r_c + D + 1 + h_s A_s (1 - 2x_0) \left(\frac{D}{h_\theta A_\theta} - \frac{1}{h_T A_T} \right) \right\}$$

$$p_c \equiv \frac{U_\theta A_\theta}{2W_\theta c_\theta} \left\{ r_c - D - 1 - h_s A_s \left(\frac{D}{h_\theta A_\theta} + \frac{1}{h_T A_T} \right) \right\}$$

$$q_c \equiv \frac{U_\theta A_\theta}{2W_\theta c_\theta} \left\{ - r_c - D - 1 - h_s A_s \left(\frac{D}{h_\theta A_\theta} + \frac{1}{h_T A_T} \right) \right\}$$

If $h_s A_s = 0$

$$y = y_0 \, exp \left[- \frac{U_\theta A_\theta}{W_\theta c_\theta} \left(1 + \frac{W_\theta c_\theta}{W_T c_T} \right) z \right] \quad \text{.......} [31]$$

If $h_s A_s \to \infty$

$$y = (\theta_0 - \theta_s) exp \left(- \frac{h_\theta A_\theta}{W_\theta c_\theta} z \right) - (T_0 - \theta_s) exp \left(- \frac{h_T A_T}{W_T c_T} z \right) [32]$$

Exchanger Length. The equation corresponding to Equation [18] is

$$\frac{W_T c_T}{h_T A_T} (T_1 - T_0) - \frac{W_\theta c_\theta}{h_\theta A_\theta} (\theta_1 - \theta_0) = \int_0^L (\theta - T) dz \quad \text{..} [33]$$

$$\int_0^L (\theta - T) dz = \int_0^L y dz = y_0 \frac{m_c}{p_c} (e^{p_c L} - 1)$$

$$+ y_0 \frac{n_c}{q_c} (e^{q_c L} - 1) \quad \text{........} [34]$$

Case 1. Given θ_0, θ_1, T_0:

$$\frac{W_T c_T}{h_T A_T} \left(\frac{\theta_1 - T_0}{\theta_0 - T_0} \right) - \frac{W_\theta c_\theta}{h_\theta A_\theta} \left(\frac{\theta_1 - \theta_0}{\theta_0 - T_0} \right) + \frac{m_c}{p_c} + \frac{n_c}{q_c}$$

$$= m_c \left(\frac{1}{p_c} + \frac{W_T c_T}{h_T A_T} \right) e^{p_c L} + n_c \left(\frac{1}{q_c} + \frac{W_T c_T}{h_T A_T} \right) e^{q_c L} \quad \text{.....} [35]$$

Case 2. Given θ_0, T_1, T_0:

$$\frac{W_T c_T}{h_T A_T} \left(\frac{T_1 - T_0}{\theta_0 - T_0} \right) - \frac{W_\theta c_\theta}{h_\theta A_\theta} \left(\frac{T_1 - \theta_0}{\theta_0 - T_0} \right) + \frac{m_c}{p_c} + \frac{n_c}{q_c}$$

$$= m_c \left(\frac{1}{p_c} + \frac{W_\theta c_\theta}{h_\theta A_\theta} \right) e^{p_c L} + n_c \left(\frac{1}{q_c} + \frac{W_\theta c_\theta}{h_\theta A_\theta} \right) e^{q_c L} \quad \text{.....} [36]$$

Case 3. Given θ_0, θ_1, T_1:

$$\frac{W_T c_T}{h_T A_T} \left(\frac{\theta_0 - T_1}{\theta_1 - T_1} \right) - \frac{W_\theta c_\theta}{h_\theta A_\theta} \left(\frac{\theta_0 - \theta_1}{\theta_1 - T_1} \right) + \frac{m_c}{p_c} + \frac{n_c}{q_c}$$

$$= m_c \left(+ \frac{1}{p_c} + \frac{W_T c_T}{h_T A_T} \right) e^{-p_c L} + n_c \left(+ \frac{1}{q_c} + \frac{W_T c_T}{h_T A_T} \right) e^{-q_c L} . [37]$$

Note that x_0 becomes x_1 in m_c and n_c.

Case 4. Given θ_1, T_1, T_0:

$$\frac{W_T c_T}{h_T A_T} \left(\frac{T_0 - T_1}{\theta_1 - T_1} \right) - \frac{W_\theta c_\theta}{h_\theta A_\theta} \left(\frac{T_0 - \theta_1}{\theta_1 - T_1} \right) + \frac{m_c}{p_c} + \frac{n_c}{q_c}$$

$$= m_c \left(+ \frac{1}{p_c} + \frac{W_\theta c_\theta}{h_\theta A_\theta} \right) e^{-p_c L} + n_c \left(+ \frac{1}{q_c} + \frac{W_\theta c_\theta}{h_\theta A_\theta} \right) e^{-q_c L} . [38]$$

Note that x_0 becomes x_1 in m_c and n_c.

Example

An exchanger of the type shown in Fig. 1 is employed to exchange heat between two air streams. Stream θ with a flow of 10,000 lb per hr is to be cooled from 1000 F to 350 F by 9500-lb per hr air entering at 300 F. Each air stream has a surface of 50 sq ft per ft of exchanger length and a corrected film coefficient of 20. The exchanger is lightly insulated, and the outside conductance, based upon a surface of 5 sq ft per ft of exchanger length, may be considered uniform at 1 Btu per hr per sq ft per deg F. Consider the specific heat of the air for each stream as constant and equal to 0.25 Btu per lb. Surroundings are at 100 F. Calculate the outlet temperature of the cold stream and the required exchanger length.

Solution. (Countercurrent exchanger)

$$D = \frac{10{,}000}{9500} = 1.053$$

$$h_s A_s = 5 \qquad x_0 = \frac{\theta_0 - \theta_s}{\theta_0 - T_0} = \frac{350 - 100}{350 - 300} = 5$$

$$h_\theta A_\theta = 1000$$

$$h_T A_T = 1000$$

$$a = 5 \left(\frac{1.053}{1000} + \frac{1}{1000} \right) = 0.010265$$

$$r = \sqrt{(0.053 + 0.010265)^2 + 0.04106} = 0.212$$

$$m = \frac{1}{0.424} \left\{ 0.212 - 0.053 - 5(1 - 10)(0.002053) \right\} = 0.593$$

$$n = \frac{1}{0.424} \left\{ 0.212 + 0.053 + 5(1 - 10)(0.002053) \right\} = 0.407$$

$$U_\theta A_\theta = \frac{1}{\dfrac{1}{1000} + \dfrac{1}{1000} \left(1 + \dfrac{5}{1000} \right)} = 498$$

$$p = \frac{498}{2 \times 10{,}000 \times 0.25} \left\{ 0.212 - 0.053 - 5 \left(\frac{1.053}{1000} - \frac{1}{1000} \right) \right\}$$

$$= 0.01583$$

$$q = \frac{498}{20{,}000 \times 0.25} \left\{ -0.212 - 0.053 - 5 \left(\frac{1.053}{1000} - \frac{1}{1000} \right) \right\}$$

$$= -0.0264$$

$$\theta_0 = 350 \qquad \theta_1 = 1000 \qquad T_0 = 300$$

$$\frac{\theta_1 - T_0}{\theta_0 - T_0} = \frac{1000 - 300}{350 - 300} = 14 \qquad \frac{\theta_1 - \theta_0}{\theta_0 - T_0} = \frac{1000 - 350}{350 - 300} = 13$$

Using Equation [21]

$$\left(\frac{9500 \times 0.25}{1000} \times 14 \right) + \left(\frac{10{,}000 \times 0.25}{1000} \times 13 \right) + \frac{0.593}{0.01583}$$

$$+ \frac{0.407}{-0.0264} = 0.593 \left(\frac{1}{0.01583} + \frac{9500 \times 0.25}{1000} \right) exp(0.01583L)$$

$$+ 0.407 \left(- \frac{1}{0.0264} + \frac{9500 \times 0.25}{1000} \right) exp(-0.0264L)$$

Simplifying

$$2.26 = exp(0.01583L) - 0.372 \, exp(-0.0264L)$$

By trial and error

$$\underline{L = 54 \text{ ft}}$$

Cold Gas Outlet:

$$\frac{y_1}{y_0} = 0.593\ exp(0.01583 \times 54) + 0.407\ exp(-0.0264 \times 54) = 1.491$$

$$\frac{1000 - T_1}{350 - 300} = 1.491$$

$$\underline{T_1 = 925\text{ F}}$$

For Adiabatic Exchanger:

$$10{,}000 \times 650 = 9500(T_1 - 300)$$

$$\underline{T_1 = 984\text{ F}}$$

Mean temperature difference = L.M. of 50 and 16 = 30 deg F
$U = 10$

$$\text{Exchanger length} = \frac{2500 \times 650}{30 \times 10 \times 50} = 108\text{ ft}$$

TABLE 1 COMPARISON OF EXACT AND ADIABATIC VALUES

	Exact	Adiabatic
Outlet temperature, cold gas, deg F...............	925	984
Temperature difference, hot end, deg F...........	75	16
Exchanger length required, ft.....................	54	108

ACKNOWLEDGMENT

The writer wishes to express his appreciation to the M. W. Kellogg Company for permission to publish this paper.

Discussion

R. S. Leigh.[3] In reviewing this paper, mathematics and discussion of the author's derivation are to be omitted, since the writer agrees with this method as being one of several equally accurate methods available for use, although a much simpler method will now be given.

For the better understanding of the problem two sets of curves have been drawn, Figs. 4 and 5 of this discussion, based on varying lengths and varying degrees of insulation, with all other factors as in the paper. Case 1 is set up as a completely insulated exchanger where the loss through the insulation is taken as zero. Case 2 is a moderately insulated exchanger with the over-all coefficient from either the gas side or the air side to the outside equal to 0.214 Btu hr^{-1} ft^{-2} deg F^{-1}. This corresponds to 2 in. of 85 per cent magnesia. Case 3 is the lightly insulated exchanger as used in the example by the author, where the over-all coefficient equals 0.952 Btu hr^{-1} ft^{-2} deg F.$^{-1}$

After calculating the exchanger, taking lengths of 27, 54, 81, and 108 ft, we find that the losses through the insulation and out the stack result in the curves shown in Fig. 3. In any heat-transfer apparatus or part of a heat cycle, the item of interest is the over-all efficiency. In this case, it may be seen that for a lightly insulated (case 3) exchanger, there is a maximum efficiency at about 65 ft of length. To equal the efficiency of a 54-ft-long (case 3) exchanger, the moderately insulated exchanger (case 2) would be only 37 ft long and the (case 1) perfectly insulated exchanger would be 35 ft long. Naturally, the lightly insulated exchanger would have the lower gas temperature leaving, but this would be due to losses, not to greater efficiency.

Fig. 5 illustrates the various gas and air temperatures obtainable with varying amounts of insulation and varying exchanger lengths. Of interest is the fact that the log-mean-temperature difference is fixed for any length, regardless of the amount of insulation. The quantity of heat actually transferred from the gas to the air divided by the over-all coefficient, 10 Btu hr^{-1} ft^{-2}

[3] Project Engineer, Development Department, Chase Brass & Copper Company, Waterbury, Conn.

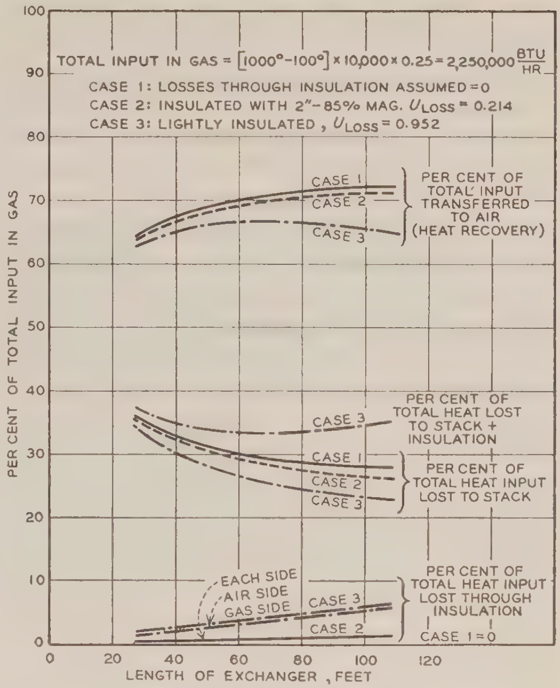


Fig. 4 Heat Balance

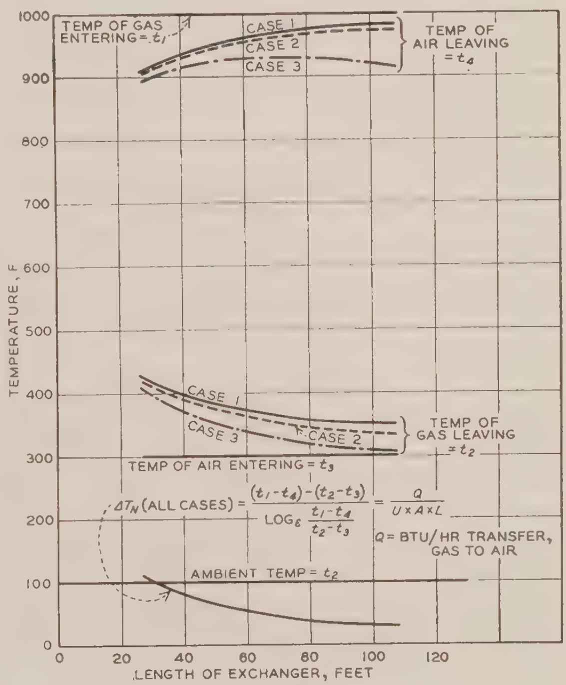


Fig. 5 Gas and Air Temperatures

deg F^{-1} times the area per foot of length times the length in feet equals the log-mean-temperature difference and is also constant for any given length. A solution based on this should give a simpler derivation than that in the paper. The amount of heat removed from the gas is simply this given amount plus the losses on the gas side. The amount of heat leaving the exchanger in the air is equal to this given amount minus the losses on the air side.

The solution in which the insulation is neglected (case 1) is simple to calculate. From this calculation the number of Btu per hour actually transferred may be determined, as well as the log-mean-temperature difference for any given length. To this quantity of heat add the gas-side losses and divide by pounds of gas per hour times the specific heat, to obtain the gas temperature drop. Then subtract the air-side losses from the given quantity of heat transferred and divide by pounds of air per hour times the specific heat, to obtain the air temperature rise.

Of interest is the slight difference between a moderately insulated (2 in. of 85 per cent magnesia) exchanger, case 2, and a completely insulated exchanger, case 1, indicating that the results obtained in the paper were exceptional only because the losses were allowed to be excessive.

The effect of the friction should be included in any problem such as this, since the friction on either the gas or air side tends to increase the leaving temperature slightly. This gain would tend to offset the loss through the insulation.

The use of standard symbols and nomenclature should be encouraged and every effort bent to this end.

Author's Closure

The author is not aware of any other methods to solve the heat-loss problem, other than a tedious step-by-step integration. Unfortunately, no references are given in the discussion. The author cannot agree with Mr. Leigh that his method offers a solution to the problem. It should be obvious that the logarithmic mean cannot be used as the mean temperature difference in a nonadiabatic exchanger.

Contrary to Mr. Leigh's statement, the effect of frictional work on a fluid in a heat exchanger usually has negligible significance on the fluid temperature. This is true because the change in temperature with change in pressure, under adiabatic conditions (Joule-Thomson coefficient), is very small except for a gas near the critical region. For a perfect gas the Joule-Thomson coefficient is zero. If one wishes to take the effect into account the result is almost always a cooling, not a heating, of the fluid.

The author regrets that, through ignorance, standard symbols were not used in the paper.

Temperature Uniformity in Heating-Up Processes

By M. P. HEISLER,[1] NEW YORK, N. Y.

The treatment of temperature-uniformity problems by the method first introduced by Paschkis is extended and enlarged. It is shown that uniformity calculations can be reduced to the solution of a simple trigonometric relation which can be readily solved and analyzed. Standard Gurney-Lurie charts for cylinders and spheres are given in a more complete form than is available at present. A chart is included for determining temperature history in plates subjected to two-dimensional heat flow. Finite cylinders are discussed and three-dimensional heat-flow problems are briefly considered.

Introduction

A temperature-uniformity problem results when a body is to be heated to a given surface and a given center temperature. Such a problem cannot be directly solved by means of the standard heating charts because the furnace temperature and the heating time required to produce the specified uniformity are not known. The possibility of applying the simple heating and cooling charts of Groeber, Hottel, Schack, Newman, and others to temperature-uniformity calculations was first recognized by Paschkis.[2] He prepared a set of curves from the Groeber charts and thereby introduced a fundamentally new parameter, which he called the "temperature-uniformity factor." This temperature-uniformity factor, which he defined as (surface temperature — center temperature)/(surface temperature — temperature of surroundings), was obtained by eliminating the furnace temperature from the standard dimensionless temperature ratio Y, used in all of the charts mentioned. By eliminating the furnace temperature Paschkis was able to derive a new set of curves from which, given only the final surface and center temperatures to which the workpiece was to be heated, the unknown furnace temperature and heating time could be found, provided that the following conditions prevailed:

1 Workpiece was of a simple shape; plate, cylinder, or sphere.

2 A suddenly impressed and thereafter constant furnace temperature was used.

3 Workpiece had constant thermal properties, i.e., properties were temperature-invariant.

4 There was constant boundary resistance throughout the heat period.

5 Uniform surface temperatures existed, e.g., on opposite sides of a slab or plate.

Unfortunately, these conditions are never met in practice, although in particular cases they may be approached. In general, the use of constant average thermal properties in place of the true temperature-varying properties results in relatively slight error. The assumption of simple shape introduces an error proportional to the degree to which the actual shape approximates the assumed one, but it is difficult to measure the exact degree of approximation. The influence of boundary resistance is even more difficult to gage, because boundary resistance is, among other things, a function of the temperature difference existing between furnace and surface of stock and therefore is subject to wide variations during the heating cycle.

In fuel-fired furnaces, constant furnace temperatures are virtually nonexistent. If the charge and furnace are heated-up together, the cold walls and charge will lower the initial furnace temperature. Even after the walls and charge are heated up there will still be temperature fluctuations caused by the temperature control. It may be argued that these latter temperature fluctuations are never very large, yet if they are compared with the temperature differences encountered in the stock at the end of the heating period it will be seen that they may have considerable influence on the temperature uniformity. Similar arguments can be made for cases where the cold charge is suddenly introduced into the hot furnace.

Finally, uniform surface temperatures seldom prevail, whether the charge be heated from one or from all sides. Skids, hearth, contact with neighboring stock, corners, edges—all tend to produce unequal temperature distributions on the surface. Nevertheless, for the sake of obtaining some numerical values average conditions are generally assumed. Very often, even though only approximate, analytical methods yield valuable results.

Of these factors the relative boundary resistance m is of particular importance because the accuracy of the analytical solution depends largely upon how accurately m is known. It has become standard practice to use an average surface heat-transfer coefficient in calculating the relative boundary resistance, the proper choice of the average value leaning heavily upon practical experience. In many instances the engineer has little to use as a guide and the boundary resistance used in his calculations may then largely be the result of concentrated wishful thinking.

In general, surface heat transfer occurs mainly by radiation from some constant furnace temperature to a lower varying surface temperature. It is suggested here, as an initial step in placing the determination of a suitable average boundary resistance on a rational basis, that an average surface temperature be used in calculating the surface heat-transfer coefficient. For example, consider a slab, initially at 70 F, heated in a 2000 F furnace. If it can be assumed that heat transfer takes place largely by radiation, then the conditions illustrated in Table 1 exist. In the table, column 1 shows various surface temperatures of the slab; column 2 shows h values for black-body radiation from 2000 F to the given surface temperature; column 3 shows h

TABLE 1 HEATING-UP DATA FOR SLAB IN 2000 F FURNACE

Surface temperature, deg F	h Btu/(sq ft)(hr)(deg F)	$h/102$	Temperature (deg F)/2000
0	31.2	0.306	0
200	35.0	0.344	0.1
400	38.8	0.390	0.2
600	43.3	0.425	0.3
800	49.0	0.481	0.4
1000	54.8	0.537	0.5
1200	62.0	0.608	0.6
1400	70.0	0.686	0.7
1600	80.0	0.785	0.8
1800	90.0	0.883	0.9
2000	102.0	1.0	1.0

[1] Heat Transfer Research Laboratory, Columbia University.

[2] "Electrische Industrieofen für Weiterverarbeitung," by V. Paschkis, Julius Springer, Berlin, Germany, 1932.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., November 26–29, 1945, of The American Society of Mechanical Engineers.

expressed as a fraction of its value at 2000 F; and column 4 the surface temperature expressed as a fraction of the final value of 2000 F. It can be seen that h is directly proportional to the surface temperature over a large part of the temperature scale, the proportionality becoming better as the temperature increases. This latter fact is particularly important because the surface will be at high temperatures for a considerably longer time than it will be at low temperatures. It is evident, then, that a time average of the surface temperature will be nearly proportional to a time average of the boundary resistance m. A relatively simple method for determining the time-average surface temperature will be given later.

In most instances the workpiece is heated to a center temperature that will be as near to the final surface temperature as an economical heating time will permit. In problems of this type two m values must be used. One based upon the time-average surface temperature, to find the heating time; and the other based upon final surface temperature, to find the temperature at the center. To make this clear consider two identical slabs, one at the temperature of the surroundings, the other at a uniform temperature of 1000 F. The slabs are put into a 2200 F furnace at the same instant and are kept there until the surfaces reach a temperature of 2180 F. It is evident that when the surfaces of the slabs reach 2180 F (at different times, of course) their respective center temperatures will, for all practical purposes, be very nearly identical. However, if the center temperatures are calculated in the customary manner from heating and cooling charts, they will not appear to be the same because the average boundary conductances will be based in one case upon initial radiation from 2200 to 1000 F, and in the other case on initial radiation from 2200 F to the temperature of the surroundings. What actually happens is the following:

As the final surface temperature is approached there is little change in the boundary resistance because of little change in surface temperature. As a result, the center temperature which originally lagged considerably behind the fast-changing surface temperature, has time to catch up to the now slowly changing surface temperature and assumes values determined, not by the time-average surface temperature, but by the actual surface temperature at the end of the heating cycle. It is evident then that in any heating to a high degree of uniformity the center temperature should be determined from the boundary conductance at the end of the heating cycle. This can then be adjusted to take care of the slight lag which will still exist between center temperature and surface conductance.

A temperature-uniformity problem therefore resolves itself into determining average and end boundary conditions. The first will yield the heating time, the second the furnace temperature. The determination of the latter is the simpler and will be considered first.

Nomenclature

m = relative boundary resistance, (conductivity)/(surface heat-transfer coefficient) (half thickness)

m_x = relative boundary resistance for two-dimensional heat flow, based upon half thickness in direction of x-axis

m_y = relative boundary resistance for two-dimensional heat flow, based upon half thickness in direction of y-axis

t' = furnace temperature or temperature of hot ambient

t = surface temperature

t° = center temperature

t_b = uniform base temperature of solid

t_a = time-average surface temperature

t_x = temperature in solid at distance x from center

t^* = edge temperature of long rectangular rod

X = dimensionless time (diffusivity)(time)/(half thickness squared)

X_0 = value of dimensionless time when surface has reached final temperature

X_x = dimensionless time for two-dimensional heat flow, based upon half thickness along x-axis

Y_x = dimensionless temperature ratio at distance x from center of solid (furnace temperature—temperature at point x)/(furnace temperature — base temperature)

Y = dimensionless temperature ratio at surface of solid (furnace temperature—temperature of surface)/(furnace temperature—base temperature)

Y° = dimensionless temperature ratio at center of solid, (furnace temperature—center temperature)/(furnace temperature—base temperature)

Y_a = dimensionless temperature ratio for finding average surface temperature (furnace temperature—average surface temperature)/(furnace temperature—base temperature)

One-Dimensional Heat Flow

It was pointed out that in furnace calculations it is customary to make use of dimensionless parameters, m, X, and Y. Many charts by Newman, Hottel, Bachmann, Gurney-Lurie, etc., have been plotted which give the temperature history of simple shapes like semi-infinite plates, infinite cylinders, and spheres. All of these charts, with the exception of Bachmann's, are plotted as functions of m, X, and Y.

Some modifications must be made in these standard charts before they can be used for temperature-uniformity calculations, because neither the furnace temperature nor the heating time is known. However, once the surface and center temperatures of the workpiece are specified and the relative boundary-resistance factor m is determined, the furnace temperature and the heating time are fixed. To see that this is true consider the following equation, which applies to the heating-up of semi-infinite plates

$$Y_x = \frac{t' - t_x}{t' - t_b} = \sum_{k=1}^{\infty} \frac{2(\sin w_k)(\cos w_k x) \exp(-w_k^2 X)}{w_k + (\sin w_k)(\cos w_k)} \quad [1]$$

where w_z is defined by the characteristic number equation $mw = \cot w$, and

t_x is temperature, varying with distance x and dimensionless time X

t' is temperature of hot ambient—furnace temperature

Y_x is relative temperature at position x and time X

t_b is initial temperature of slab

Now, it is a fact that in the expansion all terms after the first can be neglected for X in the neighborhood of 0.2, since for values of X larger than 0.2, these terms rapidly vanish. To illustrate the extreme rapidity of convergence, consider the first two roots of $mw = \cot w$ when m is equal to 1. The first root is $w = \cot w = 0.8603$; the second root is $w = \cot(w + \pi) = 3.4256$. Suppose X is equal to 1. Then, for the first root $\exp(-w^2X)$ becomes 0.477, and for the second root 0.0000076. Thus if only the exponential factor is considered, it is obvious that the second term in the expansion becomes insignificant; the trigonometric factor serves to reduce the numerical value even more. Since the values of X encountered in temperature-uniformity calculations are generally very much greater than 0.2, Equation [1] can be reduced to

$$Y_x = \frac{2(\sin w)(\cos wx) \exp(-w^2X)}{w + (\sin w)(\cos w)} \quad [2]$$

where the first root of $mw = \cot w$ determines w. In the derivation of Equation [1] it was assumed that the origin was at the mid-plane of the slab. Therefore at the surface $x = 1$ and at the center $x = 0$, and

$$\text{(surface)} \quad Y = \frac{t' - t}{t' - t_b} = \frac{2(\sin w)(\cos w)\exp(-w^2X)}{w + (\sin w)(\cos w)} \quad [3]$$

$$\text{(center)} \quad Y^\circ = \frac{t' - t^\circ}{t' - t_b} = \frac{2(\sin w)\exp(-w^2X)}{w + (\sin w)(\cos w)} \quad [4]$$

where t is the surface temperature and t° the center temperature of the slab. Dividing Y by Y° gives

$$Y/Y^\circ = \cos w = (t' - t)/(t' - t^\circ) \quad [5]$$

Equation [5] is plotted in Fig. 1 as a function of m; and since m, t, and t° are known, the unknown furnace temperature t' can be found. To find t', enter Fig. 1 with the known m as abscissa, go vertically to the curve for the plate and thence horizontally and read the ordinate. The ratio $(t' - t)/(t' - t^\circ)$ will hereafter be called the relative-temperature-uniformity ratio, since it is a measure of the relative temperature uniformity existing between stock and furnace at the end of the heat period.

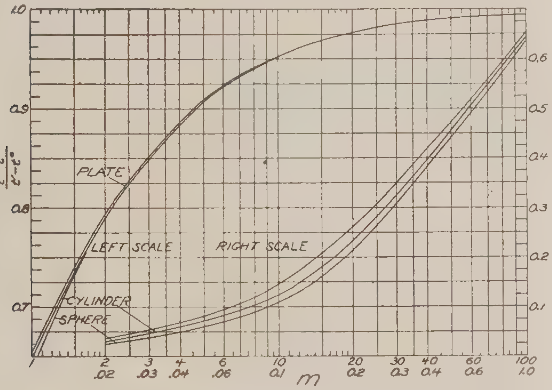


FIG. 1 CHART FOR DETERMINING RELATIVE TEMPERATURE-UNIFORMITY RATIO

Equation [5] shows that the furnace temperature is a function only of the relative boundary-resistance factor m and the given surface and center temperatures. If it were possible to have an infinite heat-transmission coefficient h, m would become zero, $\cos w$ would also become zero, and the surface temperature would equal the furnace temperature. For any given surface and center temperatures m is a direct measure of the required furnace temperature. If m is large, which, other things being equal, is tantamount to having poor surface heat transmission, $\cos w$ approaches 1. This means that a high furnace temperature is necessary to produce the desired uniformity. Conversely, if m is small the furnace temperature approaches the surface temperature as a minimum. The equation clearly shows that lower furnace temperature and better temperature uniformity within a given time are obtained by increasing surface heat transmission from flame to stock.

By comparison with the Gurney-Lurie-type charts it can be seen that small m's also imply low heating times. The relation of furnace temperature and heating time to boundary resistance is shown in Fig. 2. The curves were obtained from Fig. 1 and illustrate how heating times and furnace temperatures increase with boundary resistance. The curves shown hold for a slab

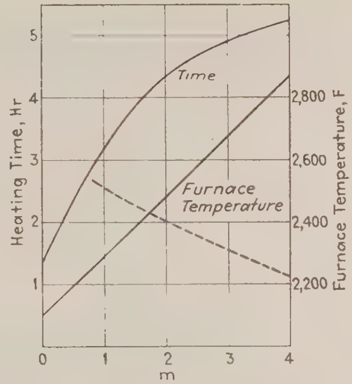


FIG. 2 VARIATION OF HEATING TIME AND FURNACE TEMPERATURE WITH RELATIVE BOUNDARY RESISTANCE

heated to a surface temperature of 2100 F and a center temperature of 2000 F. The dimensions and thermal properties of the slab are not given because the curves are intended to show trends only. For the sake of illustration no distinction was made between average and end boundary conditions.

In the introduction it was pointed out that the relative boundary resistance m is a function of the furnace temperature. Since the furnace temperature is not known, the relative boundary resistance cannot be directly determined, and inversely, since the boundary resistance is not known the furnace temperature and heating time cannot be calculated. One of the necessary calculations in furnace design is a plot showing the relation of the surface heat-transmission coefficient to furnace temperature. This chart must be calculated from the various flame characteristics, such as thickness of flame, composition of the hot gases, amount of dissociation, luminosity of the flame, emissivity and temperature of the surface, and the like. From this chart the surface heat-transmission coefficient can readily be converted into the relative boundary resistance m. Since the furnace temperature increases with increasing h, it will decrease with decreasing m. Suppose that the broken line in Fig. 2 is this calculated temperature versus end m relation for the given flame properties. Then the intersection of the broken temperature line with the solid temperature line (determined from Fig. 1) gives the value of m and the furnace temperature that will, approximately, produce the desired temperature uniformity.

The heating time is determined from the same heat-transmission curve from which the furnace temperature was determined, but in this case an average m is used. To illustrate, suppose such a heat-transmission chart is available and that the time-average surface temperature over the heat period is known. With this average surface temperature the average surface heat-transfer coefficient can be read directly and therefore the average m is determined. With this m, the heating time can be found from the standard heating-cooling charts. The average surface temperature is found as follows:

$$Y_{\text{avg}} = \int_0^{X_0} \frac{YdX}{X_0}$$

where X_0 is the value of the dimensionless time when the final surface temperature is reached. Therefore

$$Y_{\text{avg}} = \int_0^{X_0} \sum_{k=1}^{\infty} \frac{2(\sin w_k)(\cos w_k)\exp(-w_k^2X)dX}{X_0(w_k + \sin w_k \cdot \cos w_k)}$$

$$= \sum_{k=1}^{\infty} \frac{2(\sin w_k)(\cos w_k)[1 - \exp(-w_k^2 X_0)]}{w_k^2 X_0(w_k + \sin w_k \cdot \cos w_k)}$$

Since X_0 is greater than 0.2 this can be simplified to

$$Y_{avg} = \frac{Y\,[\exp(w^2X_0) - 1]}{w^2X_0} = \frac{t' - t_a}{t' - t_b}$$

where t_a is the average surface temperature during the interval of exposure. Eliminating Y, this can be written

$$\frac{t' - t_a}{t' - t} = Y_a = \frac{\exp(w^2X_0) - 1}{w^2X_0} \quad \ldots\ldots\ldots\ldots [6]$$

Values of w^2 can be read from Fig. 3 and values of Y_a from Fig. 4. The use of the charts will become clear from the following example: A 6-in. steel slab of diffusivity 0.5 sq ft per hr, conductivity 25 Btu/(ft)(hr)(deg F) is to be heated to a surface temperature of 1970 F in a furnace at 2070 F. The temperature of the surroundings is 70 F. Heat transfer is assumed, for simplicity, to take place by radiation only, although it should be noted that the method will apply to any heat transfer for which a curve can be drawn. What is the average surface temperature over the interval? What will be the temperature at the center at the end of the heat period?

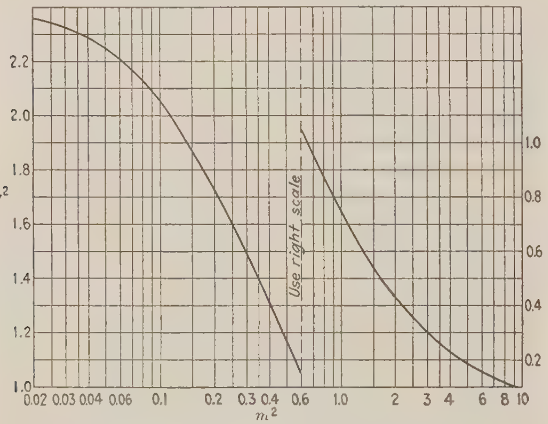


FIG. 3 CHART FOR DETERMINING SQUARE OF CHARACTERISTIC NUMBER w

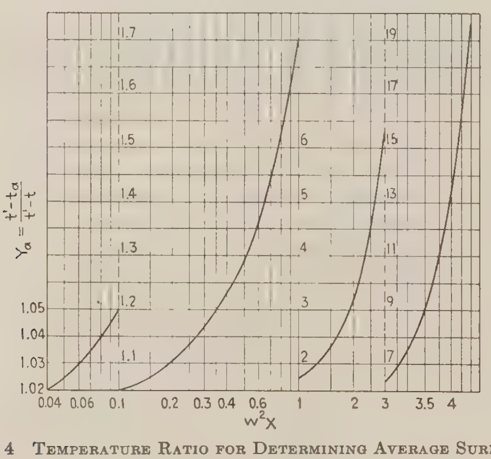


FIG. 4 TEMPERATURE RATIO FOR DETERMINING AVERAGE SURFACE TEMPERATURE OF SLAB

The temperature-time curve of the surface is reasonably parabolic. Therefore assume an initial average surface temperature of (2/3)(1970) = 1315 F. From radiation charts,[3] for radiation between two infinite parallel plane surfaces, one at 2070 F and the other at 1315 F, the surface heat-transmission coefficient h is found to be 70 Btu/(sq ft)(hr)(deg F). Therefore m = (25)/(70)(3/12) = 1.43. From standard cooling charts,[3] for m = 1.43 and Y = (2070 — 1970)/(2070 — 70) = 0.05, X_0 is found to be 4.9. From Fig. 3, when m is equal to 1.43, w^2 is 0.565. Therefore w^2X_0 = 4.9 × 0.565 = 2.77. From Fig. 4, Y_a = 5.45 = (2070 — t_a)/(2070 — 1970). Solving this, t_a is found to be 1525 F.

The procedure is now repeated with the new t_a. For t_a = 1525, h is 80 Btu/(sq ft)(hr)(deg F). Solving for m, we find it to be 1.25. From this X_0 = 4.35 is obtained and also w^2 = 0.628. From Fig. 4, for w^2X_0 = 0.628 × 4.35 = 2.73, Y_a is read as 0.53. Solving, it is found that the average surface temperature is 1540 F. This is sufficiently close to the previous value and further refinement is unnecessary.

To find the center temperature proceed as follows: For radiation between 2070 and 1970 F, McAdams[3] gives an h of 110 Btu/(sq ft)(hr)(deg F). From this m is 0.91 and from Fig. 1 $(t' - t)/(t' - t_0)$ is 0.629. Solving for t_0 gives t_0 = 1875 F.

The difference between average and end-boundary-resistance ratios is pronounced, and it is evident that quite different center temperatures will be obtained with both m values. Strictly speaking, neither temperature will be exact, but that obtained from the final boundary conditions will certainly be more correct than that obtained from average boundary conditions.

The foregoing analysis was carried through for a semi-infinite plate. Equations similar to Equation [5] are derived in the Appendix for infinite cylinders and for spheres. It is sufficient here to remark that the curves shown in Fig. 1 for cylinders and spheres can be used in the manner described for the semi-infinite plate. Average surface-temperature equations can be easily derived. One restriction is placed on all these curves; they apply only for values of X equal to or greater than 0.2. However, this does not detract from their applicability because, as is shown, values of X encountered in temperature-uniformity calculations are always far greater than 0.2.

Figs. 5 and 6 are included here because no standard heating-cooling charts of the Hottel type are available for cylinders and spheres, and temperature-uniformity calculations frequently require the use of curves which give small temperature ratios accurately. Only the center temperatures are given. To find the surface temperatures, multiply the ordinate from Fig. 5 or 6 with the corresponding ordinate from Fig. 1. Figs. 5 and 6 give $Y° = (t' - t°)/(t' - t_b)$, and Fig. 1 gives $(t' - t)/(t' - t°)$. The product gives $(t' - t)/(t' - t_b)$, from which the surface temperature can be found. A more complete discussion is given in the Appendix and the relation is extended to include all in-between points.

It should be emphasized that the relative temperature-uniformity ratio and the uniformity factor are not single-valued parameters and do not in themselves determine a unique solution of the problem. To see that this is true consider the uniformity factors 1 — (1900/2000) and 1 — (3800/4000). The factors are numerically equal and give the same heating times, based upon the Paschkis charts,[2] but they will not result in the same furnace temperatures. Similarly, a given relative temperature-uniformity ratio has an infinite number of dimensionless time ratios X. The system is uniquely determined only by the parameters Y, m, and X. The relative temperature-uniformity ratio and the uniformity factor are auxiliary parameters which show certain

[3] "Heat Transmission," by W. H. McAdams, second edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1942.

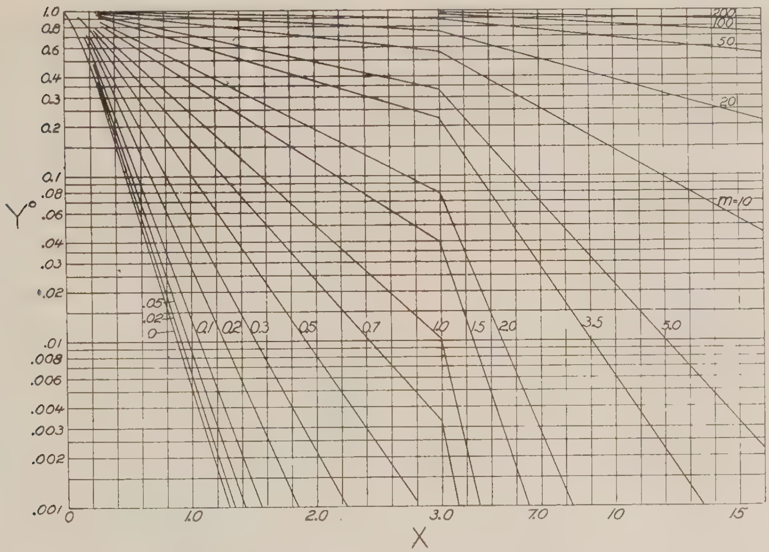


Fig. 5 Chart for Determining Temperature History at the Axes of Cylindrical Bodies

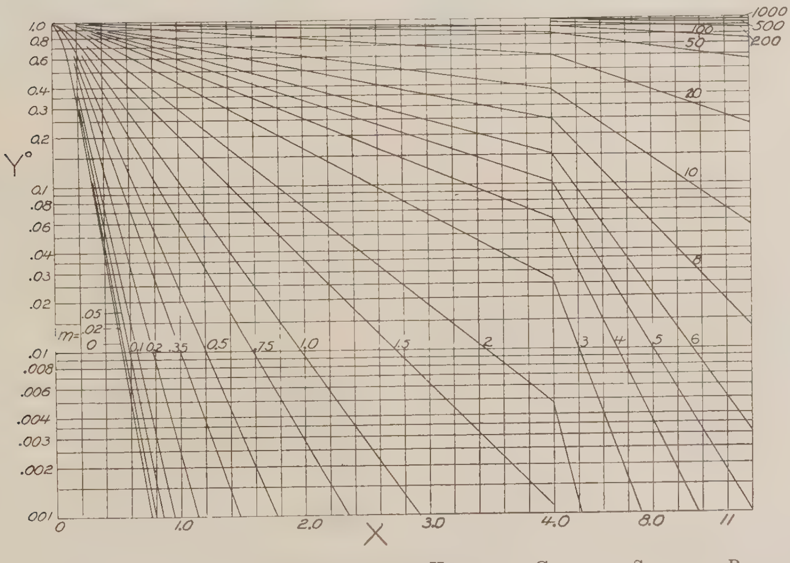


Fig. 6 Chart for Determining Temperature History at Center of Spherical Bodies

relations that exist among the different variables; the former ties in the furnace temperature and the latter the heating time with boundary resistance and surface and center temperatures. Each depends upon the other; if taken together they eliminate the necessity for using the standard cooling charts, but they cannot be combined into a single chart.

Summary. In the preceding paragraphs simple equations have been presented which enable uniformity calculations to be made readily. It is pointed out that the simplicity of the relations derived makes it a comparatively easy task to determine the mutually dependent furnace temperature and relative boundary resistance. Temperature-time-space curves are given to supplement and add to those available at present.

Two-Dimensional Heat Flow

The definition of temperature uniformity is based upon maximum surface and minimum mid-plane temperatures. In two-dimensional heat flow maximum surface temperatures occur at the edge. In the heating of a plate infinite in one direction only, for example, a long rod, or a short plate insulated on two opposite faces, if the origin of a rectangular system of co-ordinates is thought of as placed at the geometric center of the plate, Fig. 7, it is shown in the Appendix that edge, center, and furnace temperatures are related by

$$(t' - t^*)/(t' - t^\circ) = (\cos w)_x(\cos w)_y \qquad [7]$$

where as before t' and t° are furnace and center temperatures,

respectively; t^* is the edge temperature; $\cos(w)_x$ and $(\cos w)_y$ are found from Fig. 1 by calculating the relative boundary resistances m_x and m_y for heat flow in the x- and y-directions, respectively, just as is done for one-dimensional heat flow. Since t^* and t° are given by the specifications and m is assumed for the present to be known, the furnace temperature t' can be found

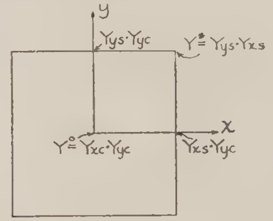


Fig. 7 Y-Factor Relations for Two-Dimensional Heat Flow in Plate

A chart similar to Fig. 1 can be plotted, but in place of a single curve there will be a family of curves with either m_x or m_y as abscissa and m_y or m_x as parameter. In general m_x will not be equal to m_y.

If the sides of the body are of equal length, $(\cos w)_x$ will equal $(\cos w)_y$ and $(t' - t^*)/(t' - t^c)$ will equal $(\cos w_x)^2 = (\cos w_y)^2$, since the two will be numerically equal. In this case the solution can be obtained from Fig. 1 and the standard heating-cooling charts. For example, let $m_x = m_y = 1$, and let the surface temperature be 2100 F and the center temperature be 2000 F. From Fig. 1 and Equation [2]

$$(t' - t^*)/(t' - t^\circ) = (0.654)^2 = 0.428$$

Therefore $t' = 2172$ F. From Equation [18] of the Appendix the temperature ratio at the center is

$$\sqrt{(2172 - 2000)/(2172)} = \sqrt{0.0781} = 0.2795$$

From the cooling curves for $m = 1$, X is found to be 1.88.

When the sides are not of equal length, the cooling charts for one-dimensional heat flow cannot be used. As yet only a few two-dimensional charts have been prepared (an unpublished work by Rohsenow and his co-workers at Yale University, under direction of Professor Wohlenberg). To cover completely all conditions requires an inpracticably large number of such graphs. In the second section of the Appendix it is shown that the system, in so far as temperature-uniformity problems are concerned, can be completely defined by the six parameters m_x, m_y, x/y, X_x, $1 - t^\circ/t'$, and $1 - t^*/t'$. X_x is the dimensionless time ratio, with thickness squared being the square of the length of the smallest side; x/y, a shape factor, is the ratio of the lengths of the sides. The other parameters have already been described. To cover completely all practical combinations of these variables would require (for 15 values of m_y and 5 of x/y) 75 charts of conventional form for each of the two temperature ratios, or 150 charts in all. Such a presentation is obviously out of the question. If an intersection chart is substituted, however, it is possible, by imposing a few restrictions which will be considered later, to cover in a single chart most of the cases met in practical applications. A chart of this type, applying to plates only, is shown in Fig 8.

The chart is somewhat more difficult to use than the simple cooling charts, but some sacrifice in simplicity must be made in order to obtain generality. For temperature-uniformity calculations m_x, m_y, x/y, and Y° will be known (Y° having been found from Fig. 1) and the unknown will be the relative heating time X_x. To find X_x:

1 With the given m_y as abscissa go to the first quadrant and locate the intersection of the vertical from m_y with the curve of constant m_x. From the point of intersection proceed horizontally to the second quadrant and locate the point of intersection with Y°.

2 With the given x/y as abscissa enter the fourth quadrant, go horizontally to m_y; from m_y proceed perpendicularly to m_x and thence horizontally to X. The intersection of this horizontal with a perpendicular from the value of Y° located in the first step determines X.

As an example, the case of the square bar, solved previously by using the one-dimensional cooling chart, is shown (dotted line in Fig. 8). In this case enter the fourth quadrant at $x/y = 1$ and the first quadrant at $m_x = 1$. The intersection in the third quadrant gives $X_x = 1.88$.

Fig. 8 is not restricted to temperature-uniformity calculations, but can be applied to heating and cooling problems in general. However, in any problem neither X_x nor X_y should be less than 0.2 for values of m equal to zero. For larger m values, X less than 0.2 may be used, a simple criterion being to refer to the Gurney-Lurie charts and find for what value of X the lines of constant m have not begun to curve.

In the terminology of this paper X_y will always be less than X_x, because $X_y = X_x(x/y)^2$. Therefore for small values of x/y, X_y should always be calculated to find out if it is within the imposed limit. In general, though, small values of x/y will mean that heat flow in the y-direction is of negligible influence; therefore these cases can be treated as one-dimensional problems, using the standard charts. Fig. 8 can be applied to nearly all problems that cannot be treated one-dimensionally.

Appendix

Infinite Cylinders. Bodies having circular symmetry can be treated in a manner similar to that used for plates. The equation, in dimensionless form, for the heating-up of an infinite cylinder, or a short cylinder insulated at both ends, can be written

$$Y(x,X) = \sum_{k=1}^{\infty} \frac{2J_1(u_k)J_0(u_kx)\exp(-u_k^2X)}{J_0^2(u_k) + J_1^2(u_k)} \cdot \frac{1}{u_k} \quad [8]$$

where u_k is defined by $J_0(u) = muJ_1(u)$.

It is true also for cylinders that the values of X encountered in temperature-uniformity calculations are such that in the expansion all terms after the first become negligible. Therefore Equation [8] can be reduced to

$$\text{(surface)} \qquad Y = \frac{2J_1(u)J_0(u)\exp(-u^2X)}{J_0^2(u) + J_1^2(u)} \cdot \frac{1}{u} \quad [9]$$

$$\text{(center)} \qquad Y^\circ = \frac{2J_1(u)\exp(-u^2X)}{J_0^2(u) + J_1^2(u)} \cdot \frac{1}{u} \quad [10]$$

Proceeding as before

$$Y/Y^\circ = J_0(u) = (t' - t)/(t' - t^\circ) \quad [11]$$

Equation [11] is plotted in Fig. 1 (see curve for cylinder). The furnace temperature is found in the same manner as was prescribed for the plate. The heating-up time can be found from the cooling chart for the cylinder, Fig. 5. Charts other than the Gurney-Lurie type are not suited for temperature-uniformity calculations because they do not have accurate readability in the small temperature ratios. Charts are given only for center temperatures because, as was pointed out previously, temperatures for the surface can be found from the relation $Y^\circ(Y/Y^\circ) =$

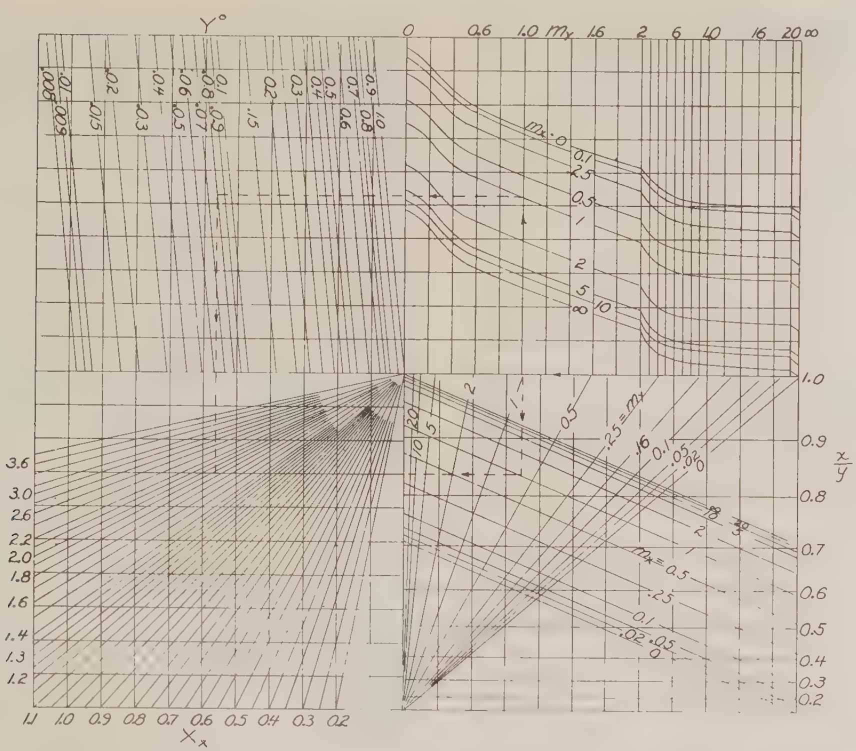


FIG. 8 CHART FOR DETERMINING TEMPERATURE HISTORY AT MID-PLANE OF PLATE SUBJECTED TO TWO-DIMENSIONAL HEAT FLOW

Y. This relation is exact for all values of Figs. 5 and 6 in which the temperature curves are straight lines and this covers practically the entire range of the charts. Little is lost by not being able to use this relation in the small remaining portion of the charts because Gurney-Lurie type charts are not suited for temperature ratios at small values of X; these can be found best by charts derived from short-time expansion formulas.

Spheres. The equation for the sphere is

$$Y(x,X) = \sum_{k=1}^{\infty} 2\frac{(\sin v_k - v_k \cos v_k)\exp(-v_k^2 X)}{v_k - (\sin v_k)(\cos v_k)} \cdot \frac{\sin(v_k x)}{v_k x} \quad [12]$$

and the corresponding characteristic number equation is $mv = (m - 1)\tan v$. Again, in temperature-uniformity calculations, all terms after the first can be neglected and the equation reduced to

$$\text{(surface)} \quad Y = 2\frac{(\sin v - v \cos v)\exp(-v^2X)}{v - (\sin v)(\cos v)} \cdot \frac{\sin v}{v} \quad [13]$$

$$\text{(center)} \quad Y^\circ = 2\frac{(\sin v - v\cos v)\exp(-v^2X)}{v - (\sin v)(\cos v)} \quad [14]$$

From the equations for the surface and center

$$Y/Y^\circ = (\sin v)/v \quad [15]$$

Summary. In the preceding paragraphs it has been shown that by eliminating the dimensionless time from the heat equations for plates, spheres, and cylinders, simple equations result which relate surface, center, and furnace temperatures to boundary resistance. From these elemental relations dimensionless temperature ratios can be formed which make it possible to use standard heating-cooling curves to find the time required to obtain the desired temperature uniformity. In addition, the simple temperature and boundary-resistance relations make it comparatively easy to determine the unknown boundary resistance.

Temperature-time-space curves for cylinders and spheres are given to supplement those available at present. These curves are given for center temperatures only, but can be readily extended to surface temperatures by the use of Fig. 1, and to all in-between points by forming the products $Y^\circ \cos(wx)$, $Y^\circ J_0(ux)$, and $Y^\circ(\sin vx)/v$ for plates, cylinders, and spheres, respectively. In all instances these relations are restricted to values of X equal to or greater than 0.2.

TWO-DIMENSIONAL HEAT FLOW

Supplementing the nomenclature previously given, the following additions will be used:

Y° relative temperature for center $= (1 - t^\circ/t')_x, (1 - t^\circ/t')_y$

Y^* relative temperature for edge of a finite plate $= (1 - t/t')_x, (1 - t/t')_y$

Y_{ys}, Y_{xs} relative temperatures at surfaces perpendicular to x- and y-axes, respectively, when body is considered as a semi-infinite solid $= (1 - t/t')_x, (1 - t/t')_y$

Y_{yc}, Y_{xc} relative temperatures at mid-planes perpendicular to x- and y-axes, respectively, when body is considered as a semi-infinite solid $= (1 - t^\circ/t')_x, (1 - t^\circ/t')_y$

x distance from vertical mid-plane to surface

y distance from horizontal mid-plane to surface

d diffusivity

θ time (not dimensionless)

t_b assumed to be zero

Finite Plate. In two-dimensional heat flow essentially the same methods are applied as in one-dimensional heat flow. The heat-flow equation must now satisfy the differential equation

$$\frac{\partial t}{\partial \theta} = d\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2}\right) \qquad [16]$$

where x and y are rectangular co-ordinates indicating the direction of heat flow, and d is the diffusivity. Newman[4] showed that if Y_x was the solution of

$$\partial t/\partial\theta = d(\partial^2 t/\partial x^2)$$

and its boundary conditions, and Y_y was the solution of

$$\partial t/\partial\theta = d(\partial^2 t/\partial y^2)$$

and its boundary conditions, then $Y_x Y_y$ was a solution of Equation [16]. McLachlan,[5] following the method of Newman, showed that in the heating of solids the partial differential equation is satisfied by a solution of the form $1 - Y_x Y_y$, where Y_x is the expansion part of Equation [1] and Y_y is the expansion part of the same equation with y introduced in place of x. The corresponding heating-up temperatures are $H_x = 1 - Y_x$ and $H_y = 1 - Y_y$.

In two-dimensional heat flow the maximum temperatures will occur at the edges of the body and the minimum temperature at the center. The desired temperature uniformity will be given in terms of the edge and center temperatures t^* and t°, respectively, and the following relations (see Fig. 7) then hold

(edge) $\qquad Y^* = 1 - t^*/t' = Y_{ys}Y_{xs} = 1 - H^*$

(center) $\qquad Y^\circ = 1 - t^\circ/t' = Y_{yc}Y_{xc} = 1 - H^\circ$

To simplify the form of the equations let

$$a_x = 2(\sin w)/(w + \sin w \cos w)$$

$$b_x = \cos(wx)$$

$$e_x = \exp(-w^2 X_x)$$

for heat flow in the direction of the x-axis. For heat flow in the direction of the y-axis, use the same notation but replace the subscript x by y. Then

(edge) $\qquad Y^* = (abe)_{xs}(abe)_{ys} \qquad [17]$

(center) $\qquad Y^\circ = (ae)_{yc}(ae)_{xc} \qquad [18]$

Dividing Equation [17] by Equation [18] gives

$$(t' - t^*)/(t' - t^\circ) = (\cos w)_x(\cos w)_y \qquad [19]$$

Solution of this equation yields the furnace temperature. The heating time requires a solution of

$$Y^\circ = 1 - t^\circ/t' = (2\sin w)_x(2\sin w)_y\{\exp -X_x[w_x^2 + (w_y x/y)^2]\}/(w + \sin w\cdot\cos w)_x \cdot (w + \sin w \cdot \cos w)_y$$

This equation is plotted in Fig. 8. Only center temperatures are plotted. The surface temperatures can be found from Y°, Fig. 1, and the following relations, Fig. 7:

(vertical mid-plane) $\qquad Y_{ys}Y_{xc} = Y^\circ(\cos w)_y \qquad [20]$

(edge) $\qquad Y_{ys}Y_{xs} = Y^\circ(\cos w)_x(\cos w)_y \qquad [21]$

(horizontal mid-plane) $\qquad Y_{xs}Y_{yc} = Y^\circ(\cos w)_x \qquad [22]$

Finite Cylinders. The handling of finite cylinders is the same as for finite plates. If the axis of the cylinder is taken in the direction of the x-axis, it follows, from the similarity between the heat-flow equations for bodies having circular symmetry and those for plates, that

$$(t' - t^*)/(t' - t^\circ) = J_0(u)(\cos w)_x$$

Values of $(\cos w)_x$ and $J_0(u)$ can be obtained from Fig. 1; from their product the furnace temperature can be calculated. As in the case of two-dimensional heat flow in slabs, the solution of the heating time is dependent upon the preparation of the requisite temperature-time-space charts. A chart similar to Fig. 8 can be prepared, or perhaps other methods of plotting, more convenient in form, may be evolved. Equations for determining temperatures at points other than at the edge and center, for example, equations similar to Equations [20], [21], and [22] can be readily derived, the only restriction being that neither dimensionless time ratio be less than 0.2.

The treatment of three-dimensional heat-flow problems follows directly from the methods given for one- and two-dimensional heat flow.

In two- and three-dimensional heat flow an additional complication enters in determining the average boundary resistance, because of the variation of surface temperature from edge to center of the face. Not only will a time average of surface temperature have to be determined, but also a space average, since the faces do not have uniform temperature distribution, and therefore do not have uniform boundary resistances.

Comment. In the preceding discussion only homogeneous solids have been considered. In the case of laminated solids in which the thermal conductivity may be different along the various axes, the value of thermal diffusivity may be correspondingly adjusted for the different directions, and the calculation carried out as usual. If the surface heat-transmission coefficient is different at the various faces, these different values of h may be used, the only limitation being that of maintaining symmetry; the values of m must be the same for any pair of parallel faces.

Acknowledgment

This paper is largely an extension of the work on temperature uniformity originally initiated by Dr. Victor Paschkis, head of the Heat and Mass Flow Analyzer Laboratory at Columbia University. The author is indebted to him for having brought the subject to his attention.

Discussion

F. S. Bloom.[6] To the engineer who is familiar with practical furnace design and operation, the paper would appear to offer some difficulty, because of the necessary assumptions which the author was forced to make in obtaining a practical solution.

However, he has developed from his mathematical analysis certain facts which agree with test observations of heating furnaces. Some of these statements are worth while emphasizing. For instance, the author states, "As the final surface temperature is approached, there is little change in the boundary resistance, because of little change in surface temperature. As a result, the center temperature, which originally lagged considerably behind the fast-changing surface temperature, has time to catch up to the now slowly changing surface temperature."

[4] "Heating and Cooling Rectangular and Cylindrical Solids," by A. B. Newman, *Industrial and Engineering Chemistry*, vol. 28, 1936, pp. 545–548.

[5] "Complex Variable and Operational Calculus," by N. W. McLachlan, Cambridge University Press, 1939; also The Macmillan Company, New York, N. Y., 1942.

[6] Bloom Engineering Company, Pittsburgh, Pa.

By this statement he approaches the conclusions of Williams and Allen, who have previously stated that center temperatures, or temperatures under the surface of the material, follow the pattern of the surface temperature, and ultimately increase or decrease at the same rate as the surface temperature. From this we can conclude that a temperature history or curve of center temperature will follow, at some later time, the temperature history of the surface. In other words, if we plot surface-temperature history for a piece being heated, we can approximate the history of the center temperature.

If we go further in this analysis, we approach the conclusion that temperature uniformity "center to surface" must be considered with time.

The center temperature knows nothing about what happens to surface temperature at any specific time, although it is similarly affected after some delay. This delay or time lag is a factor dependent upon material size, but is constant regardless of speed of heating, that is, a 6-in. square can be heated in 20 min or 6 hr, but still the time for center temperature to equal surface temperature is the same. The magnitude of temperature difference is very large with fast heating, but after the proper equal time interval, temperature uniformity is the same. We do not produce temperature uniformity by slow heating.

The author's work as expressed in this paper stresses temperature uniformity. The writer does not think he intended to convey this impression. Let us face this problem. We heat a slab to a surface temperature of 2300 F, center temperature 2250 F. It is taken from the furnace and lies on the mill table for 15 sec. The surface cools down while the center continues to increase. We end up in the mill with a mixture of temperatures which cannot be equalized. So why demand or think of perfect uniformity.

Temperature-uniformity data are lacking. What we need is a complete study of temperature differences "surface to center" as related to specific industrial-mill operation. The author's method of calculation could be used for such a study. As is quite general, we get the "cart before the horse." Similarly, we get ourselves involved in calculations which get nowhere except to scare engineers into being cautious where it is not necessary. Let us find out the temperature uniformity demanded by the process.

The author also states, as his conclusion of the calculations, that a lower furnace temperature and better temperature uniformity can be obtained by higher heat transmission. He means luminous flames, thick flame blanket, and greater wall area, which in turn means ample roof height. This factor has been seen time and again on many billet-heating furnaces where a heavy luminous flame produces a better scale condition and a better working piece in the mill operation.

He touches gently on temperature control with which the writer fully agrees. There is no use soaking steel if the surface temperature is changing up and down. A 50-deg F temperature difference is easy to produce with such conditions. Good temperature control on heating furnaces is a necessity for uniform temperature distribution. It should be operated by measuring surface temperature, and in this manner a complicated heating-furnace operation can be reduced to the normal temperature-control problem.

The author has presented an outstanding theoretical paper. It is too complicated for the writer fully to comprehend. He must be pretty close to the right answer, because his conclusions seem accurate. It is suggested that he study mill and furnace conditions and then calculate from surface temperature and mill data the permissible out-of-balance temperatures which mill operations can stand.

Victor Paschkis.[7] The author is to be congratulated for a valuable piece of research. He refers to some previous work of the writer.[2] It may be interesting to compare the two methods of determining the temperature uniformity as follows:

In order to obtain a desired uniformity expressed either by the "temperature-uniformity factor" or by the "relative temperature-uniformity ratio," a definite furnace temperature and a definite heating time must be applied. Both should be expressed in dimensionless units. The determination from the temperature-uniformity factor or the relative uniformity ratio compare as follows:

	Heisler	Paschkis
Unknown	Determines the unknown from	
X	Gurney-Lurie charts	$\frac{t - t_0}{t - t_b} = f(m, x)$
t'	$\frac{t' - t}{t' - t_0} = f(m)$	Gurney-Lurie charts

The writer does not see a particular advantage of the new method above the known one and wonders what advantages can be claimed for it.

The most important part of the paper, as far as one-dimensional problems are concerned, is that dealing with the dual relationship between surface temperature, furnace temperature, and m value.

The recommendation that the heating time should be calculated with the time average m lacks proof, and it would be interesting to obtain definite proof. Incidentally, the step-by-step method described for finding t_a could be conveniently replaced by a procedure similar to that used in Fig. 2 of the paper.

It seems regrettable that the paper does not contain curves for cylinders and spheres, similar to those presented in Figs. 3 and 4, for slabs; also that the curves for the relative temperature-uniformity ratios for rectangular bodies are not developed. Only if curves are available can it be hoped that functions such as those presented in this paper will be widely used in industry. It is hoped that the author will develop such curves and present them at a later time. The author seems to exclude from consideration convection-type furnaces which today certainly gain in importance.

W. Trinks.[8] While the author presents an excellent mathematical study, it is, however, too involved and obscure for the average engineer and will, for that reason, not be appreciated. Many an engineer will spend a couple of hours on it and will then lay it aside. With a few changes the usefulness of the paper could be vastly increased.

A sketch of the temperature distribution in a solid with notation of the temperature differences which are sought would be of great value.

In the "Introduction," condition No. 6 should be added; namely, that at the beginning of the heating process the work blank has uniform temperature throughout.

What is meant by the "time average" of boundary resistance m? This question is asked in view of the author's statement that boundary resistance is to be constant. It may be advisable to define boundary resistance and to give the units. Several definitions of boundary resistance are current.

The writer is at a loss to understand how the heat-transfer coefficient h is obtained when both the surface temperature and the flame temperature are alike. Can there be any heat transfer in that case?

The author does not say how Equations [1] and [8] were obtained. It is surmised that J_0 and J_1 are Bessel's functions. If so, that fact should be stated.

[7] Head, Heat and Mass Flow Laboratory, Columbia University, New York, N. Y.

[8] Consulting Engineer, Pittsburgh, Pa. Mem. A.S.M.E.

In modern heating practice, we do not bring the surface of the work blank up to furnace temperature and soak it a long time. We do something else. We heat the work blank quickly to a surface temperature which is above the desired final temperature and then soak it in a somewhat cooler furnace section, where the surface temperature drops and the center temperature rises. It would be wonderful if the author could solve, with Bessel's functions, the final temperature distribution which we obtain with that heating method.

Author's Closure

In the text the author has mentioned that Roshenow and coworkers' charts for two-dimensional heat flow have not yet been published. These charts have since appeared in the February, 1946, issue of the transactions of the A.S.M.E.

Both Mr. Bloom and Mr. Trinks stressed the desirability of solving problems involving step temperature functions. Mr. Bloom mentioned the case of a hot slab cooling on a mill table, and Mr. Trinks that of a slab heated quickly to a surface temperature which is above the final desired temperature and then soaking in a somewhat cooler section of the furnace. Many problems of a similar nature arise in metal heating. For example, in some types of heating operations, after the charge is brought to a given surface temperature, the furnace is turned off, and both charge and furnace are allowed to cool together. Another case occurs in surface hardening by high-frequency induction heating followed by internal quenching. Numerous others could be cited. Though the thermal operations involved are entirely unrelated physically, yet all of them are decidedly similar when considered in a mathematical sense. For instance the problem mentioned by Mr. Trinks is mathematically identical to the problem of surface hardening by high-frequency induction heating followed by an internal quench. Both of these can be readily solved by means of existing temperature charts, such as those shown in Figs. 6 and 7. In general all the problems mentioned simply involve relaxing from an initial known temperature distribution. The author is at present preparing a paper on the general solution of problems of this type.

Mr. Bloom points out that center temperatures can be approximated when surface temperatures are known, because of the constant time lag between equal surface and center temperatures. The magnitude of the time lag for any given surface-center temperature can be found from

$$t = \frac{-L^2 \log_e (\cos w)}{d\, w^2}$$

where

r = time lag in hours.
d = diffusivity in sq ft per hr
L = half thickness of slab in feet
w = first root of $m\, w = \cot w$

Values of w^2 are plotted in Fig. 3 and cos w in Fig. 1. Obviously, the time lag varies directly as the square of the thickness and inversely as the diffusivity. Because of the factor w, it also depends on the boundary resistance, decreasing as the boundary resistance increases.

Regarding the "time-average" of boundary resistance, mentioned by both Mr. Trinks and Mr. Paschkis: It is true that in the mathematical problem boundary resistance is considered to be constant. However, in actual heating-up processes boundary resistance varies throughout the heating cycle. Therefore, in applying mathematical formulas to actual thermal problems, it is necessary to estimate in some way or other the average value of the boundary resistance over the heating period, substitute this average value into the formulas, and solve as though the boundary resistance were constant. This average boundary resistance can be based on various thermal concepts. The author has suggested basing it on a time-average surface temperature. Even though, as Mr. Paschkis points out, this method lacks proof, it is nevertheless based on what appears to the author to be much more reasonable assumptions than those which are usually made. In fact, to the author's knowledge, it is the only method advanced so far which is not based on hit or miss guessing.

In regard to Mr. Paschkis' query as to the advantages of the present method of solving temperature-uniformity problems over that previously used: The two methods are in fact complementary. In the previous method heating times and in the present furnace temperatures can be found directly without resorting to Gurney-Lurie type charts. The present method covers a wider range of variables and requires just one simple chart, Fig. 1, in place of the three—each with a family of curves—which are required in the method of Paschkis.

Mr. Trinks is at a loss to understand how the heat-transfer coefficient h can be obtained when both the surface temperature and the flame temperature are alike. By definition, for black-body radiation

$$h = 0.173\,\frac{[(T_a/100)^4 - (T_s/100)^4]}{T_a - T_s}$$

where

T_a = flame temperature in degrees Rankine (F + 460)
T_s = surface temperature in degrees Rankine
h = Btu per sq ft, hr, F

After factoring $T_a - T_s$ from the numerator, this reduces to

$$h = \frac{0.173}{(100)^4}(T_a^2 + T_s^2)(T_a + T_s)$$

and when T_s is equal to T_a

$$h = \frac{(0.173)\ (4)\ (T_a)^3}{(100)^4}$$

Although h is finite, there can be no net exchange of heat because no temperature difference exists.

Dynamic Behavior and Design of Servomechanisms[1]

By G. S. Brown[2] and A. C. Hall,[3] Cambridge, Mass.

This paper aims to set forth the criteria that are important in the design of certain servomechanisms and to show how certain analytical procedures can be applied to almost any problem to define quantitatively the properties that the control should have in order to make the servomechanism perform in the manner desired. The nomenclature used is essentially that proposed by Draper (1).[4] It is used in the treatment of the problem of synthesizing a controller for a servomechanism whose controlled member is to be rotated and can be defined in terms of inertia plus viscous and coulomb friction, plus a load torque. The broad objective of the designer is assumed to be that of designing a controller which will keep the dynamic error in controlled angular position of the controlled member a minimum under certain operating conditions. The transient and steady-state error for certain forms of motion, the criteria for stability, the effects of viscous friction, coulomb friction and load torques on the desired performance, and other related matters are treated. Several nondimensional charts and tables are included in the paper and show quantitatively the performance of any particular servomechanism with a controlled member of the kind treated, when the system has the various forms of control.

1 Introduction

Servomechanisms (2) or, more generally, closed-cycle automatic-control systems have become during the past decade exceedingly important elements in, for example, manufacturing, the process industries (3, 4), the steering (5) and operation of ships and aircraft, and many important scientific devices (6). As the applications have been increased in number, the demands imposed on the reliability, precision, and speed of operation of the mechanisms have also been increased. Fortunately, many of these demands have been satisfied because of developments in the various branches of engineering and physics that have taken place simultaneously with the growth of automatic control, but at the same time the complexity of the mechanisms has often been increased. The result of all this has been that the design of servomechanisms or automatic-control systems has now become almost a science, so that for the skillful and straightforward design of a high-caliber system one requires, (*a*) a clear appreciation of the fundamental principles involved in the correct functioning of an automatic-control system, and (*b*) an accurate knowledge of the properties of the mechanisms of which it is composed.

The purpose of this paper is to set forth in fairly simple terms the criteria that are important in the design of servomechanisms and to show how certain simple analytical procedures can be applied to almost any system to define the properties that the control should have in order to make the mechanism perform in the manner demanded.

2 Factors That Characterize a Servomechanism

A servomechanism is a power-amplifying automatic-control system characterized by the presence of a control element that is actuated by some function of the "difference" between the response desired of the system and its actual response. In other words, the system is error-sensitive, and the control is some function of the "error" in the behavior or state of the system. Usually a continuous change in the actuating quantity or the input signal is to be followed by a continuous action of the controller. The input signals usually come from low-power sources and may be random with time, but are preferably continuous. Systems of this kind are closed-cycle continuous-control systems, and it is this closed-cycle property that identifies them as servomechanisms. Their component elements generally tend to be complex, and because of the dependence of the operation of their control upon the result of the control operation, their design must conform to certain basic principles.

Because of widespread application of servomechanisms the actuating and controlled quantities are found in diverse forms. In the servomechanisms used for the automatic steering of ships or aircraft the actuating quantity or input signal is given by a low-power device sensitive to the course setting on a magnetic or gyro compass, and the output or controlled quantity is the direction in which the ship is headed. In servomechanisms used for the control of an automatic die-cutting machine, for example, the cutting head might be required to follow automatically a pattern laid out on a master pattern, or the contour of an existing member, as in replica production. An application such as this would probably result in maintaining the angular position of a shaft operating at a high-power level automatically in synchronism (or in step) with the angular position of a shaft as established by a low-power mechanism. The input signal might be the angular position of a cam-operated shaft, and the controlled quantity or output might be the angular position of the lead screw carrying the cutting head. In practice the angular positions of the two shafts are compared by one of several means common in the art, a measuring device detects the error in the position of the output and establishes a control signal, a controller is actuated by this control signal and operates either directly on the shaft of the high-power member or through some accessory device to bring the output shaft into the desired position.

Confusion occasionally exists among those working in automatic control, concerning the designation of a servomechanism

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[1] The first portion of this paper was prepared during the summer of 1940, for the Subcommittee on Machine Design, Machine-Shop Practice Division of the A.S.M.E. At the request of the N.D.R.C. it was withdrawn from open publication and privately printed for restricted release. Sections 1 through 9 are substantially as originally written. Sections 10 through 12 have been added for this printing. Sections 13 through 19 have been adapted from an M.I.T. doctorate thesis by A. C. Hall and published for restricted distribution by the Technology Press.

[2] Director, Servomechanisms Laboratory, Massachusetts Institute of Technology.

[3] Assistant Professor, Electrical Engineering, Massachusetts Institute of Technology.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Instruments and Regulators Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

and a servo. In this paper the entire array of apparatus beginning with the input member and going through to the high-power output member and including all measuring and control equipment is called a servomechanism. In particular, the torque, force, or power-producing member that is actuated by the control to restore the equilibrium condition is called a servo, or servomotor.

3 Basic Design Problem

Since a servomechanism should perform in a manner that meets a certain specification, the design of the system should be carried out only after an analysis that yields performance data for each component mechanism, in a form readily subject to quantitative interpretation for design purposes, has been concluded. The component elements can be described in terms of certain quantities, and the analysis should be carried out in a manner that shows the relations that must exist among these quantities in order to provide (*a*) a system that is stable and (*b*) a system whose transient and steady-state operating errors are within the specified limits.

This work is aided by subdividing the system into its essential component mechanisms, for example, the input member, the error-measuring element, the controller, the controlled member, and the output member. With all systems, good judgment is necessary in carrying out this subdivision, and in the more complicated systems a rather detailed subdivision may be desirable (see section 10). A block diagram representative of the manner of interconnection of these component mechanisms is, as a rule, readily drawn. After the analysis has been completed the performance of the mechanisms assigned to the various boxes in the block diagram can be defined, and their design and construction eventually carried out. If the performance specifications are reasonable the designer is limited only by cost, space or weight restrictions, availability of devices having the desired rating and characteristics, and so forth. In other words, the designer's problem is basically that of reducing certain requirements to practice.

To illustrate the procedure for the analysis of a simple mechanism, consider the block diagram shown in Fig. 1. Here the entire system is represented by five elements: namely, the input member, the error-measuring device, the controller, the controlled member, and the output member. Each component element may comprise much apparatus but this is unimportant for the present. For a generalized treatment let the system be defined in terms of the symbols shown in Fig. 2, where

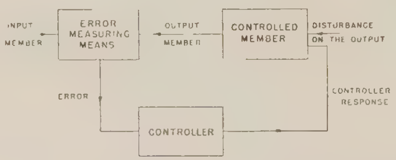


Fig. 1 Block Diagram for a Servomechanism

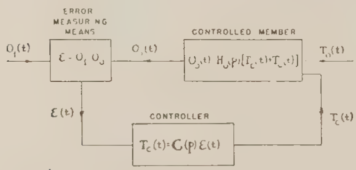


Fig. 2 Block Diagram for a Servomechanism Showing Symbolic Representation of Elements

$\theta_i(t)$ = input disturbance as a function of time
$\theta_0(t)$ = response of output as a function of time
$\epsilon(t)$ = error between desired and actual response as a function of time
$T_c(t)$ = response of controller as a function of time
$T_0(t)$ = any disturbance applied to output as a function of time

Also let these symbols be related in the following manner

$$\epsilon(t) = \theta_i(t) - \theta_0(t) \quad [1]$$

$$T_c(t) = C(p)[\epsilon(t)] \quad [2]$$

$$\theta_0(t) = H_0(p)[T_c(t) + T_0(t)] \quad [3]$$

In Equation [2] $C(p)$ is a differential operator in the form of a polynomial in p, where p represents d/dt. It operates on the time expression for the error and thereby gives the time expression for the controller response. It is called the "controller operator" and is to be determined from the analysis. $H_0(p)$ of Equation [3] is also an operator called the "output member operator." It is characterized by the physical properties of the device or process to be controlled.

Since, in general, the control is intended to keep the error zero, the extent to which the objectives of the control have been achieved is best indicated by deriving the expression for the error in terms of the operators $C(p)$ and $H_0(p)$, the input signal $\theta_i(t)$, and the output disturbance $T_0(t)$. From Equations [1], [2], and [3]

$$\epsilon(t) = \frac{1}{1 + C(p)H_0(p)}[\theta_1(t) - H_0(p)T_0(t)] \quad [4]$$

$$\theta_0(t) = \frac{C(p)H_0(p)\theta_i(t) + H_0(p)T_0(t)}{1 + C(p)H_0(p)} \quad [4a]$$

wherein $C(p)$ is frequently the only design variable. Equation [4] is the basic equation for a closed-cycle control system such as that shown in Fig. 2, and the expression $[1 + C(p)\,H_0(p)]^{-1}$, being an over-all system operator or characteristic equation, summarizes the system behavior. This use of an over-all system operator $[1 + C(p)H_0(p)]^{-1}$ to summarize the theory of design of a closed-cycle system of automatic control originated, to the authors' knowledge, with John Taplin in 1937, while he was a special student in electrical engineering at the Massachusetts Institute of Technology. Taplin no doubt recognized the similarity between the closed-cycle control problem and the feedback-amplifier problem as presented by Black (7) and Nyquist (8).

The applications of automatic control are numerous and varied and it is impossible to say that θ, T, or ϵ always represents any particular quantity. Reference to a few of the applications cited in the literature will show that θ, T, or ϵ may be an angular displacement, a flow, a voltage, a torque, a light flux or any of the physical quantities commonly encountered. However, since interest in an analysis is usually stimulated by speaking in terms of some physical application, the control of an automatic replica-cutting machine is considered as an example. It is assumed that $\theta_i(t)$ is an angular displacement indicated by a delicate cam-operated mechanism which displaces a low-power input shaft in accordance with the motion $\theta_0(t)$ desired of a high-power output shaft. The error $\epsilon(t)$ is assumed to be a voltage made proportional to error angle by any of the commonly used methods. The controller response $T_c(t)$ is assumed as a torque applied to the controlled member. The controlled member is a lead screw carrying a cutting head. Thus the controller, in effect, serves also as the prime mover or servomotor for the lead-screw drive. Fig. 3 shows a schematic arrangement of the essentials of this system.

The output member operator $H_0(p)$ can now be formulated. The cutting-head lead-screw drive of a typical milling machine

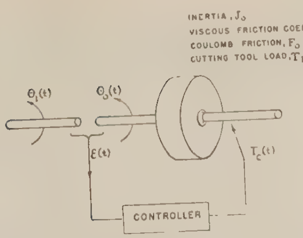


Fig. 3 Schematic Diagram of Servomechanism Used for Illustrative Example

would comprise shafts, screws, gears, and tool holder and could be defined in terms of an inertia J_0, and a viscous-friction coefficient f_0. The output member would be subjected to a coulomb friction torque and a load torque, which jointly can be represented as $-T_0(t)$. On the assumption that the parameters are constant or the system is linear for small ranges of operation, the differential equation of motion for the output member when considered as a single unit is

$$J_0 \frac{d^2\theta_0}{dt^2} + f_0 \frac{d\theta_0}{dt} = T_c(t) - T_0(t) \ldots\ldots\ldots [5]$$

It is expedient now to express $\frac{d\theta_0}{dt}$ by $p\theta_0$ and $\frac{d^2\theta_0}{dt^2}$ by $p^2\theta_0$, and to write Equation [5] as

$$(J_0p^2 + f_0p)\theta_0(t) = [T_c(t) - T_0(t)] \ldots\ldots\ldots [5a]$$

where the expression in parenthesis preceding $\theta_0(t)$ is an operator, which operates on $\theta_0(t)$ as indicated. When solved for $\theta_0(t)$, Equation [5a] gives

$$\theta_0(t) = \frac{1}{J_0p^2 + f_0p} [T_c(t) - T_0(t)] \ldots\ldots\ldots [6]$$

which, by analogy with Equation [3], gives the output member operator as

$$H_0(p) = \frac{1}{J_0p^2 + f_0p} \ldots\ldots\ldots [7]$$

The substitution of $H_0(p)$ from Equation [7] into Equation [4] gives, after some reduction

$$\epsilon(t) = \frac{1}{J_0p^2 + f_0p + C(p)} [(J_0p^2 + f_0p)\theta_i(t) + T_0(t)] \ldots [8]$$

Alternatively, if the input disturbance is a velocity

$$\omega_i(t) = \frac{d\theta_i(t)}{dt} = p\theta_i(t)$$

the error equation becomes

$$\epsilon(t) = \frac{1}{J_0p^2 + f_0p + C(p)} [(J_0p + f_0)\omega_i(t) + T_0(t)] \ldots [9]$$

Expressions typified by Equations [8] and [9] are extremely useful in the design or analysis of control systems since they express the error in terms of the parameters of the system, the input disturbance, the disturbing or load torques on the controlled member, and the controller operator $C(p)$. When any mechanism is to be converted to automatic control, the problem for the designer is the synthesis of the simplest and most practical controller that will have the operator $C(p)$ required to meet the specified operating conditions. The next sections present a generalized treatment of the problem of deriving a form of the operator $C(p)$ required by the various conditions of servomechanism operation commonly encountered in practice.

4 Simple Form of Controller Operator

In any particular application it would be desirable to know the expected form of the input disturbance and to choose the controller operator $C(p)$ on the basis of a certain specified performance under the expected operating conditions. In general, however, the input disturbance is random, so that its description in an analytical form which is of much use in the analysis is usually impractical. It is common practice therefore to evaluate the performance afforded by possible designs on the basis of the nature of the transient and steady-state responses of the respective systems to certain test disturbances.

The well-known transient response of a system defined by Equation [8] or [9] is

$$\epsilon(t) = A + Be^{r_1t} + Ce^{r_2t} + \text{etc.} \ldots\ldots\ldots [10]$$

A direct guide to the simplest form of controller operator that will give a workable system is afforded by a characteristic demanded of the exponents r_1 and r_2 of Equation [10]. For instance, r_1 and r_2 are the roots of the operator $J_1p^2 + f_0p + C(p)$ which may conveniently be a polynomial. Clearly the simplest form of $C(p)$ that will make this operator a polynomial is

$$C(p) = k \ldots\ldots\ldots [11]$$

and for the corresponding simple controller, Equation [2] reduces to

$$T_c(t) = k\epsilon(t) \ldots\ldots\ldots [12]$$

Also, for the system to be stable r_1 and r_2 must have negative real parts, and if they are complex the magnitudes of their imaginary parts must fall within a certain range because of operating restrictions. Hence the magnitude of k is obviously restricted.

A further restriction on k can be deduced from the steady-state operating error given by the quantity A of Equation [10], which can be written down by inspection from Equation [8] or [9]. For a constant input angle θ_i, or a constant input velocity ω_i, and constant load torque, the steady-state error is given by putting the operator p zero everywhere in Equations [8] and [9]. If then $C(p)$ is defined by Equation [11], the steady-state errors become, respectively

$$\epsilon_{ss} = \frac{T_0}{k} \ldots\ldots\ldots [13]$$

for θ_i and T_0 constant; and

$$\epsilon_{ss} = \frac{f_0\omega_i + T_0}{k} \ldots\ldots\ldots [14]$$

for ω_i and T_0 constant.

A controller having an operator as just defined applies torque to the controlled member in proportion to the indicated error. Since this definition makes no allowance for response lag in the controller, which here is assumed to include the servomotor, a perfect element is assumed. Actually, of course, response lags exist in all controller servomotors encountered in practice though frequently the lag is negligible in comparison with the periods of response of the system as a whole. Section 10 discusses the problem of treating the controller synthesis when the servomotor has response lags. Servomechanisms employing controllers characterized by Equation [11] are found frequently in practice. They are called simple-error controller servomechanisms and, as a rule, are confined to the class that involves prime movers rated less than a few watts for reasons that later are made clear.

The problem now to be handled is that of selecting the best value of k. Unfortunately, the selection involves a compromise because from Equations [13] and [14] high precision of operation requires that k be large and f_0 and T_0 small, whereas the stability restriction implied in Equation [10] imposes a limit on the largeness of k or the smallness of f_0, because the entire damping is here supplied by viscous friction in the output. Obviously the restrictions on the magnitude of T_0 are practical operating ones. The answers to these questions are best obtained from an examination of the transient response of the system, which is the subject of the next section.

5 Transient Response of Error-Controlled Servomechanism

The transient response of a servomechanism following the sudden application of a constant disturbance to the input member gives data that are useful (*a*) for appraising the quality of the performance of the system as a whole and (*b*) for proportioning f_0 and k to meet certain performance specifications. The response to this suddenly applied input disturbance completely characterizes the system.

The transient solutions are herein given as dimensionless expressions in a form convenient for engineering use, using the procedure and notation introduced by Draper (1). Coulomb friction and applied torque are neglected during the preliminary analyses since it is unnecessary to include them in order to evaluate the quality of the transient performance. Furthermore, their effects can be readily determined and taken into account by applying the principle of linear superposition of causes to give the total effects.

For identification purposes a servomechanism comprising a simple-error controller defined by Equation [12] is classified herein as type 1. For this controller, and with coulomb friction and load torque neglected, Equation [9] has the following form

$$\epsilon(t) = \frac{1}{J_0 p^2 + f_0 p + k_1}\,(J_0\,p + f_0)\omega_i(t) \qquad [15]$$

where the subscript 1 in Equation [15] and the following indicates that the parameters are for the system with a type 1 controller.

It is convenient now to define certain new symbols ζ_1 and ω_{n1} by the relations (1)

$$\zeta_1 = \frac{f_0}{2\sqrt{J_0 k_1}} \qquad [16]$$

$$\omega_{n1} = \sqrt{\frac{k_1}{J_0}} \qquad [17]$$

and to rewrite Equation [15] in terms of these symbols thus

$$\epsilon(t) = \frac{1}{p^2 + 2\zeta_1\omega_{n1}\,p + \omega_{n1}}\,(p + 2\zeta_1\omega_{n1})\omega_i(t) \qquad [18]$$

In Equation [18] ζ_1 is the damping ratio, that is, the ratio of the actual damping to the damping which would cause aperiodic response of the system. It is a dimensionless parameter and of course equals unity for critical damping; ω_{n1} is the undamped natural frequency of the system.

The response of the system defined by Equation [18] is oscillatory, aperiodic, or overdamped depending upon whether the roots of the denominator of Equation [18] when equated to zero, are conjugate complex, equal reals, or unequal reals, that is, whether ζ_1 is less than unity, unity, or greater than unity. The general form of the expression for the roots is

$$r_1,\ r_2 = -\zeta_1\omega_{n1} \pm j\omega_{n1}\sqrt{1-\zeta_1^2} \qquad [19]$$

$$= -\frac{1}{\tau_1} \pm j\omega_1 \qquad [20]$$

where τ_1 is the characteristic time and ω_1 is the actual angular frequency.

The complete solution of Equation [18] for the case when a velocity ω_i is suddenly applied to the input member with the system at rest and ω_i is afterward maintained constant, and when the roots r_1, r_2 are conjugate complex, is

$$\epsilon(t) = \frac{2\zeta_1\omega_i}{\omega_{n1}}\left[1 - e^{-\frac{t}{\tau_1}}\left(\cos\frac{\sqrt{1-\zeta_1^2}}{\zeta_1}\,\frac{t}{\tau_1} + \frac{2\zeta_1^2-1}{2\zeta_1\sqrt{1-\zeta_1^2}}\,\sin\frac{\sqrt{1-\zeta_1^2}}{\zeta_1}\,\frac{t}{\tau_1}\right)\right] \qquad [21]$$

Equation [21] thus gives the transient response and can be written in a nondimensional form convenient for plotting as

$$\frac{\epsilon(t)}{\epsilon_{ss}} = 1 - \frac{1}{2\zeta_1\sqrt{1-\zeta_1^2}}\,e^{-\zeta_1\omega_{n1}t}\sin\left(\sqrt{1-\zeta_1^2}\,\omega_{n1}t + \phi\right) \qquad [22]$$

where

$$\epsilon_{ss} = \frac{2\zeta_1}{\omega_{n1}}\,\omega_i \qquad [23]$$

and

$$\phi = \tan^{-1}\frac{2\zeta_1\sqrt{1-\zeta_1^2}}{2\zeta_1^2-1} \qquad [24]$$

For the critically damped case Equation [22] reduces to

$$\frac{\epsilon(t)}{\epsilon_{ss}} = 1 - e^{-\omega_{n1}t}\left(1 + \frac{\omega_{n1}t}{2}\right) \qquad [25]$$

where

$$\epsilon_{ss} = \frac{2\omega_i}{\omega_{n1}} \qquad [26]$$

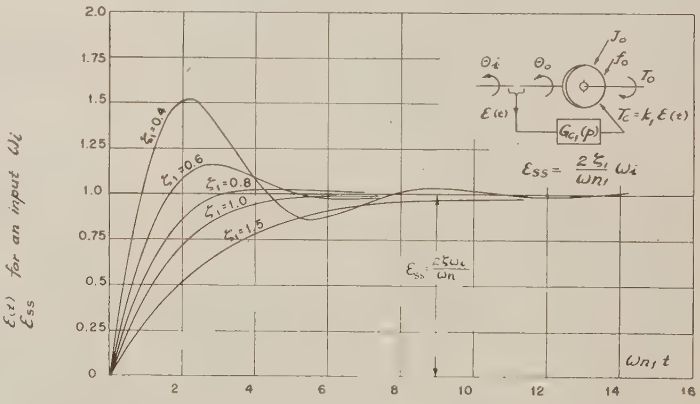


Fig. 4 Dimensionless Transient Error Curves for a Servomechanism With Type 1 Controller When Subjected to Suddenly Applied Input Velocity ω_i

The nature of the transient response is well illustrated by the family of curves in Fig. 4 which shows the dimensionless ratio $\frac{\epsilon(t)}{\epsilon_{ss}}$ as a function of the dimensionless quantity $\omega_{n1}t$ for various values of ζ_1. Load torque and coulomb torque are assumed to be zero. If the viscous-damping coefficient f_0 is assumed to be the only variable in the system, ω_{n1} is constant. Hence when the family of curves is examined with respect to the dimensionless abscissa scale $\omega_{n1}t$, the relative positions of the curves on the plot indicate the relative times of duration of the transient for the particular values chosen for ζ_1.

An inspection of Fig. 4 shows at once: (*a*) that for a type 1 servomechanism with constant ω_{n1} the actual time required for the system to reach substantially its steady-state error is practically independent of ζ_1 when $0.6 < \zeta_1 < 1.0$, whereas the time required for the system to attain any substantial part of its final steady-state error decreases as ζ_1 decreases; (*b*) that the overshoot or tendency to oscillate increases as ζ_1 is decreased; and (*c*) that the steady-following error is proportional to the ratio ζ_1/ω_{n1}. It follows therefore that any design of a type 1 controller is inherently a compromise unless certain specific performance requirements predominate, such for example as a given steady-state error at a given maximum velocity of operation of a given controlled member.

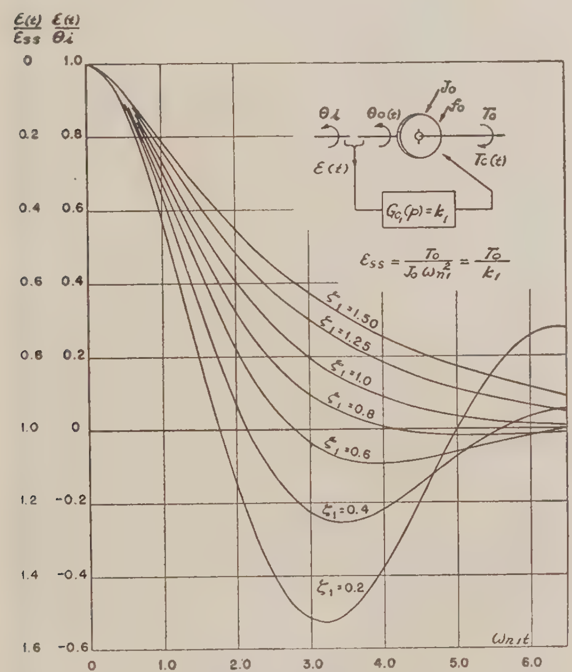


Fig. 5 Dimensionless Transient Error Curves of a Servomechanism With a Type 1 Controller When Subjected to a Suddenly Applied Input Angle θ_i or a Suddenly Applied Load Torque T_0

Fig. 5 shows the dimensionless solutions of Equation [8] for the angle and torque disturbances considered separately. Notice that a single family of curves satisfies both solutions if the ordinate scale for $\epsilon(t)/\theta_i$ for an angle disturbance is inverted to give the ordinate scale $\frac{\epsilon(t)}{\epsilon_{ss}}$ for the torque disturbance.

6 Illustrative Example

The effectiveness with which the foregoing data may be used to direct the path of a typical design, and also the fundamental objection usually leveled at mechanisms with a type 1 controller, are readily demonstrated by the following example:

Suppose a given output member is to be automatically controlled in angular velocity and position and that the controller is to be capable of driving this member at 120 rpm with a steady-state following error, due only to viscous friction, of not greater than π radians. Assume that the moment of inertia of the controlled member plus its connected servomotor is 1 slug ft². Assume also that the system is to be critically damped. Determine the control constant k_1, and the viscous friction coefficient f_0.

From Equation [23]

$$\omega_{n1} = \frac{2\zeta_1\omega_1}{\epsilon_{ss}} = \frac{2 \times 4\pi}{\pi} = 8 \text{ radians per sec} \qquad [27]$$

From Equation [17]

$$k_1 = \omega_{n1}^2 J_0 = 64 \text{ lb-ft per radian} \qquad [28]$$

From Equation [16]

$$f_0 = 2\zeta_1\sqrt{J_0k_1} = 16 \text{ lb-ft sec per radian} \qquad [29]$$

From the curves in Fig. 4 the system reaches the steady-state velocity in a dimensionless time $\omega_{n1}t$ of approximately 5.5, or an actual time t of $5.5/8$ = approximately 0.7 sec.

The design constants and the time of duration of the transient are now known quantitatively. However, when it is realized that the power dissipated in viscous friction at maximum speed is $f_0\omega_i^2$, which in practical units is

$$\frac{16 \times (4\pi)^2}{550} \text{ or } 4.57 \text{ hp} \qquad [30]$$

the objection to a type 1 mechanism is realized. It is clearly apparent that the designer faces a difficult task when he comes to devise means to dissipate this power in viscous friction and to devise a controller that will be essentially instantaneous and linear in operation and develop torques as great as 200 lb-ft. Furthermore, the entire viscous power serves only to damp the system, and serves no real productive purpose.

Smaller damping losses are achieved by resorting to underdamped operation, accepting a larger error, or decreasing inertia. For the same steady-state error and inertia, but with ζ_1 reduced to 0.6, the new constants are

$$\left.\begin{aligned} \omega_{n1} &= 4.8 \text{ radians per sec} \\ k_1 &= 23 \text{ lb-ft per radian} \\ f_0 &= 5.75 \text{ lb-ft sec per radian} \end{aligned}\right\} \qquad [31]$$

The task of the designer has now been eased in proportion to ζ_1^2, and the power dissipated in damping reduced to 1.65 hp. The transient now exists for a dimensionless time $\omega_{n1}t$ of approximately 5, so that $t = 5/4.8$, or about 1 sec, compared with 0.7 sec formerly. The power lost in damping is still appreciable, however, and is the reason why viscous-damped servomechanisms are rarely built for power ratings greater than a few hundred watts.

7 Controllers in Absence of Output Damping

The basic-error equation will now be examined to see whether the controller characteristic can be modified to permit the elimination of output damping power loss. Assume first that substantially all the viscous damping in the output member is eliminated by appropriate modifications to the design. The damping term then disappears from the denominator of Equation [9], the damp

ing ratio ζ_1 becomes substantially zero, the system tends to permit sustained oscillations and hence is impracticable. If by appropriate redesign of the controller, however, a new controller operator $C(p)$ can be obtained which has a term proportional to the time derivative of the error as well as to the error, a damping term is re-established in the denominator. The controller operator then has the form

$$C(p) = k + lp \qquad [32]$$

and for the system to be stable it is necessary only that the magnitude of the coefficient l be such that the roots of the expression

$$J_0{}^2 + C(p) = 0 \qquad [33]$$

have negative real parts.

This method of arriving at a desirable controller characteristic, while perhaps somewhat heuristic, is actually a very useful one because once the need for improvement in servomechanism performance is realized, the possibilities for achieving it by the design of a controller having the appropriate operator $C(p)$ become obvious after relatively little thought.

The use of control coefficients proportional to the derivative of the error is not new. The features which make this so-called derivative control desirable were pointed out by Minorsky (5) as early as 1922, and used by him in experiments on the automatic steering of ships prior to 1930. Later treatments are given by Hazen (6), and Mitereff (23). For purposes of identification a controller whose operator involves only the error and its derivatives is herein classified as type 2. To increase the scope of the analysis both first- and second-derivative control are assumed and the controller operator is written as

$$C_2(p) = k_2 \pm l_2p \pm m_2p^2 \qquad [34]$$

In this expression k_2, l_2, and m_2 are positive real numbers, the subscripts 2 signify control coefficients for a type 2 controller, and the algebraic signs signify the possibility of designing the controller to apply components of torque in a positive or a negative direction for a given error or time rate of change of error. Obviously, there is no choice regarding the algebraic sign before k_2.

The substitution of the expression for $C_2(p)$ in Equation [9] gives, when coulomb friction and load torque are neglected

$$\epsilon(t) = \frac{(J_0p + f_0)\omega_i(t)}{(J_0 \pm m_2)p^2 + (f_0 \pm l_2)p + k_2} \qquad [35]$$

$$= \frac{(J_0p + f_0)\omega_i(t)}{J_2p^2 + f_2p + k_2} \qquad [36]$$

where

$$J_2 = J_0 \pm m_2 \quad \text{and} \quad f_2 = f_0 \pm l_2 \qquad [37]$$

When Equation [36] is written in terms of nondimensional parameters, the expression is

$$\epsilon(t) = \frac{\sigma_2p + 2\alpha_2\zeta_2\omega_{n2}}{p^2 + 2\zeta_2\omega_{n2}p + \omega_{n2}{}^2}\,\omega_i(t) \qquad [38]$$

where

$$\left.\begin{aligned} \sigma_2 &= \frac{J_0}{J_2}, & \zeta_2 &= \frac{f_2}{2\sqrt{J_2k_2}} \\ \alpha_2 &= \frac{f_0}{f_2}, & \omega_{n2} &= \sqrt{\frac{k_2}{J_2}} \end{aligned}\right\} \qquad [39]$$

The general form of the roots of the denominator of Equation [38] are

$$r_1, r_2 = -\zeta_2\omega_{n2} \pm j\omega_{n2}\sqrt{1-\zeta_2{}^2} = \frac{1}{\tau_2} \pm j\omega_2 \qquad [40]$$

The solution of Equation [38] in dimensionless form for the case of a velocity ω_i suddenly applied to the input member with the system at rest, with ω_i afterward maintained constant, and with ζ_2 less than unity is

$$\frac{\epsilon(t)}{\epsilon_{ss}} = \left[1 - e^{-\frac{t}{\tau_2}}\left(\cos\frac{\sqrt{1-\zeta_2{}^2}}{\zeta_2}\frac{t}{\tau_2} + \frac{2\alpha_2\zeta_2{}^2 - 1}{2\alpha_2\zeta_2\sqrt{1-\zeta_2{}^2}}\sin\frac{\sqrt{1-\zeta_2{}^2}}{\zeta_2}\frac{t}{\tau_2}\right)\right] \qquad [41]$$

With critical damping the solution is

$$\frac{\epsilon(t)}{\epsilon_{ss}} = 1 - e^{-\frac{t}{\tau_2}}\left(1 + \frac{t}{\tau_2}\right) + \frac{1}{2\alpha_2}\frac{t}{\tau_2}e^{-\frac{t}{\tau_2}} \qquad [42]$$

where in each instance

$$\epsilon_{ss} = \frac{2\alpha_2\zeta_2}{\omega_{n2}}\,\omega_i \qquad [43]$$

Equations [41] and [42] therefore indicate the transient response of a servomechanism with a type 2 controller when subjected to a velocity disturbance.

When obtaining the transient response of a servomechanism with a controller which has derivative response, it should be remembered that $p^2\theta_i$ is an infinite impulse at $t = 0$ when a velocity ω_i is suddenly applied, and that both $p\theta_i$ and $p^2\theta_i$ are infinite impulses at $t = 0$ when an angle is suddenly applied. If the law of control given by Equation [34] is assumed to hold at all times, and these particular test disturbances are applied, the controller is required to deliver an infinite torque for zero time at $t = 0$. This, of course, is a requirement that no physical controller can fulfill. Thus a word of warning is appropriate here concerning the matter of using the methods of Heaviside's operational calculus (2, 9) to obtain the solution of expressions of the form of Equation [38] for an applied step function of velocity or angle. Operational calculus methods as ordinarily applied will assume the controller to be ideal and to comply with the law of control at all times. Solutions performed using the methods of LaPlace (9, 22) will introduce the boundary conditions applicable to the particular system conditions. The results given herein for the transient response of servos using derivative control recognize (*a*) that the maximum torque of the controller is limited and (*b*) that a finite torque applied to a mass member for zero time causes no change in its velocity or angle. The solution given by Equations [40] and [41] are derived on the basis that at a time immediately following the application of a suddenly applied velocity, that is, at $t = 0^+$

$$\left.\begin{aligned} &\epsilon, \theta_i, \text{ and } \theta_0 = \text{zero};\ p\epsilon = p\theta_i = \omega_i \\ &p\theta_0 = \text{zero};\ p^2\theta_i = \text{zero} \\ &p^2\epsilon = -p^2\theta_0 \end{aligned}\right\} \qquad [44]$$

An examination of Equations [36], [41], and [42] indicates certain points regarding the algebraic sign which should be used with the coefficients l_2 and m_2 of Equation [34] to give the best system performance. For example, since the coefficient f_2 in Equation [36] must be positive to insure stability, and since the principal argument for derivative control is to allow output damping to be negligible, only the use of the positive algebraic sign with the coefficient l_2 has practical significance.

For design purposes quantitative data concerning the effects of varying the magnitude of first-derivative response in the controller are highly desirable. If it is assumed, in the interests of simplicity, that the entire damping is supplied by l_2, (the so-called derivative damping) α_2 is zero and the steady-state error as given by Equation [43] is zero. Equations [41] and [42] for the transient response can then be written in a form convenient for the study of the effects of varying l_2 as

$$\frac{\epsilon(t)\omega_{n2}}{\omega_i} = \frac{1}{\sqrt{1-\zeta_2^2}} e^{-\zeta_2\omega_{n2}t} \sin\sqrt{1-\zeta_2^2}\,\omega_{n2}t \Bigg|_{\alpha_2=0} \quad \ldots [41a]$$

for the oscillatory case, and as

$$\frac{\epsilon(t)\omega_{n2}}{\omega_i} = \omega_{n2}t\, e^{-\omega_{n2}t} \Bigg|_{\alpha_2=0} \quad \ldots\ldots\ldots\ldots [42a]$$

for the critically damped case.

The nature of the transient response of the system now obtained is well illustrated by the family of curves in Fig. 6 which shows the plot of $\frac{\epsilon(t)\omega_{n2}}{\omega_i}$ as a function of the dimensionless quantity $\omega_{n2}t$ for various values of ζ_2. To the extent that angle is considered dimensionless the curves in Fig. 6 are dimensionless. Load torque and coulomb friction are assumed zero in addition to output damping being zero. Since the abscissa variable ω_{n2} is not affected by changes in l_2 this family of curves indicates directly the true relative times of duration of the transient for the values of ζ_2 assumed. The curves also show some rather interesting conditions. For example, the error is substantially independent of ζ_2 when $\omega_{n2}t$ is approximately 3. Also any benefits which might result from attempting to decrease the time of the duration of the transient by decreasing ζ_2 are offset to some extent by an increase in the peak magnitude of the transient error. Specifically, a decrease in ζ_2 from unity to 0.8 decreases the time of the duration of the transient by about 30 per cent, but increases the peak error during the transient by about 13 per cent. Since a decrease in ζ_2 tends to decrease the controller design difficulties and costs, there appears therefore to be genuine merit in the use of a type 2 controller adjusted to give values of ζ_2 such that $0.6 < \zeta_2 < 1.0$, unless a system that oscillates even to only a slight extent is barred because of circumstances peculiar to the particular application.

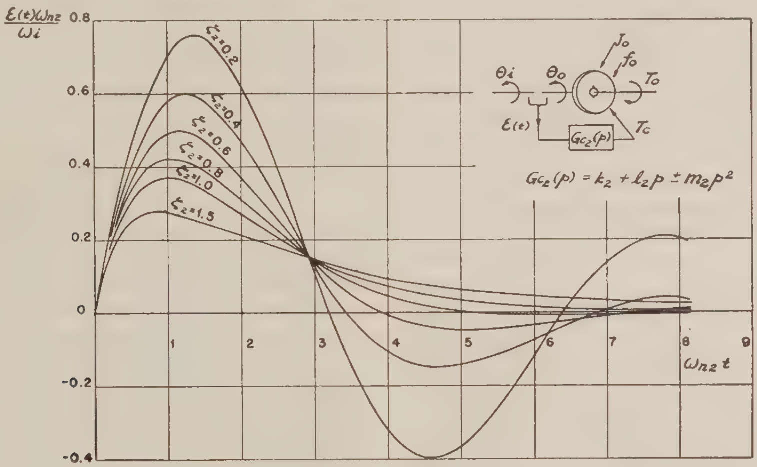


Fig. 6 Transient Error Curves for a Servomechanism With a Type 2 Controller With $\alpha_2 = 0$ and With σ_2 Constant, When Subjected to a Suddenly Applied Input Velocity ω_i

The task of making available a controller that will establish the component of first-derivative response with a high degree of fidelity is frequently somewhat difficult. When it comes to making available a controller that will establish a component of second-derivative response the task becomes one of another order of difficulty. Even the problem of providing a component of response that merely resembles second derivative is frequently difficult. It is highly desirable therefore that a designer have available quantitative data to aid him in balancing the cost of producing a certain amount of second-derivative response against the improvement in the performance of the system which the use of this amount affords.

This matter concerns the question of both the algebraic sign and the magnitude of m_2 which should be used in any particular system. On a heuristic basis it might be argued that the introduction of second-derivative control is in effect the introduction of inertia. Negative inertia as afforded by the use of the negative sign with m_2 might therefore appear to be preferable in order to decrease the magnitude of the apparent inertia of the system. On the basis of this argument a coefficient m_2 equal in magnitude to J_0 and used with the negative algebraic sign might appear to provide the optimum adjustment. It should be clear, however, that values of m_2 greater than J_0 used in conjunction with the negative algebraic sign give negative values of J_0, make the system unstable, and hence are barred. Unfortunately, this argument is inadequate for appraising quantitatively the benefits of specific amounts of second-derivative response.

The work involved in preparing quantitative data for this problem can be simplified without much loss generality of the results by again assuming output damping to be zero, thereby making α_2 and the steady-state error zero. If both l_2 and k_2 are held constant as $\pm m_2$ is varied, it follows readily from Equations [39] that ζ_2 and ω_{n2} are proportional to $\sqrt{\sigma_2}$, and hence dependent upon $\pm m_2$. Since the scales of ordinate and abscissa in Fig. 6 involve ω_{n2}, Equations [41a] and [42a] are not in a particularly convenient form for studying the effects of varying amounts of second-derivative control. It is possible, however, to express the error in a form which does not involve ω_{n2} by using a new characteristic time τ_2' defined as

$$\tau_2' = \frac{f_0 \pm l_2}{k_2} = \frac{2\zeta_2}{\omega_{n2}}$$

giving

$$\tau_2' = 2\zeta_2^2\tau_2 \ldots\ldots [45]$$

In terms of this new characteristic time a dimensionless form of the error equation convenient for studying the effect of varying $\pm m_2$ is

$$\frac{\epsilon(t)}{\omega_i\tau_2'} = \frac{1}{2\zeta_2\sqrt{1-\zeta_2^2}} e^{-\frac{2\zeta_2^2 t}{\tau_2'}} \sin 2\zeta_2\sqrt{1-\zeta_2^2}\,\frac{t}{\tau_2'} \ldots [41b]$$

for the oscillatory case and

$$\frac{\epsilon(t)}{\omega_i\tau_2'} = \frac{t}{\tau_2'} e^{-\frac{2t}{\tau_2'}} \ldots\ldots [42b]$$

for the critically damped case.

Fig. 7 gives a plot of a family of curves of the dimensionless quantity $\frac{\epsilon(t)}{\omega_i\tau_2'}$, as a function of the dimensionless time $\frac{t}{\tau_2'}$, for various values of ζ_2. Load torque and coulomb friction are assumed zero in addition to output damping being assumed zero. Neither the ordinate nor the abscissa variables are here functions of $\pm m_2$. Therefore the family of curves indicates directly the true relative times of duration of the transient and the true relative magnitudes of the peak transient error for the assumed values of ζ_2. It must be remembered, however, that variations in the magnitude or the algebraic sign used with the component of second-derivative response of the controller vary ζ_2, since ζ_2 is proportional to $\sqrt{\sigma_2}$. Changes in controller design which affect the second-derivative response are nevertheless readily transferred into quantitative data giving the changes in the performance of the system. The procedure is to compare the response indicated by the curve for the damping ratio in effect before the design is changed, say, ζ_{2a}, and the curve for a ζ_2 equal to $\zeta_{2a}\sqrt{\sigma_2}$. For example, assume that a particular design has $\zeta_{2a} = 0.5$. Then assume that a proposed design modification will change $\pm m_2$ by an amount sufficient to increase σ_2 by a factor of 4. The system then has $\zeta_2 = \sqrt{4}\zeta_{2a} = 1.0$, and the effect of the design change on the relative magnitudes of peak transient error and the times of duration of the transient is shown quantitatively by the curves for $\zeta_2 = 0.5$ and $\zeta_2 = 1.0$, respectively. At first glance these results may appear to be strange for they show that a system originally underdamped and having a rather large peak error can be made to have a substantially smaller peak error, be less oscillatory, and be substantially faster in response by increasing σ_2.

The data in Figs. 6 and 7 serve (*a*) to confirm the heuristic conclusion that the negative algebraic sign should be used with m_2; (*b*) to show that the magnitude of m_2 must be substantial if a substantial decrease in the time of the duration of the transient be brought about; and (*c*), that operation with less than critical damping justifies consideration. It should be remembered, however, that it is probably undesirable to approach too close to the condition in which $\sigma_2 > 10$, that is, $m_2 = 0.9\ J_0$, because of the likelihood that the system will become unstable if accidentally J_2 becomes negative.

The optimum damping ratio for use in any application depends, of course, on the magnitude of the transient departure from the steady-state error that is permissible. Since there is likely to be both viscous and coulomb friction in the output member, the steady-state error is actually finite. A net damping ratio such that $0.6 < \zeta_2 < 0.8$ is then generally satisfactory from the standpoint of small overshoot and preferable from the standpoint of speed of response, because the error is brought within the steady-state operating band about as rapidly as is practicable.

On the basis of the foregoing arguments and the data in Figs. 6 and 7 there should be no doubt about the economic value of introducing a component of first-derivative response into the controller, since by doing so the cost and difficulty of adding damping to the output member are eliminated, the unusually large power rating required of the servomotor, and the ridiculously low over-all efficiency of such a system are obviated. The data herein given permit the designer to evaluate quantitatively the relative merits of first- and second-derivative response when designing a specific system. Considering the practical difficulties that are encountered when the task of designing a controller to yield a second-derivative response with high fidelity is undertaken, there are, no doubt, many instances where the returns hardly justify the costs. Obviously circumstances alter cases, and in many important control applications it is entirely proper to pay a rather large premium to decrease the time of duration of the transient or the magnitude of the peak transient error by a factor of even 2 or less. At such times only a moderate amount of control response, which even then only approximates second-derivative control, may prove to be extremely valuable.

Transient solutions for angle or torque disturbances applied to a system with a type 2 controller are readily obtained by the procedure given. For the boundary conditions at the instant immediately following the application of torque or angle, that is, at $t = 0^+$ such that

$$\epsilon = \theta_i;\ \theta_0 = \text{zero};\ p\epsilon = p\theta_i = p\theta_0 = \text{zero}$$

$$p^2\epsilon = -p^2\theta_0 \text{ and } p^2\theta_i = \text{zero}$$

the time solutions are identical for those obtained for a system with a type 1 controller with the exception that ζ_2 and ω_{n2} replace ζ_1 and ω_{n1}, respectively. The curves in Fig. 5 therefore apply to a type 2 controller if ζ_2 and ω_{n2} are used instead of ζ_1 and ω_{n1}.

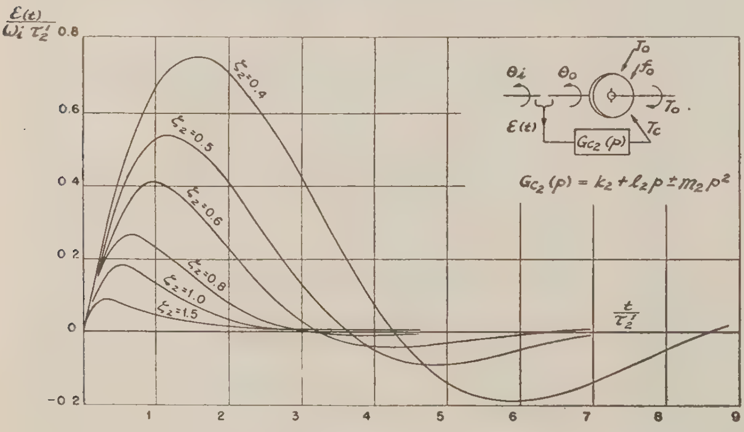


FIG. 7 DIMENSIONLESS TRANSIENT ERROR CURVES FOR A SERVOMECHANISM WITH A TYPE 2 CONTROLLER WITH $\alpha_2 = 0$ AND WITH l_2 CONSTANT, WHEN SUBJECTED TO A SUDDENLY APPLIED INPUT VELOCITY ω_i

8 CONTROLLERS TO COUNTERACT COULOMB OR LOAD TORQUES

In control applications where the output member is principally inertia, an acceptable steady-state error is frequently obtained by decreasing the viscous and coulomb frictions in the output member to a practical minimum and introducing derivative control to establish transient stability. At such times type 2

control is adequate. However, in applications such as precision machine-tool control, calculating machine servomechanisms, and so forth, exceedingly precise performance is often demanded even when the output member is required to operate against a substantial disturbing or load torque T_L, in addition to coulomb friction. At such times it is really difficult to obtain highly precise performance.

Although only mechanical systems have been discussed thus far, it should be remembered that where the mechanisms involve electrically operated elements or hydraulic elements, resistance in the electric circuits or fluid circuits is just as effective in causing steady-state error as mechanical viscous friction. Similarly, the induced voltage of an electromagnetic machine, or the operating fluid pressure of a hydraulic system, and similar phenomena, usually cause effects analogous to those of an output load torque. Also, magnetic hysteresis in the magnetic circuits of electromagnetic machines, and dielectric hysteresis in the dielectric of capacitive elements, are analogous to coulomb fricton and equally undesirable. Thus there is inherently a definite limit to the degree of precision with which almost any servomechanism having type 1 or type 2 controller can operate.

If means to combat these difficulties are to be devised they can come about only as a result of a modification to the controller, since the controller characteristic is really the only design variable. Accordingly the basic-error equation will again be examined, but this time with a view to devising for the controller operator $C(p)$ an expression that will insure not only a stable and rapid system, but one which will counteract the effects of coulomb or load torques.

For the system under consideration the error equation is, by analogy with Equation [9]

$$\epsilon(t) = \frac{(J_0p + f_0)\omega_i(t) + F(\omega) + T_L(t)}{J_0p^2 + f_0p + C(p)} \quad \text{[46]}$$

where the symbol F signifies coulomb friction torque which is a function of the velocity of the output shaft. Since the direction of this torque reverses as the direction of rotation reverses, Equation [46] is nonlinear and is difficult to handle mathematically. If, however, it is assumed that coulomb friction can be represented as a constant torque which appears only after t = zero, the analysis becomes relatively straightforward and the results are still sufficiently general to be useful. On this basis the error equation can be written for a velocity input as

$$\epsilon(t) = \frac{(J_0p + f_0)\omega_i(t) + F + T_L}{J_0p^2 + f_0p + C(p)} \quad \text{[47]}$$

For optimum performance it is desirable that the controller counteract the results of F and T_L during the period of transient operation as well as during the steady state. The transient-error behavior is indicated to a considerable extent by the nature of the roots of the denominator of Equation [47]. The steady-state error for a constant disturbance, however, is given directly by the foregoing error equation if p is put equal to zero after the equation is converted to the ratio of two polynomials. Therefore from the mathematical viewpoint at least, the steady-state error would be made zero if the controller were modified so that the law of control would have the effect of multiplying F and T_L by p in Equation [47]. Then when this is done an attempt might be made to adjust the magnitudes of the resulting function $C(p)$ to give the desired transient performance. Thus both objectives would be achieved.

The process of synthesizing a controller operator $C(p)$ which would achieve these objectives can logically start by multiplying the numerator and denominator of Equation [47] by p. The error equation then has the form

$$\epsilon(t) = \frac{p(J_0p + f_0)\omega_i(t) + p(F + T_L)}{p[J_0p^2 + f_0p + C(p)]} \quad \text{[48]}$$

The steady-state error is now zero, provided the denominator is a polynomial containing a constant term. The restrictions that may be imposed on the operator $C(p)$ to achieve satisfactory transient response are that the roots of the polynomial

$$\left.\begin{array}{ll} & p[J_0p^2 + f_0p + C(p)] = 0 \\ \text{or} & \\ & J_0p^3 + f_0p^2 + pC(p) \;\;= 0 \end{array}\right\} \quad \text{[49]}$$

have substantial negative real parts and, if complex, have magnitudes of the imaginary parts that provide reasonable periods for the oscillation of the system. Since one of the criteria to be satisfied in order for the roots to have negative real parts is that the polynomial have all the coefficients of the same algebraic sign, the simplest form for the function $pC(p)$ is

$$pC(p) = ap + b \quad \text{[50]}$$

where a and b are positive real quantities.

The result of this analysis has been to show that in order to counteract constant output torques a controller is required whose response is proportional to the error plus the time integral of the error. Actually, it has been known for some time that this response is of value in control problems of the type herein discussed, since it was demonstrated by Minorsky (5) as early as 1922 and treated by Mitereff (23) in 1935.

To those already familiar with automatic-control theory, moreover, it may appear that the foregoing analysis amounts to much ado about nothing This really is not so, however, for while to some persons it may have been obvious after a little thought that the inclusion of an integral term in the controller response would achieve the results desired in the simple example cited, it is not always so easy to reach reliable conclusions in more complicated control problems. On the other hand, the procedure here presented whereby the form of controller characteristic is deduced (*a*) by first examining the error equation with a view to seeing what properties it must have to yield the solution desired and (*b*) by manipulating it mathematically to give it these properties, is quite powerful and widely applicable in control problems. It readily yields data that are directly interpretable in terms of the design of a suitable controller.

In the analysis which follows the integral-plus-proportional controller is referred to as type 3. For the purposes of generality, and also in the interests of improved transient response of a servomechanism with this type of controller, the general form of the operator for type 3 control is taken as

$$C_3(p) = k_3 \pm l_3p \pm m_3p^2 + \frac{n_3}{p} \quad \text{[51]}$$

where k_3, l_3, m_3, and n_3 are all positive real quantities and, except for l_3 and m_3, they can be associated only with the positive algebraic sign because of stability criteria.

The substitution of $C_3(p)$ from Equation [51] in Equation [48] gives for the error of a servomechanism with a type 3 controller

$$\epsilon(t) = \frac{(J_0p^2 + f_0p)\omega_i(t) + p(F + T_L)}{J_3p^3 + f_3p^2 + k_3p + n_3} \quad \text{[52]}$$

where

$$J_3 = J_0 \pm m_3 \quad \text{and} \quad f_3 = f_0 \pm l_3 \quad \text{[53]}$$

9 Transient Solution of Servomechanisms With Type 3 Controller

Equation [52] does not readily yield a solution adaptable to simple graphical presentation from the design standpoint, because

of the large number of design variables and the complexity resulting from the third-order denominator. As Equation [52] stands, however, it shows that the steady-state error is zero for either a constant input angle, a constant input velocity, or a constant disturbing torque on the controlled member. However, if the input undergoes a constant acceleration, $p\omega_i$, the steady-state error is

$$\epsilon_{ss} = \frac{f_0}{n_3}(p\omega_i) \qquad [54]$$

The question of an ideal versus a nonideal controller arises when a solution of Equation [52] is attempted for a step-function disturbance of angle or velocity. It is seen by inspection of Equation [52] written in the form of a step function of angle, that the operational solution at $t = 0$ for the equation as it stands is

$$\epsilon(0) = \frac{J_0}{J_3}\theta_i$$

This result can be obtained by taking the limit of the error equation as p tends to infinity. For this value of $\epsilon(0)$ to occur the output member would have to undergo a finite displacement in zero time. No physical controller can accomplish this with an output member that comprises mass because the mass cannot be moved in zero time by only a finite torque. This argument is given only briefly for simplicity. If the differential equation of Equation [52] is solved by classical methods subject to physically realizable boundary conditions at $t = 0$, and especially that $\epsilon(0) = \theta_i(t)$ at $t = 0$, the solution will be correct. Alternatively, if methods of formulating the equation such as those of LaPlace (9) are followed, the term J_3 appears in the numerator of the error equation, and not J_0. For these reasons the liberty is taken here of writing the error equation in corrected operational form for a velocity disturbance as

$$\epsilon(t) = \frac{(J_3p^2 + f_0p)\omega_i + p(F + L)}{J_3p^3 + f_3p^2 + k_3p + n_3} \qquad [55]$$

and for an angle disturbance as

$$\epsilon(t) = \frac{(J_3p^3 + f_3p^2)\theta_i + p(F + L)}{J_3p^3 + f_3p^2 + k_3p + n_3} \qquad [56]$$

It is desirable to attain some simplification of Equations [55] and [56] before attempting a solution. Simplification occurs, partially at least, by writing the equation in a dimensionless form. Two forms have been found useful. One follows from a method originally indicated by Weiss (10). The other follows by recognizing that the factors of a cubic equation can be written as a quadratic times a real root. Both forms are helpful in design studies. Weiss has prepared useful charts (10) giving the roots of the cubic in forms that aid many design problems. Liu (11) and Evans (12) have prepared charts in the form that is of special significance when the cubic is written as a quadratic factor and a real root. These charts have been found somewhat more useful than those of Weiss, because they permit the solution to be written in terms of damping ratios and undamped natural frequencies, with the result that the data already given in dimensionless form for type 1 and type 2 servos can be made immediately applicable semiquantitatively simply by matching characteristics of the oscillatory components for ζ and ω_n.

The dimensionless form that follows from the work of Weiss writes the denominator of the type 3 servo

$$p^3 + \frac{f_3}{J_3}p^2 + \frac{k_3}{J_3}p + \frac{n_3}{J_3} = 0 \qquad [57]$$

in the form

$$p^3 + 2\zeta_c\omega_n p^2 + \omega_n^2 p + S\omega_n^3 = 0 \qquad [58]$$

where the terms ζ_c, ω_n, and S are merely defined by applying the approach used for nondimensionalizing the quadratic, and in the case of ζ_c and ω_n do not represent a damping ratio or undamped natural period. By definition

$$\left.\begin{aligned} S &= 2\zeta_3\gamma_3 = \frac{n_3}{k_3}\sqrt{\frac{J_3}{k_3}} \\ \zeta_c &= \frac{f_3}{2\sqrt{J_3k_3}}, \quad \omega_n = \sqrt{\frac{k_3}{J_3}} \end{aligned}\right\} \qquad [59]$$

where γ_3 comes from a stability ratio $\frac{J_3n_3}{f_3k_3}$ using Rouths' stability criteria.

The alternate form, as used by Liu and Evans, writes the denominator as

$$(p + \xi\omega_{nq})(p^2 + 2\zeta_q\omega_{nq}p + \omega_{nq}^2) = 0 \qquad [60]$$

where ζ_q and ω_{nq} now represent specifically the damping ratio and undamped natural period of the quadratic factor or oscillatory component.

The relations between S, ζ_c, ω_n and ξ, ζ_q, and ω_{nq} are

$$(2\zeta_q + \xi)\omega_{nq} = 2\zeta_c\omega_n = \frac{f_3}{J_3} \qquad [61a]$$

$$(2\zeta_q\xi + 1)\omega_{nq}^2 = \omega_n^2 = \frac{k_3}{J_3}; \text{ and } \xi\omega_{nq}^3 = S\omega_n^3 = \frac{n_3}{J_3} \qquad [61c]$$

An interesting application of the merits of these so-called dimensionless forms of equations results from a study of the following problem: Assume that integral response is to be introduced into a controller which initially has a ratio of output damping to total damping $\alpha_3 = 0.2$ and a damping ratio ζ_2 which is also $\zeta_c = 0.8$. Assume that it is desired to investigate the form of the transient response to a suddenly applied velocity or angle as integral response is introduced into the controller in amounts represented by $S = 0, 0.2, 0.4$. Since the servo without integral control has already been studied in terms of curves using $\omega_n t$ as the dimensionless time, see Figs. 4, 5, and 6, the same form of dimensionless time can again be used to advantage.

Table 1 shows the dimensionless solution for the error equation of the form

$$\epsilon(t) = \frac{[p^2 + \omega_{nq}\alpha_3(2\zeta_q + \xi)p]\omega_i}{(p + \xi\omega_{nq})(p^2 + 2\zeta_q\omega_{nq}p + w_{nq}^2)} \qquad [55a]$$

$$\epsilon(t) = \frac{[p^3 + \omega_{nq}(2\zeta_q + \xi)p^2]\theta_i}{(p + \xi\omega_{nq})(p^2 + 2\zeta_q\omega_{nq}p + \omega_{nq}^2)} \qquad [56a]$$

where load torques are neglected.

Figs. 8 and 9 show the solution of $\frac{\epsilon(t)\omega_n}{\omega_i}$ and $\frac{\epsilon(t)}{\theta_i}$ as a function of $\omega_n t$.

The numerical solutions were obtained using the cubic chart prepared by Liu (11). Values of ζ_c and S are known and are the values used to enter the chart for determining the values of ζ_q, ξ. Finally ω_{nq} is computed from S and ξ using the relation

$$\omega_{nq} = \left(\frac{S}{\xi}\right)^{1/3}$$

One observation predominates from an inspection of Figs. 8 and 9, namely, that a system which initially is well damped and

TABLE 1 RESPONSE OF A SERVOMECHANISM WITH A TYPE 3 CONTROLLER

For a suddenly applied input velocity disturbance ω_1

$$\varepsilon(t) = \frac{(J_3p^2 + f_0p)\,\omega_1}{J_3p^3 + f_3p^2 + k_3p + n_3}$$

$$\varepsilon(t) = \frac{\left[p^2 + a_3(2\zeta_q + \xi)\,\omega_{nq}\right]\omega_1}{(p + \xi\omega_{nq})\,(p^2 + 2\zeta_q\omega_{nq}p + \omega_{nq}^2)}$$

$$\frac{\varepsilon(t)\omega_n}{\omega_1} = k_1\varepsilon^{-\xi\omega_{nq}t} + k_2\varepsilon^{-\zeta_q\omega_{nq}t}\cos\left(\omega_{nq}\sqrt{1-\zeta_q^2}\,t + \theta\right)$$

$$k_1 = \frac{\sqrt[3]{\xi}\left[(2\zeta_q + \xi)\,a_3 - \xi\right]}{\sqrt[3]{S}\left[\xi^2 - 2\zeta_q\xi + 1\right]}$$

$$k_2 = \frac{\sqrt[3]{\xi}\sqrt{a_3^2\,(2\zeta_q + \xi)^2 - 2a_3\zeta_q\,(2\zeta_q + \xi) + 1}}{\sqrt[3]{S}\sqrt{(1-\zeta_q^2)\,(\xi^2 - 2\zeta_q\xi + 1)}}$$

$$\theta = \tan^{-1}\frac{\zeta_q - a_3\,(2\zeta_q + \xi)}{\sqrt{1-\zeta_q^2}} - \tan^{-1}\frac{\sqrt{1-\zeta_q^2}}{\xi^2 - \zeta_q}$$

For a suddenly applied input angular disturbance θ_1

$$\varepsilon(t) = \frac{(J_3p^3 + f_3p^2)\,\theta_1}{J_3p^3 + f_3p^2 + k_3p + n_3}$$

$$\varepsilon(t) = \frac{\left[p^3 + (2\zeta_q + \xi)\,\omega_{nq}p^2\right]\theta_1}{(p+\xi\omega_{nq})\,(p^2 + 2\zeta_q\omega_{nq}p + \omega_{nq}^2)}$$

$$\frac{\varepsilon(t)}{\theta_1} = k_1\varepsilon^{-\xi\omega_{nq}t} + k_2\varepsilon^{-\zeta_q\omega_{nq}}\cos\left(\omega_{nq}\sqrt{1-\zeta_q^2}\,t + \theta\right)$$

$$k_1 = \frac{-2\zeta_q\xi}{(\xi^2 - 2\zeta_q\xi + 1)}$$

$$k_2 = \frac{\sqrt{(1+\xi\zeta_q)^2 + (\xi\sqrt{1-\zeta_q^2})^2}}{\sqrt{1-\zeta_q^2}\sqrt{\xi^2 - 2\xi\zeta_q + 1}}$$

$$\theta = \tan^{-1}\frac{1+\zeta_q\xi}{\xi\sqrt{1-\zeta_q^2}} - \tan^{-1}\frac{\sqrt{1-\zeta_q^2}}{\xi - \zeta_q}$$

$$J_3 = J_o \pm m_3 \qquad G_c(p) = k_3 \pm \ell_3p \pm m_3p^2 \pm \frac{n_3}{p}$$

$$f_3 = f_o \pm \ell_3$$

$$\sigma_3 = \frac{J_o}{J_3}$$

$$a_3 = \frac{f_o}{f_3}$$

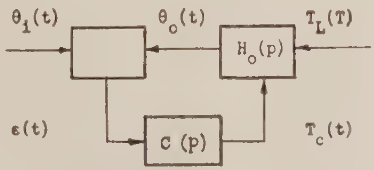
$$\zeta_3 = \frac{f_3}{2\sqrt{k_3J_3}}$$

$$2\zeta_3\,\omega_n = (2\zeta_q + \xi)\,\omega_{nq}$$

$$\omega_n^2 = (2\zeta_q\xi + 1)\,\omega_{nq}^2$$

$$S\omega_n^3 = \xi\omega_{nq}^3$$

$$\omega_{n3} = \sqrt{\frac{k_3}{J_3}}$$



fast becomes more oscillatory and faster as integral control is increased. To procure a reduction in the degree of oscillation, it is apparent immediately that ζ_q must be increased because this parameter controls the oscillatory component. A valuable feature of the dimensionless study using curves such as those prepared by Liu is now recognized because the curves of either Weiss or Liu quickly show the manner in which ζ_c or S must be varied to give the desired ζ_q. Although these parameters are dimensionless, their transformation into real parameters is simple so that actually the curves directly give quantitative data. It is interesting to note that there is always a value of ζ_c beyond which little improvement in ζ_q is obtained, thus necessitating a review of the whole design problem.

10 Systems Involving Energy Storage Elements in Controller or Output

In the preceding analysis the controller has been assumed to be an element that correctly obeys the law of control as given for $C(p)$. While this condition is approximated in many systems, certain inherent properties of physical elements cause the introduction of other considerations into the analysis. For example, the output stages of controllers, which involve power amplification or the servomotor driving the output member, will frequently include (*a*) inductance if it is electromagnetic, (*b*) elastance of oil lines or inertia of valves or pistons if it is hydraulic, (*c*) elastance of air lines if pneumatic. Thus energy storage and energy dissipation occur throughout the system with attendant so-called "response delays." The over-all controller response $T_c(t)$ is then not in phase with the signal that actually initiates the action of the servomotor elements, as is implied by Equation [12].

When the response of any element is fast compared with the system response, its response may be considered instantaneous because the energy storage can usually be neglected. When its response becomes comparable with the speed of system response it is necessary to modify the operator for the controller to introduce the effects of these energy storage elements into the analysis. To do this let a new operator $D(p)$ be used to describe the over-all performance of the controller-servomotor combination, where $D(p) = C(p)\,A(p)$ and in which $C(p)$ characterizes only the portion of the controller that receives the error signal $\epsilon(t)$ and operates on it to yield the proportional, derivative, or integral components of control, and $A(p)$ takes account of the energy storage or "response delay" in the controller output or the servomotor. Equation [2] now becomes $T_c(t) = C(p)\,A(p)\,\epsilon(t)$ and the over-all system operator from Equation [4*a*] becomes

$$1 + C(p)\,A(p)\,H_0(p)$$

As an illustration, consider that the controller comprises a

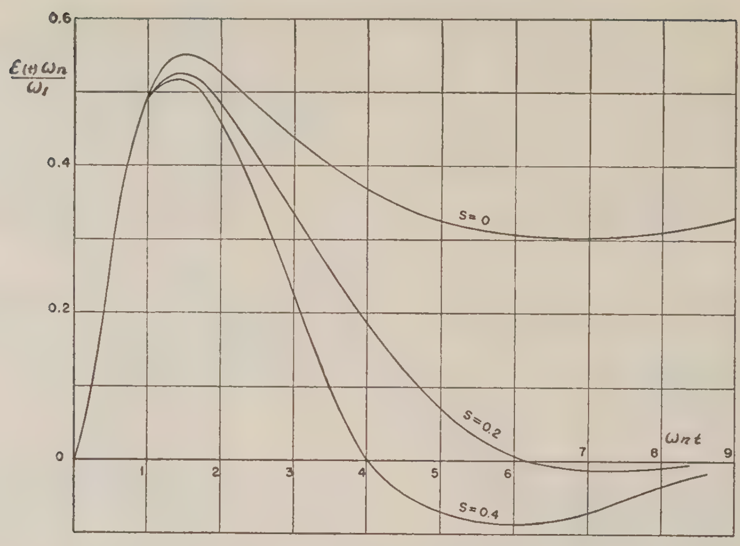


Fig. 8 Transient Error Curves for a Velocity Disturbance Showing Effect of Introducing Integral Control Into a Type 2 Servomechanism for Which Initially $\zeta_2 = 0.8$ and $\alpha_2 = 0.2$
(Amount of integral control is given by $S = 0$, 0.2, and 0.4.)

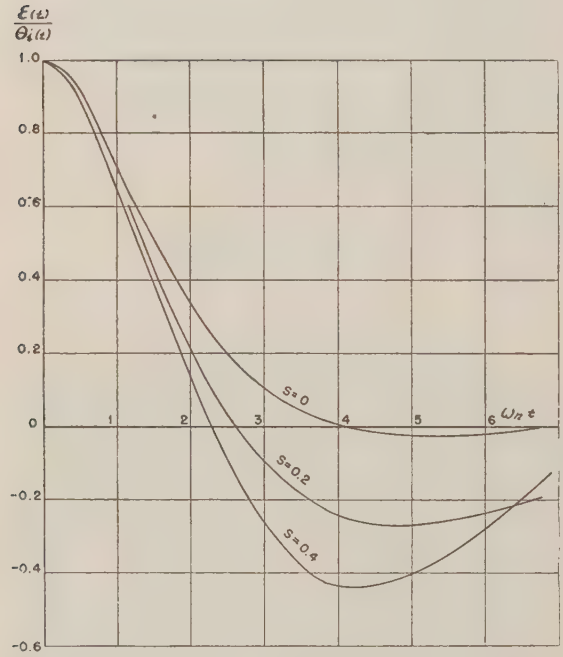


Fig. 9 Transient Error Curves for an Angle Disturbance, Showing Effect of Introducing Integral Control Into a Type 2 Servomechanism for Which Initially $\zeta_2 = 0.8$ and $\alpha_2 = 0.4$
(Amount of integral control is given by $S = 0$, 0.2, and 0.4.)

vacuum-tube amplifier that energizes the field of a generator supplying an electric servomotor. The operator $C(p)$ characterizes the vacuum-tube amplifier field circuit of the generator. If it is assumed that the build-up field flux in the generator is substantially in phase with the grid signal on the amplifier, the principal response lag will be in armature current. The build-up of servomotor torque will lag the generator-field flux because of inductance L_a and resistance R_a of the two armatures. The operator $A(p)$ characterizing the servomotor would then have the form

$$A(p) = \frac{K}{L_a p + R_a}$$

Thus for the system considered in Fig. 3, Equation [9] would take the form

$$\epsilon(t) = \frac{L_a p + R_a}{K(J_0 p^2 + f_0 p)(L_a p + R_a) + C(p)}\,[(J_0 p + f_0)\omega_i(t) + T_0(t)]$$

A typical block diagram for a system involving several ele-

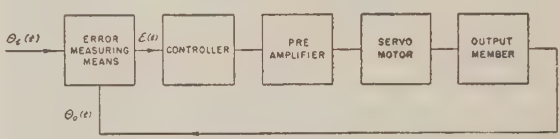


Fig. 10 Block Diagram for Servomechanism Comprising Several Energy Storage Elements in Cascade

ments, each with response lag, is shown in Fig. 10. The over all system operator for such a system is

$$1 + \left[\begin{matrix}\text{Controller}\\ \text{Operator}\end{matrix}\right]\left[\begin{matrix}\text{Preamplifier}\\ \text{Operator}\end{matrix}\right]\left[\begin{matrix}\text{Servomotor}\\ \text{Operator}\end{matrix}\right]\left[\begin{matrix}\text{Output}\\ \text{member}\\ \text{Operator}\end{matrix}\right]$$

which leads to a characteristic equation involving high powers of p for which the labor in getting transient solutions may be considerable. The matter is discussed in section 13.

11—Systems With Output Feedback

In many servomechanisms the error signal given by the error-measuring means is of such a form that it cannot even be easily differentiated once to give good control signals for stabilizing purposes, as previously discussed for a type 2 system, let alone be differentiated twice. In some cases the particular embodiment that the servomechanism takes makes it undesirable to employ

derivative networks. Frequently the error is of such form that good techniques for giving its time rate of change do not exist.

Methods of control that will yield the desired stability or dynamic performance without error differentiation therefore become necessary. The method most commonly used is to feedback (14) to the input a signal that is some function of the response of the servo output, or a response just ahead of the output. Fig. 11 shows the block diagram for a typical system involving

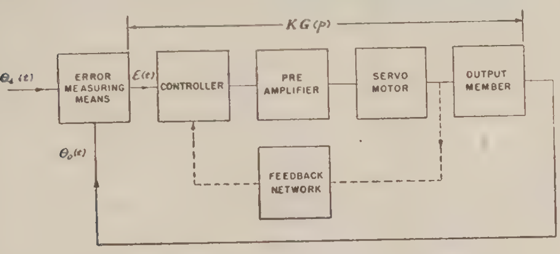


FIG. 11 BLOCK DIAGRAM FOR SERVOMECHANISM COMPRISING SEVERAL ENERGY STORAGE ELEMENTS IN CASCADE AND FEEDBACK WITHIN CLOSED LOOP

feedback. The particular functional relationship between the output and the feedback signals is established by an arrangement of elements inserted specifically for the purpose, and for which a feedback operator is known or can be deduced. No standard rule can be given for the form of the operator used for the feedback (14) since the choice of derivative, integrative, or even the number of feedback elements becomes a function of the specific application for which the servomechanism is intended.

As a general rule the introduction of feedback networks adds energy-storage elements into the system and increases the powers of p in the characteristic equation. By the application of the operator relations for the resulting network the over-all system operator becomes for Fig. 11

$$1 + \frac{\begin{bmatrix}\text{Controller}\\\text{Operator}\end{bmatrix}\begin{bmatrix}\text{Preamplifier}\\\text{Operator}\end{bmatrix}\begin{bmatrix}\text{Servomotor}\\\text{Operator}\end{bmatrix}\begin{bmatrix}\text{Output}\\\text{Operator}\end{bmatrix}}{1 - \begin{bmatrix}\text{Controller}\\\text{Operator}\end{bmatrix}\begin{bmatrix}\text{Preamplifier}\\\text{Operator}\end{bmatrix}\begin{bmatrix}\text{Servomotor}\\\text{Operator}\end{bmatrix}\begin{bmatrix}\text{Feedback}\\\text{Operator}\end{bmatrix}}$$

12 TRANSIENT ANALYSIS OF SYSTEMS INVOLVING HIGH-ORDER CHARACTERISTIC EQUATIONS

In the systems of the types discussed in sections 10 and 11 the characteristic equations invariably involve high powers of p. The transient analysis then becomes complicated because of (a) the difficulty of determining the parameters of all elements such as inertia, elastance, inductance, damping, amplification, torque gradient, and so forth, for each element in the system and (b) the labor of getting the roots and the time solutions to problems involving characteristic equation of high orders. While the analysis may be classified as tedious, the synthesis of controllers and feedback elements by the transient methods is difficult because the coefficients of the powers of p in the characteristic equation are a heterogeneous arrangement of many of the parameters of the elements of the system. Thus even when the work of obtaining the transient solution has been completed it is difficult to identify any particular element in the system that must be changed to cause any preselected modification in system behavior.

A qualitative knowledge of the transient behavior that is useful for many purposes may, however, be obtained by recognizing that the characteristic equation can be factored into the product of quadratic factors as follows

$$p^n + bp^{n-1} + cp^{n-2} + \ldots = (p^2 + 2\zeta_a\omega_{na}p + \omega_{na}^2)(p^2 + 2\zeta_b\omega_{nb}p + \omega_{nb}^2)\ldots(p + a)$$

Each quadratic factor contributes to a mode of oscillation in the solution having damping ratios ζ_a, ζ_b, ζ_c and undamped natural frequency ω_{na}, ω_{nb}, ω_{nc}, and so forth. Then by the principle of linear superposition the servomechanism response is the sum of the responses attributed to the specific modes a, b, c, etc. Thus for each component of error response the duration of the transient is given qualitatively by reference to the types of solutions given in the body of the paper for simple quadratics, and the relative magnitudes of the transients can often be approximated from the observation that the higher the magnitude of the root the smaller the coefficient of the time solution involving that root. It should be remembered that the parameters of the several elements in a system are not always known to high degrees of accuracy, and that approximate or graphical methods (11, 12, 19, 20, 22) that give the roots to the characteristic equations[5] to a few significant figures are justifiable and timesaving.

Because of the difficulties mentioned the transient method of analysis of complicated systems becomes most valuable as an aid to the visualization of the performance of a system rather than an aid in its design. If a designer has carried out the analysis of the dynamic performance based upon transient studies he can correlate the information with the actual performance if he gives any physical system a transient disturbance by any one of several means readily available to him and then observes or measures the response.

The next section of the paper treats a method of analysis that is particularly useful in the design of complicated systems. This method is sometimes termed the transfer-loci method and is based upon the frequency-response characteristic of the servomechanism.

13 FREQUENCY RESPONSE OF A SERVOMECHANISM

The difficulties of analyzing or synthesizing a complex servomechanism or automatic-control system by the transient method of analysis early became apparent to several investigators and a search was made for a more powerful and less cumbersome method of analysis. During this search it was natural to examine the methods employed in electrical analysis since complex electrical circuits had been analyzed and synthesized effectively for many years. Circuit analysis has been effectively carried out for some time through the study of its frequency-response characteristics. It appeared that the same general approach might prove effective in servomechanism design. Nyquist had developed a powerful means of determining the stability of feedback amplifiers through a study of its frequency-response characteristics and the similarity between feedback amplifiers and servomechanisms had been recognized. In 1942 Harris (21) pointed out that genuine improvements in design technique might result from applying the frequency-response method of analysis to servomechanism design, and in 1943 Hall (13) presented a thorough treatment of servomechanism analysis and synthesis by frequency-response methods. As a result of these investigations the frequency-response approach to servomechanism analysis has been developed into a powerful tool. Most of the remainder of the present paper is adapted from the paper by Hall (13).

The term "frequency-response characteristic," when applied to a mechanism, refers to the relationship between the input and the output of that mechanism when the input is a sinusoidal function of time. In servomechanism terminology, if $\theta_i(t)$ is the input and $\theta_0(t)$ is the output of a servomechanism (see Fig. 2) then if

[5] For a summary of graphical and analytical methods of obtaining the roots to characteristic equations, see Appendix C of the book by Ed S. Smith (22).

$$\theta_i(t) = A \sin \omega t \quad [57]$$

and A is small it will always be true that

$$\theta_0(t) = B \sin (\omega t + \phi) \quad [58]$$

In these equations A is known as the amplitude of the input, B is known as the amplitude of the output, ω is the angular frequency of motion of $\theta_i(t)$ and $\theta_0(t)$ (equal to $2\pi f$ where f is in cycles per second), and ϕ is the relative phase angle between $\theta_i(t)$ and $\theta_0(t)$.

The amplitude ratio B/A, and the phase angle ϕ, when determined as functions of angular frequency ω, comprise the frequency-response characteristic of the servomechanism.

As just shown, the relationship between $\theta_0(t)$ and $\theta_i(t)$, when $\theta_i(t)$ is a sinusoidal function of time, is specified by a magnitude (B/A) and an angle (ϕ). These two quantities can be considered as the defining properties of a vector whose amplitude is (B/A) and whose phase is ϕ. Thus it is frequently stated that a vector relationship exists between $\theta_i(t)$ and $\theta_0(t)$, when $\theta_i(t)$ varies sinusoidally with time. This vector relationship is represented symbolically by $\frac{\theta_0}{\theta_i}(j\omega)$ in which $j(=\sqrt{-1})$, itself, is generally thought of as a vector and emphasizes the vector properties of the ratio. The vector ratio, $\frac{\theta_0}{\theta_i}(j\omega)$ is characterized by an amplitude

$\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$, and a phase, arc $\left[\frac{\theta_0}{\theta_i}(j\omega)\right]$

The preceding development is briefly summarized as follows: If in a servomechanism

$$\theta_i(t) = A \sin \omega t \quad [57]$$

then

$$\theta_0(t) = B \sin (\omega t + \phi) \quad [58]$$

By definition the frequency response is denoted by $\frac{\theta_0}{\theta_i}(j\omega)$. The amplitude response is given by

$$\left|\frac{\theta_0}{\theta_i}(j\omega)\right| = \frac{B}{A} \quad [59]$$

and the phase response is represented by

$$\text{arc}\left[\frac{\theta_0}{\theta_i}(j\omega)\right] = \phi \quad [60]$$

The amplitude and phase response curves of a typical servomechanism are illustrated in Fig. 12.

The curves in Fig. 12 may be obtained (*a*) by calculation, if the constants of an actual or proposed design are available, or (*b*) by measurement, if the servomechanism itself is available. The frequency-response characteristic is measured by moving the input sinusoidally at a fixed amplitude but at various frequencies. At each frequency the amplitude of the output and its phase relation to the input motion is measured. The ratio of the amplitudes, plotted for each frequency, yields the first of the curves in Fig. 12. The phase difference between the input and output plotted for various frequencies gives the second of the curves in Fig. 12. The curves in Fig. 12 are readily calculated if the constants of the system and the differential equation relating the output to the input are known. The calculation is effected by replacing the operator p by the frequency operator $j\omega$ (where $j = \sqrt{-1}$) and applying conventional vector arithmetic.

The frequency response of a servomechanism may be closely correlated with its transient response. Important natural frequencies in the transient response are indicated by peaks in the amplitude-response curve. The magnitudes of the peaks of the amplitude response are measures of the relative damping of the natural frequencies of the transient response. The frequency band over which the amplitude response has a substantially constant magnitude is a measure of the speed of response to transients, since a high natural frequency (and therefore a high speed of response) is linked with a high resonant frequency in the amplitude response. When, as just indicated, the frequency-response characteristic is correlated with the transient-response characteristic, the former becomes a powerful means of analysis. The correlation between the sinusoidal and transient characteristics is illustrated by comparing the transient and frequency response of a simple servomechanism, representable by a second-order differential equation. Such a servomechanism comprises a servomotor with a moment of inertia and damping and whose output torque is proportional to the error.

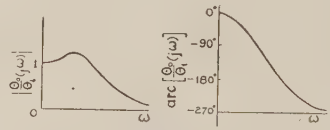


Fig. 12 Amplitude- and Phase-Response Curves of a Typical Servomechanism

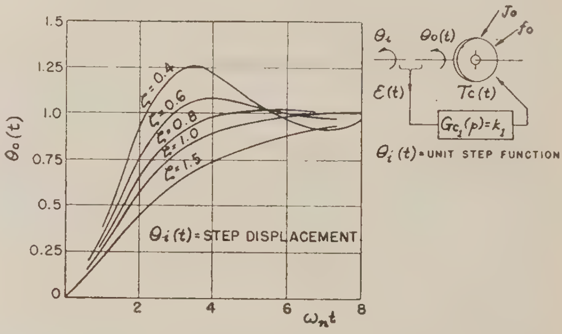


Fig. 13 Transient Responses of a Second-Order Servomechanism

If the input $\theta_i(t)$ is a step displacement and the output $\theta_0(t)$ is determined for various values of the damping ratio, the set of transient responses illustrated in Fig. 13 results. If $\theta_i(t)$ is made a sinusoidal function

$$\theta_i(t) = A \sin \omega t$$

and the amplitude response of the output $\theta_0(t)$ is determined, the set of curves in Figs. 14 and 15 is obtained. Comparison of the transient and frequency responses reveals a number of points of correspondence: (*a*) The frequency at which the transient response oscillates (the natural frequency) is approximately the same as the frequency at which the amplitude response has a peak (the resonant frequency); (*b*) as the damping ratio is reduced the transient response becomes more oscillatory, and the peak in the amplitude response is magnified; (*c*) when the damping ratio is made larger than unity the transient response becomes sluggish and the amplitude response falls off rapidly without a peak.

It is frequently convenient to combine the information contained in the amplitude-response curve and in the phase-response curve of a system by a single graph. This can be accomplished by remembering that the amplitude ratio and the phase angle are

quantities defining a vector which relates the output and input. As the frequency of the input is varied this vector varies in phase and magnitude and the amplitude- and phase-response curves present the information on the manner in which these two properties of the vector change with frequency. The same information can be provided in an alternative way by plotting on polar co-ordinate paper the path followed by the tip of the vector as the frequency varies over the band of interest. If various points of the curve are labeled with the frequency to which they correspond one curve will supply the information contained in

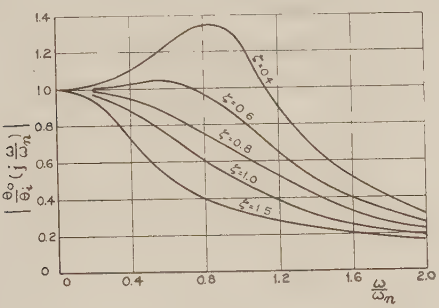


Fig. 14 Amplitude-Response Curves of Second-Order Servomechanism

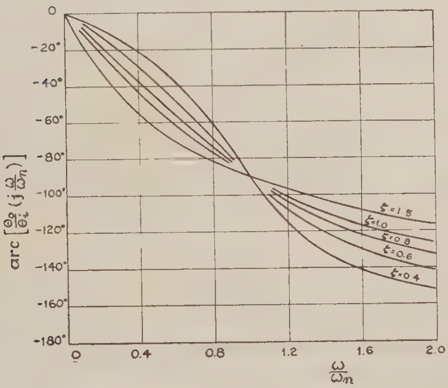


Fig. 15 Phase-Response Curves of Second-Order System

two curves when the amplitude and phase response are plotted separately. This graphical presentation is known as the locus of the frequency-response characteristic. It is illustrated in Fig. 16 which was drawn for the same second-order system to which the curves in Figs. 14 and 15 apply. The principal advantage of presenting information in this fashion lies in the means it provides for visualizing the frequency response and in the fact that it emphasizes the very important relationship that always exists between the amplitude and phase responses of a system.

14 Servomechanism Transfer Function

It is clear from the block diagram in Fig. 11 that a single function completely defines the performance of a servomechanism in which the feedback link contains no frequency-dependent elements. The defining function is the relationship between the servo output θ_0 and the error ϵ. If this relation is known in operational, sinusoidal, or time-response forms, the performance of the system is completely defined for all conditions that may be imposed upon it. This function, relating the servomechanism output to its error, has been termed the "transfer function" of the

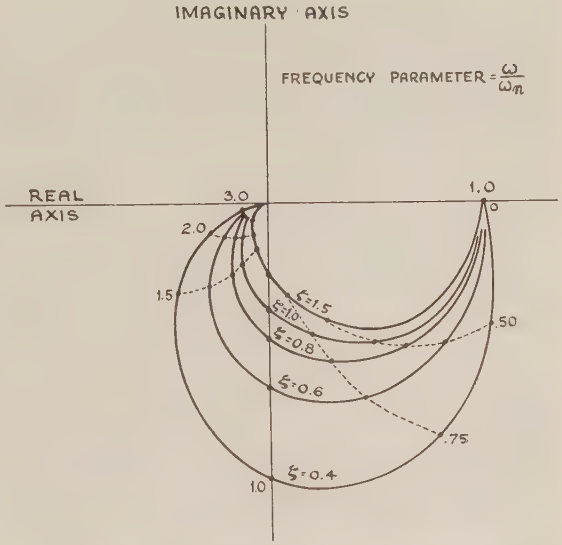


Fig. 16 Locus of Frequency-Response Characteristic of Second-Order System

servomechanism. When the sinusoidal or frequency-response form of the transfer function is studied it becomes a powerful analysis and synthesis tool. The transfer function may be derived from a known frequency-response characteristic of the servosystem, it may be measured directly, or if system constants are known, it may be calculated directly.

In terms of the nomenclature previously established in this paper the operational form of the transfer function is written $\frac{\theta_0}{\epsilon}(p)$. The sinusoidal form of the transfer function is obtained by replacing the operator p by $j\omega$, giving $\frac{\theta_0}{\epsilon}(j\omega)$. This function is a vector quantity with an amplitude and a phase characteristic just as the ratio $\frac{\theta_0}{\theta_i}(j\omega)$ is a vector quantity.

The transfer function of a servo system is always the product of two parts, one that is invariant with frequency and a second that is frequency-dependent. The fact that these two components exist is emphasized by writing the transfer function in the following form

$$\frac{\theta_0}{\epsilon}(j\omega) = KG(j\omega) \qquad [61]$$

The term K represents that part of the transfer function which is invariant with frequency. This portion is known as the gain or the sensitivity factor and is a function of amplifier gain, gear ratios, etc. The second part of the transfer function is denoted by $G(j\omega)$ and represents the portion of the transfer function which changes with frequency.

The frequency response of the servomechanism is connected with the transfer function by the following vector equation

$$\frac{\theta_0}{\theta_i}(j\omega) = \frac{KG(j\omega)}{1 + KG(j\omega)} \qquad [62]$$

The transfer function of a system is calculated by straightforward circuit-analysis techniques, or by determining the differential equation relating the output to the error and replacing $\frac{d}{dt}$ by $j\omega$.

15 The Transfer-Function Locus

The transfer function can be studied by means of its frequency characteristics just as other functions have been so studied. The amplitude- and phase-response curves of the transfer function can be drawn and these curves completely define the characteristics of the transfer function of the servomechanism and therefore completely define the system itself. Just as the phase- and amplitude-response curves were combined into a single polar plot with frequency as a parameter, so can the frequency- and phase-response curves of the transfer function be combined. This parametric polar plot of the transfer function has been called the "transfer-function locus," or simply the transfer locus of the servomechanism. The transfer locus completely defines the characteristics of the servo system. A study of its nature provides an effective method for the synthesis of servomechanisms intended for particular applications and a useful general guide to the adjustment of servomechanism parameters in order to secure optimum performance.

The reason for the effectiveness of employing the transfer function $KG(j\omega)$ as a means of analysis is explained as follows:

If it is desired to analyze the response of a servo and design compensation circuits for improving its performance, the function $\frac{\theta_0}{\theta_i}(j\omega)$ is awkward to work with directly. The reason for this arises from the complex nature of the relation between $\frac{\theta_0}{\theta_i}(j\omega)$ and the system parameters. On the other hand, it has been shown that the performance of the servo is completely determined once the transfer function $KG(j\omega)$ is known. A much simpler relation generally exists between the system parameters and the transfer function so that it is comparatively easy to synthesize a controller once the required form of the transfer function is known. The system design is applied to choosing, altering, and improving the servomotor or servo controller, the characteristics of which directly affect the transfer function $KG(j\omega)$.

Design criteria expressed in terms of restrictions upon the transfer function are therefore most easily translated into physical design. However, the final decision as to the quality of the performance of a particular servo is made from a knowledge of the character of its output function, and it is necessary therefore to translate restrictions upon the output and function into restrictions upon the transfer function. The situation is summarized as follows:

It is necessary to know the phase and magnitude of $\frac{\theta_0}{\theta_i}(j\omega)$ in order to decide if the servo is satisfactory. In synthesizing a servo it is easier to work with the transfer function $KG(j\omega)$. By correlating the transfer-function characteristics with the properties of the system frequency response, an effective design procedure can be developed. The correlation of these two functions is most easily effected graphically, as described in the following:

If a parametric frequency plot of the transfer function $KG(j\omega)$ is available, the magnitude and phase of the function $\frac{\theta_0}{\theta_i}(j\omega)$ may be found by graphical calculation. The function $\frac{\theta_0}{\theta_i}(j\omega)$ has been related to the transfer function by Equation [62].

A plot of a typical transfer function $KG(j\omega)$ is illustrated in Fig. 17. Suppose it is desired to calculate $\frac{\theta_0}{\theta_i}(j\omega_c)$ for a particular frequency ω_c, with only the transfer locus available. The transfer function at ω_c, $KG(j\omega_c)$, is represented by the vector $\overline{oc}$ while the vector $\overline{ac}$ represents the term $[1 + KG(j\omega_c)]$, since the point a is located at $(-1 + j0)$. Therefore the magnitude of $\frac{\theta_0}{\theta_i}(j\omega_c)$ is given by

$$\left|\frac{\theta_0}{\theta_i}(j\omega_c)\right| = \frac{|KG(j\omega_c)|}{|1 + KG(j\omega_c)|} = \frac{|\overline{oc}|}{|\overline{ac}|} \qquad [63]$$

while the phase of $\frac{\theta_0}{\theta_i}(j\omega)$ is given by

$$\text{arc}\left[\frac{\theta_0}{\theta_i}(j\omega_c)\right] = \text{arc}\,(oca) \qquad [64]$$

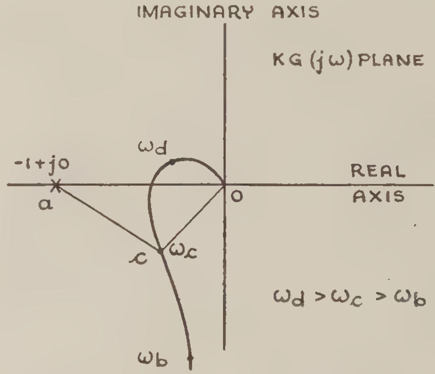


Fig. 17 Graphical Determination of $\frac{\theta_0}{\theta_i}(j\omega)$ From Transfer Locus

The angle, arc (oca), is negative. Thus the magnitude of $\frac{\theta_0}{\theta_i}(j\omega_c)$ is equal to the ratio of the magnitudes of the vectors $\overline{oc}$ and $\overline{ac}$ and the phase of $\frac{\theta_0}{\theta_i}(j\omega_c)$ is equal to the angle between these two vectors.

Equations [62] and [63] permit ready visualization or calculation of the magnitude and phase of $\frac{\theta_0}{\theta_i}(j\omega)$. At small frequencies such as ω_b (see Fig. 17), both vectors $\overline{oc}$ and $\overline{ac}$ are large and approximately equal, and their ratio is approximately unity. The angle between the two vectors, the phase of $\frac{\theta_0}{\theta_i}(j\omega)$, is small at this frequency. As the frequency increases the angle between the two vectors increases and their lengths become smaller so that differences in their lengths cause the ratio $\frac{\overline{oc}}{\overline{ac}}$ to depart from unity. Whether the ratio $\frac{\overline{oc}}{\overline{ac}}$ (the magnitude of $\frac{\theta_0}{\theta_i}[j\omega]$) increases or decreases as the frequency increases depends upon the shape of the curve relative to the origin and the point $(-1 + j0)$. A continuation of this reasoning for the remainder of the frequency range permits the general shape of the phase and magnitude of the servo output to be completely determined. If desirable, the phase and magnitude curves can be determined from the transfer locus with accuracy and ease by using a protractor and divider and measuring the angles and lengths directly from the graph.

Correlation between other servomechanism characteristics and the shape and form of the transfer locus are described in the following sections.

16 Absolute Stability Criterion

The primary requirement that almost every servomechanism

must satisfy is that of stability. Although a servo system must be more than barely stable to be satisfactory, a stability criterion of one type or another is generally the first test applied to proposed servomechanism design. Several criteria exist; however, the one described here was developed primarily for application to feedback amplifiers. It is at once apparent to those familiar with feedback-amplifier theory that the transfer locus of a servomechanism is analogous to the Nyquist diagram of a feedback amplifier. The term Nyquist diagram has been given to this type of plot for a feedback amplifier because of a very useful criterion developed by Nyquist (8) for determining the stability of a feedback amplifier. This criterion may be applied equally well to the transfer locus in order to determine from its shape and position whether or not the servomechanism for which it is drawn is stable. To apply the Nyquist stability criterion to servomechanisms, the following procedure is employed: (*a*) The transfer locus [$KG(j\omega)$ plotted in polar form] is drawn for all frequencies from zero to infinity; (*b*) the conjugate of the transfer locus is drawn. The conjugate of a curve is the mirror image of the original curve about the real axis; (*c*) if the curves so formed enclose the point $(-1 + j0)$, the system is unstable. If the curves do not enclose this point, the system is stable. The application of this criterion is illustrated in Fig. 18.

The foregoing criterion applies to curves of closed form; that is, it applies to transfer loci of such character that the loci and their conjugates join at zero and at infinite frequency. Actually the transfer loci of most servomechanisms are of the open form and some extension is required in order to apply the stability criteria to these forms of transfer loci. The open form of the transfer locus can be changed into the closed form by connecting the curve and its conjugate at the zero-frequency point by means of a circle of infinite radius. The connection should always be made in such a way that no phase discontinuity occurs along the path of the curve.

This is illustrated in Fig. 19 in which are plotted the transfer loci of two common types of servomechanisms, both of which are stable.

17 Transfer Loci for Various Types of Steady-State Performance

The performance of a servomechanism under steady-state conditions is always of great importance. If the servomechanism is primarily a positional device it is desirable that the servomechanism take up various positions without requiring an error to maintain it in that position. Similarly it is frequently necessary for the servomechanism to follow an input of constant velocity, that is, one in which $\theta_i(t) = kt$. In this case it is desirable for the servomechanism to follow various velocities as required without the necessity of a system error to maintain that velocity. Servomechanisms that satisfy the first condition frequently are termed zero-displacement-error servomechanisms and servos that meet the second condition similarly are called zero-velocity-error servomechanisms. It can be readily shown that if the transfer locus of a servomechanism approaches infinity along the negative imaginary axis, the servo will have zero-displacement error. Similarly if the transfer locus of a servomechanism approaches infinity along the negative real axis that servo will have zero-velocity error. The zero-velocity-error servo will, of course, have zero-displacement error also. In Fig. 19 is illustrated the transfer loci of a zero-velocity-error system and of a zero-displacement–error system.

This concept can be extended to servo systems that will follow a constant input acceleration without steady-state error, and so forth.

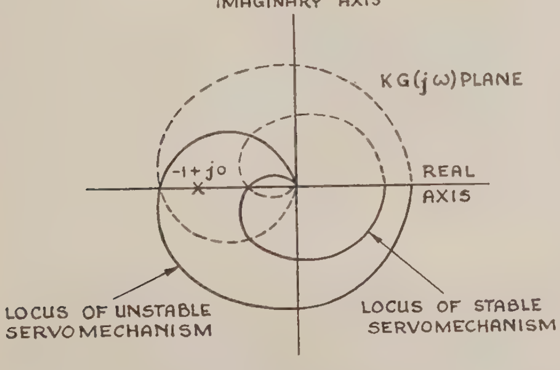


Fig. 18 Application of Nyquist Stability Criterion to Servomechanisms With Positional Error

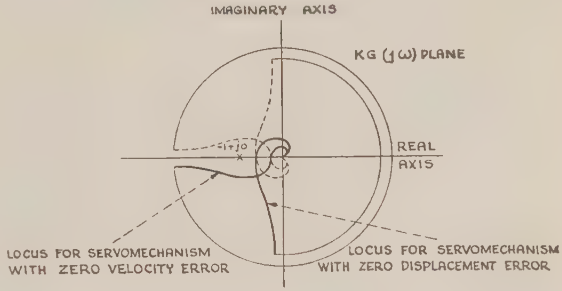


Fig. 19 Extension of Nyquist Stability Criterion to Zero-Displacement-Error and Zero-Velocity-Error Servomechanisms

18 Determination of the Gain Factor (K)

The particular servo parameter most easily adjusted is, perhaps, the gain factor K, which is the frequency invariant portion of the transfer function, $KG(j\omega)$. If the servocontroller is an electronic amplifier, the gain may be fixed, in general, by adjusting a voltage divider controlling the gain of the amplifier. The gain factor of systems incorporating controllers other than electronic amplifiers may be less easily adjusted, but the procedure in most cases is still relatively simple. Since the gain is easily adjusted and since its setting is so important in determining the characteristics of a servomechanism, it is important to develop a technique of gain adjustment that yields optimum results in servo performance.

The effects of increasing the gain or sensitivity of the servomechanism are (*a*) the velocity error of the system is reduced if the servomechanism is of the zero-displacement-error type; (*b*) errors caused by restraining torques on the servo output are reduced; (*c*) errors caused by mechanical misalignment in the servomotor or controller are reduced; (*d*) the imaginary components of the complex roots of the system (the natural frequency or frequencies of the system) are increased in most cases; (*e*) the magnitudes of the real roots of the system are increased in most cases; and (*f*) the real parts of the complex roots of the system (the damping constants) in general are decreased. The first three of these effects reduce the system steady-state error; the next two increase the speed of response of the system, and the last reduces the speed of response of the system in that the time required for a transient oscillation to damp out is increased. A general rule that can be deduced from the factors given is that

the servomechanism sensitivity should be as high as is compatible with proper servo stability.

Any physical closed-cycle system can be made unstable if the sensitivity is sufficiently increased. This is evident from the transfer locus of a physical servomechanism since such a locus always crosses the real axis and therefore as the sensitivity or gain factor K is increased, the locus will enclose the $(-1 + jO)$ point at some finite sensitivity and the servo will become unstable. The gain factor should always be adjusted to avoid instability.

Most applications, however, require not only that the servo be stable but that its degree of stability be satisfactory. The degree of stability determines the damping of the oscillatory components of the servo transient response and in general the gain factor should be so selected that these response components are well damped. When the oscillatory components are insufficiently damped the transient response requires an excessive time to reach its final value and exhibits excessive overshoot. In addition, the amplitude response possesses excessively high peaks. The effect on the transient response is illustrated in Fig. 13 which shows that as the damping ratio is decreased the oscillatory nature of the system is more pronounced. The effect on the amplitude response is indicated in Fig. 14 which shows that as the damping ratio is decreased the magnitude of the peak in the response is augmented.

When a servomechanism is designed on the transient basis the degree of stability is controlled by requiring the effective damping ratio to be of the order of 0.5. When a servomechanism is designed on the frequency-response basis it has been found that an equally satisfactory criterion is to limit the magnitude of peaks in the amplitude response to approximately $1^1/_3$. Neither requirement is based upon exact mathematical criteria; both have been found to give good results in almost every application.

The determination of the gain factor K, that will limit the maximum value of the amplitude response to a prescribed value and thus provide adequate damping in the system, is easily carried out graphically through the use of the transfer locus of the system. If the ratio $\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$ is denoted as M for a particular frequency, curves of constant M can be drawn in the $KG(j\omega)$ plane. These curves turn out to be circles whose centers and radii are easily determined. A set of these circles is illustrated in Fig. 20 together with the transfer locus of a typical servomechanism. The significance of these curves lies in the fact that wherever the transfer locus crosses one of these circles it indicates that at the frequency for which the intersection occurs the value of $\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$ is equal to the M corresponding to the intersected circle. Thus in Fig. 20 $\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$ for the system whose transfer locus is denoted by $K - 1$ will have a value of 1.1 at 1 cycle per sec, 1.5 at 2 cycles per sec, 0.75 at 3 cycles per sec, etc. A value of M can generally be found so that the circle corresponding to this M value is tangent to the transfer locus at some point. A point of tangency indicates that the function $\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$ has a maximum or a minimum at that point. Thus in Fig. 20 the transfer locus drawn for $K = 1$ is tangent to the $M = 1.5$ circle, indicating that the output function has a maximum of 1.5 at 2 cycles per sec.

Also drawn in Fig. 20 are the transfer loci for the same servomechanism with different values of gain factors. It is evident that the maximum value of the output function increases as the gain factor is increased. The proper degree of stability is determined by sc choosing the gain factor that the transfer locus corresponding to this gain factor is tangent to the circle drawn for the predetermined maximum value of M.

19 Basic-Design Procedure

Since the transfer-function approach is perhaps most useful in the design and synthesis of servomechanisms, the principal steps in the design procedure employing the frequency-response approach are summarized as follows:

(a) Determine the transfer function $KG(j\omega)$ for the first approximation to the complete servomechanism. This requires the determination of the amplitude and phase characteristics of $KG(j\omega)$.

(b) Plot this transfer function in the form of a transfer locus using the specific values determined from (a).

(c) Decide on the degree of stability required in terms of the M ratio discussed in section 18 and determine the gain factor that will produce tangency between the transfer locus and the circle drawn for the maximum permissible value of M. The gain factor can then be used to determine actual sensitivities in the servosystem. Determine the frequency at which the maximum value of $\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$ occurs.

(d) From a knowledge of the permissible sensitivity, the frequency at which the function $\left|\frac{\theta_0}{\theta_i}(j\omega)\right|$ is maximum, and the zero-frequency behavior of the transfer locus, answers can be obtained to the following questions:

1 Is the servomechanism of the type (zero-displacement-error servo, zero-velocity-error servo) suitable for the particular application?

2 Can the gain factor be adjusted to secure adequate damping?

3 Is the sensitivity permitted by adequate damping sufficiently high to minimize the effect of nonlinear factors such as sticky valves, dry friction on the output member, etc., and linear factors such as velocity and acceleration error, etc.?

4 Is the natural or resonant frequency sufficiently high to provide the speed of response required by the application?

(e) If the answer to any of the foregoing questions is negative remedial action must be taken. Readjustment of certain of the parameters or minor changes in design may be sufficient. Otherwise major design changes must be made or corrective devices incorporated in the system. These corrective devices can be designed: (a) To improve the transient response; (b) increase the sensitivity permitted by adequate stability; or (c) change the basic type (zero-displacement error, zero-velocity error) of the servo. If the corrective element is introduced as an element cascaded with the controller in such a way that no mutual reaction with other elements occurs, the new transfer function is the product of the old transfer function and the transfer function of the corrective element. The new transfer locus can be formed graphically by multiplying amplitudes and adding phases for a particular frequency. If the corrective element is inserted in the system in a different manner straightforward circuit-analysis techniques can be applied to the determination of the transfer function.

20 Summary

Two approaches to the mathematical analysis of servomechanisms have been pointed out in this paper. It has been demonstrated that, in general, a broad understanding of servomechanisms could be obtained by an understanding of both approaches, and in particular, that valuable performance and design data could be obtained. It has been shown that the transient approach

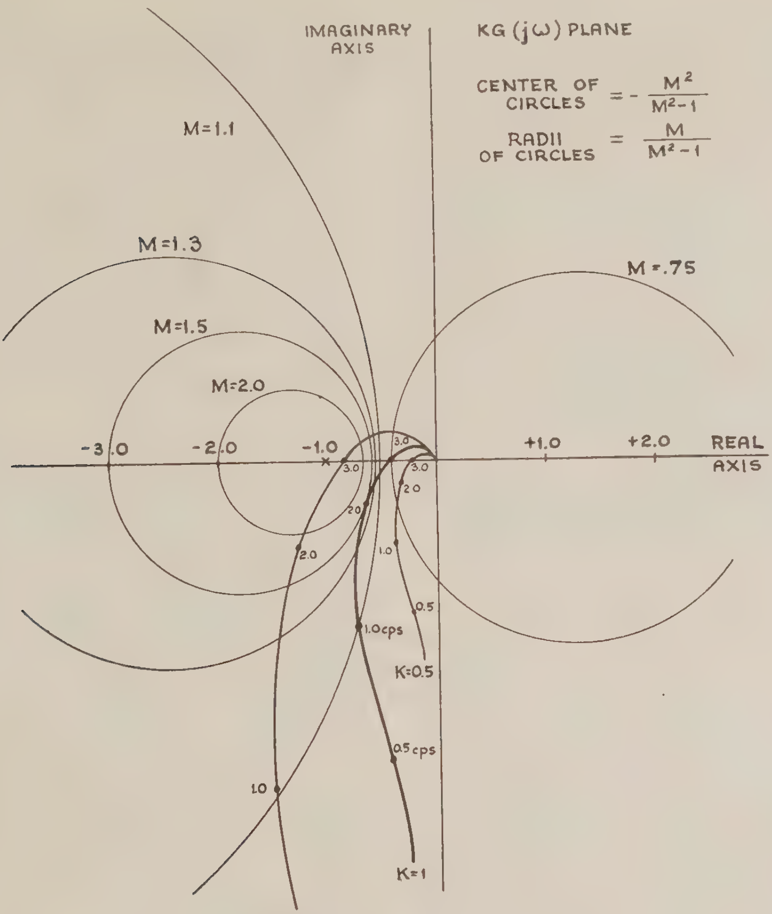


FIG. 20 GRAPHICAL DETERMINATION OF GAIN FACTOR OF TRANSFER LOCUS

is particularly valuable when the exact response to known inputs is required, and in the analysis of simpler servo systems. It has been further shown that the frequency-response approach is indispensable when the analysis and synthesis of complex servo systems is required.

ACKNOWLEDGMENT

The authors gratefully acknowledge the co-operation and assistance of their colleagues at the Massachusetts Institute of Technology. Particular thanks are due to Mr. H. T. Marcy and Mr. G. J. Schwartz for their many valuable suggestions, and to Mr. D. P. Campbell for his contributions to the transient-analysis approach.

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Discussion

C. Concordia.[6] The writer is greatly interested in the steady-state method of analysis of control systems described in this paper and would like to comment on what he considers its scope in relation to other aspects of the control problem.

If one starts from scratch and constructs a control system for a particular job, one's first task is to make sure that the system is stable. For this purpose it seems best to find regions of stable operation in terms of those parameters which are within our control by the application of Routh's stability criteria to the characteristic determinant of the system. Such a study gives information over a great range of system parameters very quickly, without the necessity of studying any one system in detail.

Following such a study, one becomes concerned with the response of the system. The most direct method might appear to be to compute directly the transient response to the various disturbances which are of interest. However, for the reasons discussed by the authors, this sort of calculation does not tell immediately what to do in order to improve performance. Thus it is convenient to devise on a semiempirical basis criteria of probable transient performance in terms of steady-state performance and to make steady-state studies of the system as they have described.

However, even after this is done, one may still wish to make sure that performance will be satisfactory by a direct study of the transient performance. This is especially true in those cases where the system may be subjected to a variety of different kinds of disturbances. An example of this is an airplane-engine turbo-supercharger regulator, which may be subjected to disturbances of the pressure-boost setting, the engine-speed setting, or the throttle setting. It will be apparent to those familiar with control-system performance that the optimum stabilizer design for one type of disturbance will not be the same as that for the others, so that the over-all optimum must be a matter of engineering judgment rather than of mathematical criteria.

For the purpose of making such studies of transient performance we have found it convenient to use the differential analyzer. Of course, once a problem is set up on the differential analyzer it is a simple matter to change parameters and run off the resulting transient-response curve, and in fact in many cases this has been so attractive that we have shortcircuited completely the intermediate steady-state analysis method.

A further reason for this is that in many cases, as the refinement of design proceeds, one becomes primarily interested in studying more and more the effects of those nonlinear elements such as friction, backlash, saturation, etc., that are left out of the linear analysis which has been described. A further related factor is that a linear analysis becomes more nearly valid as the control means become better. In many cases very limited control means may be available, and these may be essentially off-on or some other kind of contact control. In these cases it is almost necessary to determine the transient response directly if a good idea of the quality of actual performance is to be had.

[6] Engineer, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

L. A. MacColl.[7] It is shown in the paper that the function $KG\ (i\omega)$ plays a dominant role in the theory of the particular servomechanism which is under consideration. Perhaps it will be well to point out explicitly that this function, suitably generalized, is fundamental in the theory of any single-loop servomechanism, and to indicate how the generalized function can be determined experimentally.

Let us consider a general single-loop servomechanism, in which we have a "forward circuit," consisting of certain components $C_1, C_2, \ldots, C_l$ connected in tandem, and a "feedback circuit," consisting of certain components $C_{l+1}, C_{l+2}, \ldots, C_{l+m}$ connected in tandem.

The typical component C_n is characterized by a complex-valued function $Y_n\ (i\omega)$, having the following significance: In the steady state, in which all of the signals in the system are varying sinusoidally with time with the radian frequency ω, the absolute value of $Y_n\ (i\omega)$ is the ratio of the amplitude of the signal at the output of C_n to the amplitude of the signal at the input of C_n, and the angle of $Y_n\ (i\omega)$ is the phase difference between these two signals.

Now it is easily shown, by reasoning like that employed in the paper, that the function

$$Y(i\omega) = Y_1(i\omega)Y_2(i\omega)\ldots Y_l(i\omega)\ Y_{l+1}(i\omega)Y_{l+2}(i\omega)\ldots Y_{l+m}(i\omega)$$

plays the role in the theory of the present system that the function $KG\ (i\omega)$ plays in the theory of the system discussed in the paper.

Let us consider the physical significance of $Y\ (i\omega)$. Suppose that we open the feedback loop between any two consecutive components, say the components C_n and C_{n+1}, adding the proper terminations, if necessary, so that the impedance relations within the system will not be disturbed. Then let us impose a known sinusoidal signal at the input of C_{n+1}, and let us measure the amplitude and phase of the resulting signal at the output of C_n. It is readily seen that the ratio of the amplitude of the signal at the output of C_n to the amplitude of the signal at the input of C_{n+1} is the absolute value of $Y\ (i\omega)$, and that the phase difference of the two signals is the angle of $Y\ (i\omega)$. On this account $Y\ (i\omega)$ is commonly called the "loop transmission ratio of the servomechanism."

We note that where we open the feedback loop, whether between C_1 and C_2, or between C_2 and C_3, or elsewhere, is indifferent in principle. However, as far as experimental technique is concerned, it may not be practically indifferent. In many cases the physical nature of the signals changes from point to point in a servomechanism. At one point the signal may be the angular position of a shaft, at another point it may be an electromotive force, and at a third point it may be the pressure of a fluid. In

[7] Bell Telephone Laboratories, Inc., Murray Hill, N. J.

such a case it may well be that we can perform the measurements just described more accurately and conveniently at one point in the loop than at another. We should, of course, select the point at which we open the loop, and perform the measurements accordingly.

Referring to the significance and use of the Nyquist diagram: Suppose that we have a servomechanism with a Nyquist diagram which encloses the critical point (—1, 0), so that the system is unstable, or which, while it does not actually enclose the critical point, passes near to that point, so that the system is barely stable. In either case, we have to modify the system in some way, in order to make its performance satisfactory.

One rather obvious thing we may do is simply to vary the value of some one constant parameter, seeking a particular value which gives a satisfactory performance. This procedure, which is discussed at length in the paper, may be entirely satisfactory in many cases. However, it is evident that this procedure is quite restricted, and it is to be expected that we can often achieve more satisfactory results in other ways. Here the Nyquist diagram performs a great service; for it reveals a great variety of means for obtaining a satisfactory system, many of which are not suggested at all by the older methods.

In so far as satisfactory stability of a servomechanism is concerned, all that is necessary is that the curve of the Nyquist diagram shall neither enclose the critical point (—1, 0) nor pass too near to that point. In order to achieve a curve which is satisfactory in this sense, we can confine ourselves to varying element values, if that is sufficient, or we can insert additional frequency-selective elements in the system, so as to alter the very form of the curve in a drastic manner. The possibilities of the latter sort are particularly valuable when we are dealing with complicated systems subject to stringent requirements.

Ed S. Smith.[8] The authors deserve the thanks of mechanical engineers for the release of this material as soon as the wartime restrictions lifted. It is high time for this M.I.T. group of electrical engineers to claim due credit for its excellent work and real contributions to the war effort. The writer is glad to acknowledge here his professional and personal indebtedness to Bush, Hazen, Gardner, Draper, Brown, Harris, Hall, Campbell, Kochenberger, Taplin, Philbrick, and Ahrendt, to name a representative number of its more active contributors to the art. At the same time, it must be remembered that the pioneering in servomechanisms has been shared by another group of electrical engineers at the Bell Telephone Laboratories: Nyquist, Bode, Black, Ferrell, Dietzold, MacColl, and Barnes whose work with Gardner at M.I.T. is but one tie of many between these two groups.

As a mechanical engineer, the writer's primary interest is to see how these electrical-engineering techniques may be best made available to other engineers, for use in control generally. For this "transmission problem" it is good that such an experienced and co-operative teacher as Brown heads the A.I.E.E. activity in this field. For the sake of continuity, it is hoped by the writer, as the current chairman of the A.S.M.E. Instruments and Regulators Division, that such activity will be most intimately tied in with the work of this division which sponsored this paper. This division in the past has followed control theory as exemplified by papers by Ivanoff, Minorsky, and division members including Fairchild, Grebe, Mason, Sperry, and the present writer to mention a token few. The following comments are purely individual:

It might offhand seem that servomechanisms should form ideal examples of control for mechanical engineers since the familiar elements are the inertia of a mass, the damping of a dashpot, and the proportional effect of a spring. However, from the standpoint of the process-control engineer, these happen to overstress inertial and oscillatory effects. Also mechanical analogs are a bit more farfetched than the hydraulic which has the advantage of more closely corresponding with the electrical systems in which these electrical engineers really think.

Ideally, each actual system should be directly followed in setting up the mathematical formulation which would then be used in a pure form in obtaining results. But this procedure would deprive the creative engineer of the stimulus derived from working in the medium of his experience. Since so much of the literature will be on electrical-engineering techniques, it appears that each active control engineer should become adept in their use.

The experts in the servomechanism and control fields are unfortunately not exceptions to the rule that the experts in an art always ball up the terminology and notation until they can be followed only by a like expert. To a control engineer, if a regulated temperature is too high, the error would be positive and taken as $(T_{\text{actual}} - T_{\text{set}})$. To a "servomechanist," it would seem that the temperature setting T_i was not being followed by the actual temperature T_0 and that the error would be $(T_i - T_0)$. For another example; probably the most important factor in control is the variable "load" which often affects a most inconvenient part of the plant with effects echoing around and around the system, whereas too often the servomechanist either entirely neglects the load change or shrugs it off as mere "hash" or "noise," a mere random impulse without the dignity of identity.

Again, the "controlleer" is charged with habitually over-simplifying analyses, using an occasional integral or appropriate derivative in visualizing the ills and cures of regulated systems reference (16), whereas the servomechanist seems to tend to stick to the more readily available means for obtaining a typical $(1 + Tp)$ term without, in so far as the present writer can see now that the veil of wartime secrecy has been pierced, creating any essentially new form of regulator. At least the controlleer can directly express a ratio of effect to its cause and the lag without becoming involved in the width of a hypothetical frequency-transmission band and in such an acoustical abstraction as "decibels per octave."[9] The point appears to be that the tools of the different arts have different advantages to those who are familiar with their use.

Finally, on this point, the controlleer's need is for an approach which will enable him most flexibly to solve a wide variety of control problems, whereas to the servomechanist, the temptation appears to be irresistibly strong to go overboard on overspecialized treatments, e.g., the nondimensional approach, for the sake of greater efficiency in his special field at the expense of widest utility, with the result that anyone not working steadily in that field will usually save considerable time and be more certain of the correctness of his results if he starts from scratch and sticks to the closest relation between the physical phenomena and the mathematics.

A noteworthy exception to this is that grand tool, the Nyquist method of testing stability. This is essentially so simple that it is difficult for even experts to confuse although they bring in the "conjugate complex," and the idea of "negative frequencies." It may be well here to restate the Nyquist criterion following Harris' rule: A regulated system is unstable only if, 1 the sensitivity (effect/cause) exceeds unity at a lag of 180 deg, and 2, the lag there increases with increasing frequency at the least value of sensitivity which is greater than unity. Of course in

[8] Eclipse-Pioneer Division, Bendix Aviation Corporation, Teterboro, N. J. Mem. A.S.M.E.

[9] "The Servo Problem as a Transmission Problem," by E. B. Ferrell, Proceedings of the Institute of Radio Engineers, vol. 33, November, 1945, pp. 763–766.

any actual system the sensitivity finally must decrease to zero with increasing frequency.

It may lend confidence in this method to note that this was independently approached from purely practical considerations. The writer in 1936 remarked[10] that the time lags in a regulated system were best observed by taking effects in the same direction as their causes, which amounts to dropping the 180 deg for the control relation, as is now conventional. Ivanoff in 1938 (U. S. patent 2,268,285) showed familiarity with the requirement that the lag be 180 deg for steady hunting. And Fairchild in 1940 (Temperature Symposium) noted that the further requirement for steady hunting was that the over-all sensitivity be unity. (Incidentally, as Le Corbeiller noted, to tend to sustain oscillations reliably at substantially a definite amplitude, there must be such a nonlinearity that, upon a decrease of amplitude, the over-all sensitivity or "gain" increases. All other material in this discussion is limited to linear systems.)

In evaluating the advances claimed in the paper: Apparently Harris (21) advanced the Nyquist technique mainly by using the grids of the paper's Fig. 20 to show the sensitivity, with maximum and minimum values of (θ_0/θ_i) where the circles are tangent to the Nyquist loci of the vectors. The advance of Hall (13) is mainly in the systematic correlation of the circuits and networks with their control relationships; and Taplin's pioneering use of $(1 + gh)$ in the denominator, while limited to very simple systems such as elementary servomechanisms without multiple feedback loops, still had wide implications and represents a logical operational advance. The foregoing is an offhand effort to bring the work of these contemporaries into better focus.

It is unquestionably valuable to "block up" a system for the purpose of setting up the equations as a standard procedure, and to use a-c network techniques in obtaining the over-all sensitivity, gain, or transfer ratio, expressed by the appropriate *gh*. . . relations. This is useful both as a test for stability and as an indication of the transient performance of the system under such standard forcing functions as the impulse, step, ramp, etc., by the use of the frequency-analysis techniques, notably the Fourier, as recently advocated by Harris. In other words, the formal transient analysis may be entirely replaced by the frequency approach.

Since the paper refers to several publications of the authors' group, it may be a service to engineers attempting to use such publications to note the following: In reference (1), Draper and Bentley, in Figs. 3 and 6, 1.58 should be 1.74. In reference (10), Weiss, the cubic chart III*a* is incorrect in part, a correction having appeared only in a classified paper by one of the authors; and in reference (11), Liu, an approximation-producing assumption appears to be unstated therein which affects the accuracy of this approach in certain ranges which are still useful in control. It is hoped that any like points which later turn up in present and future papers in this field will be scrupulously cleared by publication.

Nothing of the foregoing discussion is to be construed as minimizing the value of the papers by this group. Instead, the writer has noted that the "servomechanism approach" is also of value[11] in control generally, paradoxically including self-operating regulators. The writer is happy to express his indebtedness to members of this group and their works.

It is the main purpose of the writer, as an occasional worker on control, objectively to direct the attention of other mechanical engineers to useful control tools wherever found.

[10] "Automatic Regulators, Their Theory and Application," by Ed S. Smith, Jr., Trans. A.S.M.E., vol. 58, 1936, pp. 291–303.

[11] "Stabilizing a Suction-Relief Valve," by Ed S. Smith, Trans. A.S.M.E., vol. 67, 1945, p. 87.

Authors' Closure

The authors appreciate the comments made by Messrs. Concordia, MacColl, and Smith, particularly for their efforts to supplement the work given in the paper.

Mr. Concordia's comments on the usefulness of a Differential Analyzer to aid in the design are very pertinent though such devices are not widely available. However, as the work on similators or transient analyzers[12] for dynamical systems progresses and more of them become available their use may get widespread adoption. The one difficulty still remaining will be that of determining the parameters of the elements of the system or the various functional relationships that define the system elements. Then, of course, when these data are known a great deal can be done by steady-state sinusoidal analyses by persons not having the use of a Differential Analyzer. By skillful direction of experimental techniques during the determination of much of the basic data on the components, the work of applying the steady-state methods of analysis is considerably reduced.

Dr. MacColl's comments are very pertinent, particularly when one realizes the wide range of physical quantities encountered in any single control system and the relative ease with which certain quantities can be measured in practice and the relative status of the art of measurement of other quantities. For example, at one end of the scale one observes the simplicity and accuracy with which time-varying electrical quantities can be measured compared with the difficulties at the other end of the scale involved in measuring time-varying quantities such as acceleration, pressure, strain, fluid flow and the like.

Dr. MacColl's comments on the need to change the shape of the locus on the Nyquist diagram are appreciated for they give added emphasis to the point which, while included in the treatment under section 19 of the paper, was probably not stressed as much as it might have been.

Mr. Ed Smith's summary of the control analysis and synthesis picture as it exists today is a very timely one. Wartime emphasis on extending analytical techniques in this field were very intense. The old oft-stated gap that used to exist between the various engineering professions is rapidly narrowing as the analytical techniques of the communications engineer are explained to the mechanical engineer, and the "systems" concept is applied more and more vigorously to control problems. The integration within the professions is being stimulated by discussions between professional groups who, until a few years ago, had few common problems. Nowadays one finds a great deal of mathematics, electronics, circuit theory, electrical machinery theory, hydraulic machinery theory being interwoven into a co-ordinated systems design with corresponding benefit to that branch of the art being developed by each group. But this interweaving or integrating is occurring with harmony and a degree of co-operation between engineers of widely different background, in a very interesting way.

[12] "Electrical Analogy Methods Applied to Servomechanism Problems," by G. D. McCann, S. W. Herwald, and H. S. Kirschbaum, Trans. A.I.E.E., vol. 65, February, 1946, pp. 91–96.

Conversion of Measurements of Power Output of Diesel Engines to Standard Atmospheric Conditions[1]

By MARTIN A. ELLIOTT,[2] BRUCETON, PA.

This report presents and analyzes data obtained on the effect of ambient or atmospheric conditions on the power output of three normally aspirated commercial four-stroke-cycle Diesel engines. Different ambient conditions were simulated by maintaining the desired pressure in each of two surge tanks, one connected to the intake and one connected to the exhaust of the engine. The simulated absolute pressures ranged from 20 to 30 in. of mercury. Most of the tests were made at the prevailing atmospheric temperature but a series of tests with one engine was made at an intake temperature of approximately 230 F. Analysis of the results showed that an algebraic relation could be obtained between so-called "derived indicated efficiency" and fuel:air ratio. This relation is independent of ambient conditions and is used in the development of rational procedures for correcting measurements of power output made at existing test atmospheric conditions to any arbitrarily selected standard ambient condition. Using certain of the basic relations developed, an expression was derived for computing the brake horsepower of an engine at any ambient density and rate of fuel consumption. Computed values were within 0.7 hp of values determined experimentally. The report also reviews previous work in the field and indicates the need for further work to obtain the complete solution of the problem desired by the A.S.M.E. Special Research Committee on Internal-Combustion Engines.

Introduction

In making performance tests of Diesel engines, differences in the density and moisture content of the ambient atmosphere are inevitable at different test locations and at different times in the same location. Since the power output of a Diesel engine operated at a given speed and throttle setting is affected by changes in the density of the intake air, it is desirable to develop methods for correcting the power output of an engine determined at existing ambient conditions to power output at some arbitrarily chosen standard ambient condition.

Members of this Society, who were instrumental in organizing the Special Research Committee on Internal-Combustion Engines, recognized the need for fundamental information on methods for expressing the power output of Diesel engines at standard ambient conditions, and a program of research (4)[3] was prepared outlining the work required to develop the necessary information. In formulating this program an analysis was made of performance data on Diesel engines obtained incidentally by the Bureau of Mines in studies of the effect of barometric pressure on the composition of the exhaust gases produced by Diesel engines. The tests were limited in scope in so far as engine performance is concerned since the main objective of the Bureau of Mines investigation was to determine the hazards of operating Diesel engines underground in places where the barometric pressure might be low. In spite of this, analysis of the data furnished considerable information on possible methods for correcting the power output of Diesel engines to standard ambient conditions and on the probable magnitude of the correction factors. The committee thought that it might be desirable to publish this information in a progress report discussing the general problem, analyzing the data available, and indicating the additional information required to obtain a complete solution. Accordingly the following report has been prepared:

The paper presents methods for converting measurements of power output determined at existing ambient conditions to power output at some arbitrarily selected standard ambient condition. This general procedure is commonly referred to in technical literature as correcting power output to standard ambient conditions and the multiplying factors for doing this are referred to as correction factors. This terminology is used in this paper but it is pointed out that the tests reported here were not broad enough in scope to permit the development of correction factors applicable to all engines. Because of this the somewhat more general term conversion has been used in the title to describe basic procedures developed.

Review of Previous Work

Tests on the effect of atmospheric conditions on the power output of Diesel engines are reported in the literature, but none of the previous investigations has provided a generally applicable method for correcting measurements of the power output of Diesel engines to standard ambient conditions.

The results of tests to determine the effect of humidity, temperature, and pressure on the power output of a four-cylinder, four-stroke-cycle Diesel engine were reported by Everett (5). From these tests it was concluded that, at a given speed and throttle setting, humidity did not affect power output significantly, and that both temperature and pressure affected power output but the fundamental variable was density.

Everett's work (5) and later work of Doolittle (2) furnish experimental evidence on the very minor effect of humidity when excess air is present.

The results obtained by Everett (5) on the effect of pressure on power output may not represent the effect under normal operating conditions because in his tests the pressures at the intake and

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[1] Progress Report No. 1, for the A.S.M.E. Special Research Committee on Internal-Combustion Engines. Published by permission of the Director, Bureau of Mines, United States Department of the Interior, Washington, D. C.

[2] Supervising Engineer, Internal-Combustion Engine Research Section, Explosives Division, Experimental Mine and Explosives Testing Station.

Contributed by the Special Research Committee on Internal-Combustion Engines and presented under the auspices of the Oil and Gas Power Division at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

TABLE 1 DESCRIPTION OF ENGINES TESTED

Engine Type	A Four-stroke-cycle, normally aspirated	B Four-stroke-cycle, normally aspirated	C Four-stroke-cycle, normally aspirated
Number of cylinders	4	4	6
Cylinder bore, in.	$4^1/_4$	4	$4^7/_8$
Piston stroke, in.	$5^1/_2$	$4^1/_2$	6
Piston displacement, cu in.	312.1	226.2	672
Maximum rated speed, rpm	1400	2600	1800
Maximum rated bhp with all accessories except fan	44	70	150
Fuel pump	Individual pump for each cylinder; fuel delivery controlled by pump-plunger by-pass	Individual pump for each cylinder; fuel delivery controlled by pump-plunger by-pass	Single-plunger, low-pressure metering and distributing type
Type of injection valve	Single hole orifice, flat-faced valve seat	Circumferential orifice (pintle nozzle); conical valve seat	Cam-actuated plunger-type injector for each cylinder. Six spray holes in each injector
Operating pressure of injection valve discharging into air at atmospheric pressure, psi	1500	1650	...
Combustion system	Cylindrical precombustion chamber with cone-shaped ends	Spherical turbulence or air-swirl chamber	Directed spray; small air-cell in piston head
Cooling system	Positive circulation, thermostatically controlled	Positive circulation, thermostatically controlled	Positive circulation, thermostatically controlled

final exhaust were not equal. For example, the pressure at the intake ranged from 10 to 16 psia, but the pressure at the final exhaust was always atmospheric (28.79–29.15 in. of mercury). Differences between the pressure at the intake and final exhaust may cause flow into or out of the cylinder during the period of valve overlap, depending upon the relation between the two pressures. This condition is abnormal, and it is desirable therefore to maintain equal pressures at the intake and final exhaust in studies of the effect of ambient conditions on the power output of Diesel engines.

Results reported by Taylor (11) and some of the results reported by Doolittle (2) also were obtained by operating engines at reduced intake pressures and atmospheric exhaust pressure and therefore may not represent normal operation at reduced ambient pressures. Data given by Taylor (11) and Doolittle (2) on the effect of temperature apply only to the engine tested and no attempt was made to generalize the results quantitatively although qualitative generalizations were made by Taylor (11).

Dennison (1) proposed a method for comparing the performance of different Diesel engines, but this method was intended to facilitate comparison of competitive designs and is not adaptable to the development of correction factors for power output.

Moore and Collins (8) have published the results of extensive tests on the effect of atmospheric conditions on the performance of a single-cylinder Diesel engine. These tests included determination of the effect of temperature at constant pressure; the effect of pressure at constant temperature; and the effect of changing both temperature and pressure simultaneously. All of these tests were made with the pressure at the intake equal to the pressure at the final exhaust. In addition, tests were made with intake pressure greater than and less than the pressure at the final exhaust. Moore and Collins contributed considerable information but many of their tests were made at fuel : air ratios far on the rich side, a condition of little practical interest. Furthermore, their analysis of the results was not carried sufficiently far to develop the fundamental theory on the effect of atmospheric conditions on power output of Diesel engines.

In preparation for an investigation by the Bureau of Mines on the effect of atmospheric conditions on the exhaust gases produced by Diesel engines, a study was made of the published results mentioned. This study disclosed that correlation of indicated thermal efficiency with fuel : air ratio had not been examined in analyzing the results of performance tests made at different ambient conditions. Such a correlation is of interest because thermodynamic considerations show that these factors are related in the theoretical fuel : air cycle (6, 10). Furthermore, it might be expected that this fundamental relationship would be unaffected by changes in ambient conditions at least over a relatively wide range of conditions. That this was true was indicated by an analysis (3) of the data of Moore and Collins (8). Once this had been established it was possible to suggest a rational basis (3) for developing methods of correcting the power output of a Diesel engine to standard ambient conditions.

Scope of Present Report

This report includes a discussion of the results of performance tests made at different simulated ambient conditions on three different commercial, four-stroke-cycle, normally aspirated Diesel engines. Methods for analyzing the data are presented and methods for correcting power output to standard ambient conditions are developed and applied to the results of actual tests. These methods are modifications or extensions of the fundamental theory presented in a previous paper (3) by the author.

APPARATUS AND PROCEDURE

Engines Tested. The principal features of the three engines tested are summarized in Table 1. Engines B and C were directly coupled to the dynamometer. Engine A was coupled to the dynamometer through a clutch. An auxiliary cooling system was used and all engines were tested without radiator or fan. The cooling-water circulating pump was driven by the engine in all tests.

Fuel. The chemical and physical properties of the fuels used are given in Table 2. Fuel F was used in tests of engine A at 600,

TABLE 2 CHEMICAL AND PHYSICAL PROPERTIES OF FUELS

Designation of fuel	F	G
Flash point (P.M.C.C.), F.	157	150
Water and sediment	Trace	Trace
Viscosity, S.U. at 100 F, sec.	34	33
centipoises at 100 F.	1.93	1.74
Carbon residue on 10% bottoms, per cent by weight	0.010	0.016
Specific gravity 60/60 F.	0.8295	0.8314
Pour point (upper), F.	—2	—20
Cetane number	Not less than 50	Not less than 50
Sulphur, per cent by weight	0.1	0.2
Hydrogen, per cent by weight	13.4	13.4
Carbon, per cent by weight	86.1	86.2
Oxygen, per cent by weight	0.4	0.3
Nitrogen, per cent by weight	0.0	0.0
Heating value, (upper), Btu per lb	19590	19560

1000, and 1400 rpm; fuel G in tests of engine B; and fuel F in tests of engine C.

Dynamometers. Tests of engines A and C were made using a water-cooled induction-type dynamometer rated at 200 hp from 1200 to 4000 rpm. A direct-current dynamometer rated at 45 hp from 1200 to 2500 rpm was used in tests of engine B. This latter dynamometer was used in determining friction horsepower in motoring tests of all three engines.

Equipment for Simulating Operation at Different Ambient Conditions

Operation at different ambient conditions was simulated by maintaining the desired pressure in each of two surge tanks. One of these tanks was connected to the intake of the engine and one to the exhaust (see Fig. 1). The intake surge tank was 17.5 in. diam and 84 in. in length. The exhaust surge tank was 22 in. diam and 60 in. in length.

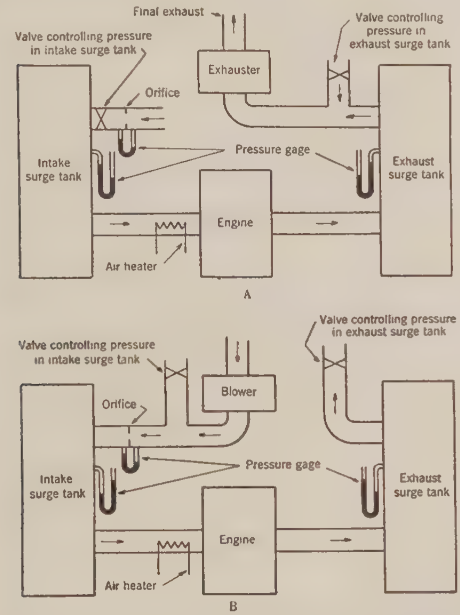

Fig. 1 Diagrammatic Sketch of Arrangements for Tests at (*A*) Ambient Pressures Less Than Prevailing Barometric Pressure and (*B*) Ambient Pressures Greater Than Prevailing Barometric Pressure

In tests at low ambient pressures (20 and 25 in. of mercury abs), the desired pressure in the intake tank was obtained by throttling a valve on the upstream side of the tank. The pressure in the exhaust surge tank was maintained the same as that in the intake surge tank by a Roots-Connersville exhauster driven in some tests by a gasoline engine and in others by an electric motor (see Fig. 1*A*).

In tests at a simulated ambient pressure of 30 in. of mercury abs, which was greater than the prevailing barometric pressure, the Roots-Connersville exhauster was used to compress air to the intake surge tank. The pressure in the exhaust surge tank was maintained at 30 in. of mercury abs by throttling a valve on the downstream side of the tank (see Fig. 1*B*).

In some tests arrangements were made for heating the intake air by manually controlled strip heaters installed in an insulated pipe 6 in. diam just upstream from the intake manifold (see Fig. 1).

Test Procedure

The desired pressures in the intake and exhaust surge tank were obtained as just outlined when the engine was operating at the speed and load selected. Final adjustments were then made in speed, load, and pressure in intake and exhaust surge tanks. The engine was then operated until equilibrium was attained and the test data recorded.

Fuel consumption was determined by an automatic fuel-weighing device which operated an electric clock and revolution counter.

Air was measured by noting the differential pressure across a thin-plate orifice upstream from the intake surge tank. The orifice and connections were installed in accordance with the published recommendations of the A.S.M.E. Special Research Committee on Fluid Meters.

In most tests with each engine the composition of the exhaust gases was determined by analysis in a Haldane apparatus. From this analysis and from the analysis of the fuel it was possible to compute the fuel:air ratio. The values obtained agreed very closely (generally within ±2 or 3 per cent) with values computed from direct measurements of fuel and air consumption.

It was not possible to motor engines connected to the induction-type dynamometer. Therefore friction-horsepower determinations were made with the engines motored by the direct-current dynamometer. In these tests the engine was warmed up

TABLE 3 RESULTS OF TESTS OF ENGINE B AT 1000 RPM NORMAL TEMPERATURES AND DIFFERENT SIMULATED AMBIENT PRESSURES

Test no.	Brake[1,2] hp.	Corrected brake hp[3]	Fuel consumption, lb per hr[2]	Fuel: air ratio, lb per lb	Volumetric efficiency, percent	Temperature at intake, F	Density at intake, lb per cu ft
Nominal absolute pressure at intake and exhaust, 30 in. of mercury[4]							
47	0.3	0.3	2.05	0.0079	90	78	0.0728
51	0.9	0.9	2.28	0.0094	87	92	0.0708
48	2.5	2.5	2.58	0.0103	88	80	0.0725
56	2.6	2.6	2.74	0.0108	87	76	0.0738
49	5.0	5.1	3.30	0.0133	88	81	0.0721
57	5.1	5.1	3.42	0.0135	88	80	0.0735
52	7.5	7.6	4.13	0.0168	87	85	0.0718
58	10.0	10.1	4.92	0.0197	87	82	0.0733
53	12.5	12.7	5.67	0.0232	87	86	0.0715
50	12.5	12.8	5.66	0.0237	84	83	0.0720
59	15.0	15.3	6.50	0.0261	87	83	0.0730
60	17.5	17.7	7.34	0.0294	87	84	0.0733
54	20.0	20.4	8.27	0.0343	86	89	0.0711
61	22.5	22.9	9.22	0.0378	86	86	0.0726
62	25.0	25.7	10.41	0.0429	85	88	0.0725
55	27.5	28.6	11.68	0.0498	84	91	0.0709
63	27.5	28.1	11.50	0.0480	84	90	0.0722
Nominal absolute pressure at intake and exhaust, 25 in. of mercury[5]							
97	0.2	0.3	1.98	0.0096	90	100	0.0588
75	0.3	0.4	2.03	0.0100	88	102	0.0587
76	5.1	5.4	3.38	0.0167	88	102	0.0587
77	10.1	10.7	5.00	0.0248	87	102	0.0589
78	15.0	16.1	6.76	0.0341	86	103	0.0587
111	15.2	16.1	6.67	0.0319	87	80	0.0612
112	17.6	18.8	7.71	0.0371	86	82	0.0611
79	20.1	22.0	8.80	0.0456	84	102	0.0587
113	22.6	24.7	10.14	0.0496	85	84	0.0610
114	23.8	26.4	11.18	0.0559	84	84	0.0608
80	25.2	29.3	12.43	0.0656	82	103	0.0586
Nominal absolute pressure at intake and exhaust, 20 in. of mercury[6]							
87	0.4	0.6	2.06	0.0118	84	100	0.0469
100	0.4	0.6	2.10	0.0131	87	100	0.0469
104	0.3	0.5	2.13	0.0133	88	100	0.0466
83	5.0	5.4	3.30	0.0151	86	101	0.0471
89	10.2	11.5	5.11	0.0328	84	101	0.0470
101	12.6	14.4	6.12	0.0396	83	100	0.0475
90	15.1	17.7	7.19	0.0467	83	101	0.0472
115	15.2	17.5	7.18	0.0434	86	73	0.0491
116	17.3	20.6	8.44	0.0520	84	81	0.0484
117	18.3	22.1	9.03	0.0570	83	82	0.0484
119	18.8	23.4	9.88	0.0625	83	86	0.0480
118	19.2	25.0	11.06	0.0706	83	82	0.0484
91	19.6	27.7	12.45	0.0840	80	100	0.0471

[1] At test intake conditions.
[2] Corrected to nominal speed of 1000 rpm.
[3] Corrected to intake conditions of 29.92 in. of mercury absolute pressure, temperature 60 F, density of 0.0765 lb per cu ft. Equation [23] used.
[4] Actual range of pressure 29.5—30.1 in. of mercury.
[5] Actual range of pressure 24.9—25.1 in. of mercury.
[6] Actual range of pressure 19.9—20.3 in. of mercury.

until the equilibrium lubricating-oil temperature was attained and motoring friction determined at this condition.

Test Results

The results of a typical series of tests on one of the engines are given in Table 3. To eliminate minor effects of deviations of actual speed from the nominal speed, the data on speed, fuel consumption, and horsepower are corrected to the nominal speed by assuming horsepower and fuel consumption to be directly proportional to speed. The correction is minor as the average speed was generally within ±5 rpm of the nominal speed and never exceeded ±15 rpm.

Discussion of Results

Basis for Comparing Results. The effect of ambient conditions on the power output of a Diesel engine can be determined by maintaining constant all engine operating conditions and all ambient conditions except the one whose effect is being studied. This means that comparisons of the results of tests in which a particular ambient condition is varied must be made at the same speed and throttle setting. In the Diesel engine the position of the throttle or throttle setting fixes the quantity of fuel injected at a given engine speed. If both speed and throttle setting are maintained constant, it is obvious that the quantity of fuel supplied, or in other words the fuel consumption, will be constant and will not be affected by ambient conditions. Therefore comparison of power outputs obtained at different ambient conditions must be made at the same rate of fuel consumption. This comparison can be made most conveniently by plotting brake horsepower against fuel consumption for each ambient condition studied.

Effect of Ambient Conditions on Brake Horsepower

The effect of ambient pressure on the brake horsepower of engines A, B, and C is shown in Figs. 2, 3, 4, and 5. The effect of ambient temperature (at ambient pressures of 30 and 20 in. of mercury abs) on the brake horsepower of engine B is shown in Fig. 6.

These results are presented to give a general idea of the magnitude of the effect of ambient conditions on brake horsepower and to show how the effect varies throughout the operating range of the engine tested. The following generalization may be made:

Reductions in ambient pressure or increases in ambient temperature cause significant reductions in brake horsepower at high rates of fuel consumption (higher loads). Ambient pressure or temperature does not affect brake horsepower significantly at low rates of fuel consumption (light loads).

The foregoing discussion is on the basis of the effect of the independent variables, pressure and temperature. It will be shown subsequently that density is the fundamental variable, and from the foregoing it is evident that reductions in ambient density cause reductions in brake horsepower at the higher loads but that ambient density does not affect brake horsepower significantly at light loads.

If data similar to those presented in Figs. 2 to 6, inclusive, were available for an engine it would be possible to determine correction factors graphically. However, it is desirable to develop other methods that might be more convenient and more generally applicable. In the succeeding sections the data are analyzed further to develop more rational procedures for determining correction factors for power output.

Relation Between Indicated Efficiency and Fuel : Air Ratio

The theoretical or maximum possible thermal efficiency of a given internal-combustion engine can be computed from the thermodynamic properties of the working fluid. Since these properties are related to fuel : air ratio it is evident that thermal efficiency must also be related to fuel : air ratio.

The relation between computed maximum thermal efficiency and fuel : air ratio for a limited-pressure fuel:air cycle and for an assumed compression ratio of 13.9 to 1 is shown in Fig. 7. The details of this computation have been reported previously (3) but it should be mentioned here that the limited-pressure cycle was chosen because it represents a rational criterion for comparing actual cycles of high-speed Diesel engines. The limited pressure at a given fuel : air ratio was obtained from data published by

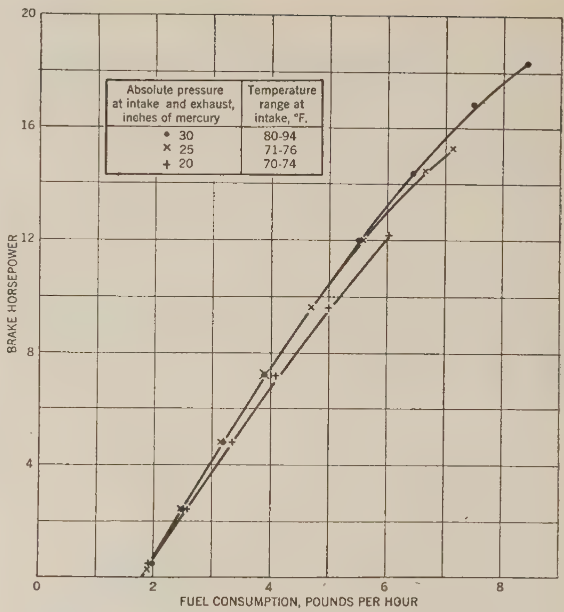

Fig. 2 Effect of Ambient Pressure on Brake Horsepower of Engine A at 600 Rpm

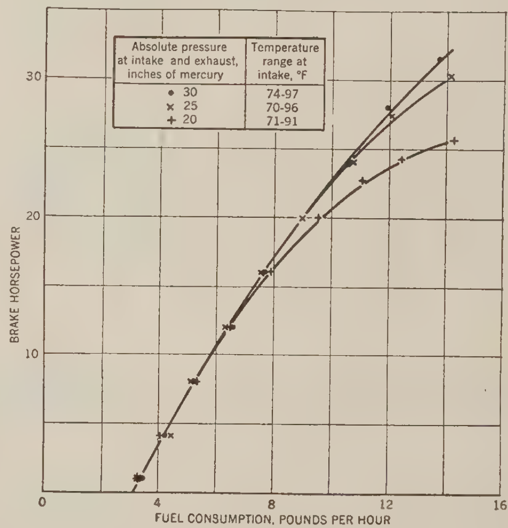

Fig. 3 Effect of Ambient Pressure on Brake Horsepower of Engine A at 1000 Rpm

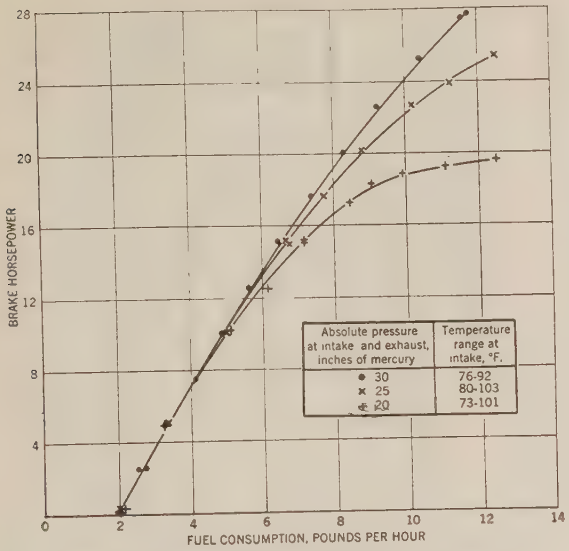

Fig. 4 Effect of Ambient Pressure on Brake Horsepower of Engine B at 1000 Rpm

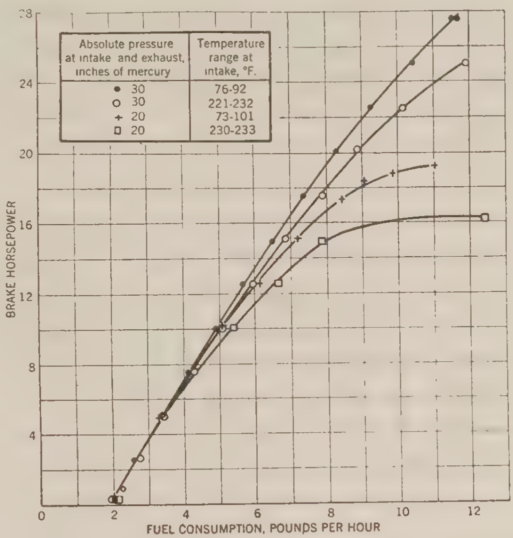

Fig. 6 Effect of Ambient Temperature on Brake Horsepower of Engine B at 1000 Rpm

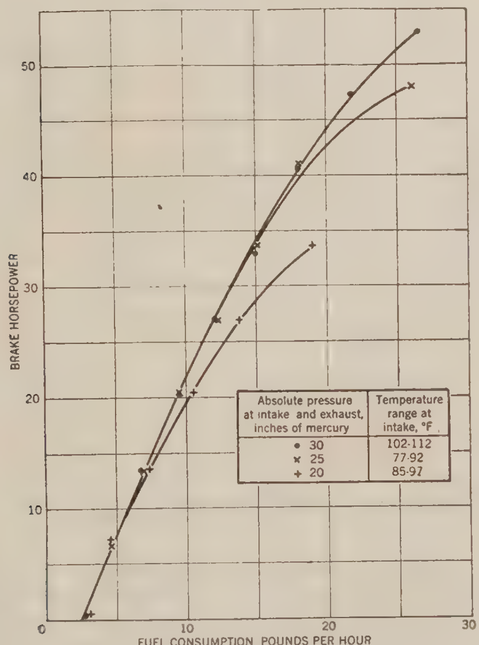

Fig. 5 Effect of Ambient Pressure on Brake Horsepower of Engine C at 600 Rpm

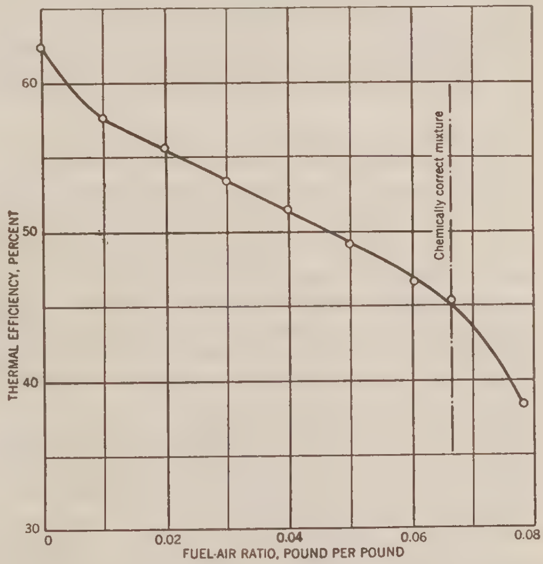

Fig. 7 Relation Between Computed Thermal Efficiency and Fuel : Air Ratio for a Limited Pressure Fuel-Air Cycle With Octane Fuel and an Assumed Compression Ratio of 13.9

Rothrock and Waldron (9) who made tests on an engine having a compression ratio of 13.9 : 1.

Fig. 7 shows that at fuel : air ratios, within the normal operating range of a Diesel engine, the relation between computed thermal efficiency and fuel:air ratio is linear. Since this relation is fundamental it might be expected that it would be independent of ambient conditions. It is of interest therefore to examine the relation between indicated efficiency and fuel : air ratio for the actual test results obtained at different ambient conditions.

Relation Between Fuel : Air Ratio and Indicated Efficiency Determined From Brake Horsepower and Motoring Friction

Facilities for determining indicated horsepower directly were not available in the Bureau of Mines tests. Accordingly, as an approximation, indicated horsepower was assumed to be equal to brake horsepower plus friction horsepower determined in motoring tests. It is recognized that indicated horsepower determined in this manner may be in error because of the well-known limitations (10) in determining friction horsepower in motoring tests. It should be mentioned, however, that errors in determining fric-

tion horsepower become less significant at the higher power outputs because if these errors are not too great they represent only a relatively small fraction of the indicated horsepower. For this reason trends at higher power outputs (higher fuel : air ratios) can be considered as being relatively unaffected by errors in determining friction horsepower.

For convenience, indicated efficiency will be expressed as indicated horsepower-hours per pound of fuel (ihp ÷ lb of fuel per hr). This factor does not have the same numerical value as indicated efficiency, but it is directly proportional to indicated efficiency when the heating value of the fuel is substantially constant as it was in the tests reported herein. For brevity, this factor will be referred to as "indicated efficiency" in this report. Obviously, indicated horsepower-hours per pound of fuel can be converted to indicated thermal-efficiency ratio by multiplying by the factor 2545 ÷ (heating value of fuel, Btu per lb). This conversion would of course be essential in analyzing results obtained with fuels having widely different heating values.

The relation between indicated efficiency determined as outlined and fuel : air ratio is shown in Figs. 8, 9, 10, 11, and 12. Figs. 11 and 12 which represent, respectively, results of test of engine B at 1000 rpm and engine C at 600 rpm, show that the relation is linear and independent of ambient conditions at fuel : air ratios greater than approximately 0.02 lb per lb. Figs. 8, 9, and 10 which represent, respectively, the results of tests of engine A at 600, 1000, and 1400 rpm, show that for all practical purposes the relation is independent of ambient conditions and could be approximated by a linear function at fuel : air ratios greater than approximately 0.025 lb per lb. However, at fuel : air ratios less than this the results of tests of engine A indicate that the relation is not independent of ambient conditions and is not linear.

Since theoretical considerations strongly suggest that the relation between indicated efficiency and fuel : air ratio should be unaffected by changes in ambient conditions, it is necessary to seek an explanation for deviations of test results from this presumption.

If combustion efficiency is affected by ambient conditions, it is obvious that indicated efficiency will not be independent of ambient conditions as predicted by thermodynamic considerations which assume complete combustion. In the tests reported here precise gas-analysis methods were used and carbon monoxide and aldehydes, which are products of incomplete combustion, were determined with considerable precision. The method used for carbon monoxide is sensitive to 0.01 per cent carbon monoxide and the method for aldehydes is sensitive to 5 ppm. The results of such gas analyses showed that the concentration of these products of incomplete combustion at a given fuel : air ratio were not affected significantly by changes in ambient pressure. Since there was no detectable increase in smoke production, it would seem that the apparent reductions in indicated efficiency at the low ambient pressures and low fuel : air ratios could not be accounted for on the basis of reductions in combustion efficiency.

Combustion knock was observed at the lower ambient pressures. Ignition delay probably is greater under such conditions because of the lower partial pressure of oxygen. The apparent effect of ambient conditions on indicated efficiency might have been due to the increased tendency to knock. However, if this were the case it would seem that a significant effect would have been observed at the higher fuel:air ratios where combustion knock was most severe. Since the indicated efficiency was apparently affected significantly by ambient pressure only at the lower fuel:air ratios, it is possible that combustion knock may not have been a factor. This cannot be stated definitely without further refinements in testing methods.

Indicated horsepower is only approximated when it is deter-

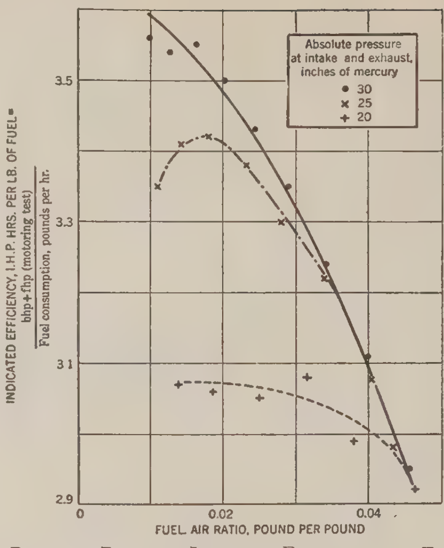

Fig. 8 Relation Between Indicated Efficiency and Fuel : Air Ratio in Tests of Engine A at 600 Rpm

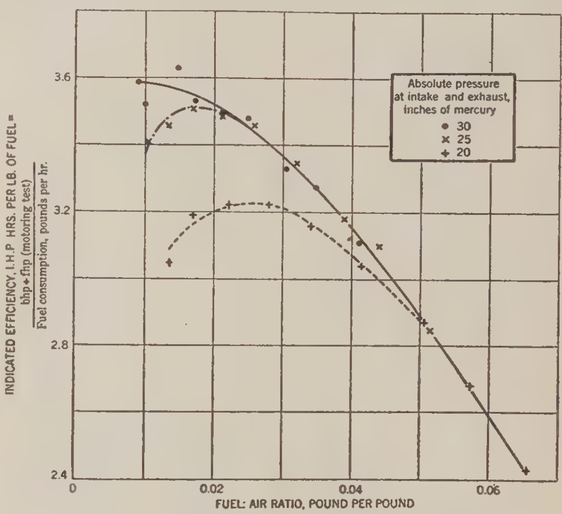

Fig. 9 Relation Between Indicated Efficiency and Fuel : Air Ratio in Tests of Engine A at 1000 Rpm

mined from measurements of brake horsepower and friction horsepower in motoring tests. As mentioned previously any inaccuracy in determining friction horsepower will be a larger proportion of the indicated horsepower at light loads (low fuel : air ratios) than at higher loads. It is possible therefore that the apparent effect of ambient pressure on indicated efficiency at the lower fuel:air ratios may have been only the result of unavoidable inaccuracies in determining friction horsepower.

It seems desirable therefore to seek a method for approximating indicated horsepower and ultimately indicated efficiency that does not involve determination of friction horsepower in a motoring test. Such a method is described in the following section.

Derived Indicated Efficiency

As mentioned previously, theoretical considerations point to a relation between indicated efficiency and fuel : air ratio that is

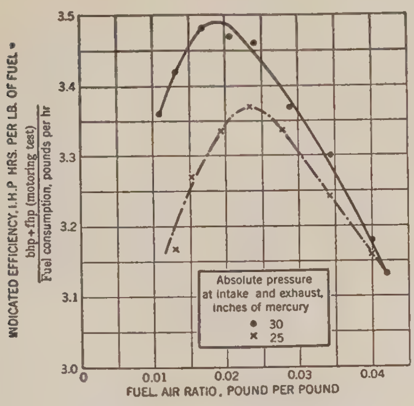

FIG. 10 RELATION BETWEEN INDICATED EFFICIENCY AND FUEL : AIR RATIO IN TESTS OF ENGINE A AT 1400 RPM

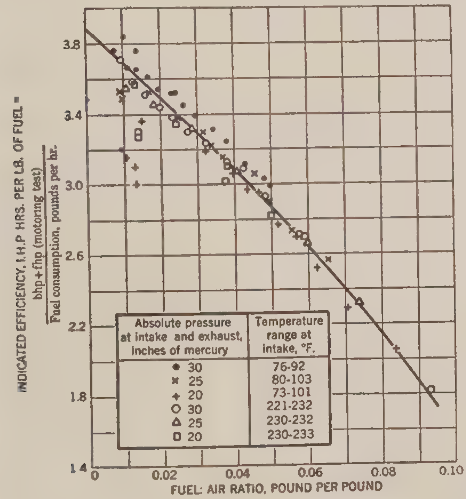

FIG. 11 RELATION BETWEEN INDICATED EFFICIENCY AND FUEL : AIR RATIO IN TESTS OF ENGINE B AT 1000 RPM

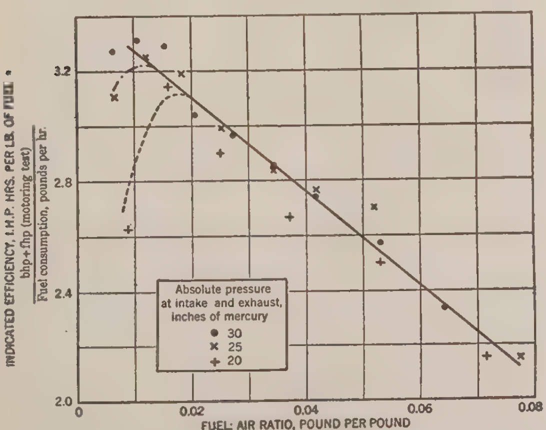

FIG. 12 RELATION BETWEEN INDICATED EFFICIENCY AND FUEL : AIR RATIO IN TESTS OF ENGINE C AT 600 RPM

independent of ambient conditions. The test results analyzed, as outlined, show that for all practical purposes this is true at the higher fuel : air ratios and that the relation is linear under these conditions. Let us assume therefore that linear relation exists in the actual engine and let the indicated efficiency determined from this relation be designated as "derived indicated efficiency." The assumed linear relation can be expressed by the following equation

$$e = e_0 - af \quad \ldots [1]$$

in which

e = derived indicated efficiency, expressed as ihp-hr per lb of fuel

f = fuel : air ratio, lb per lb

e_0 = intercept on y-axis (constant for a given engine and given engine speed)

a = slope (constant for a given engine and given engine speed)

Using the following nomenclature:

b = brake horsepower

c = derived friction horsepower (assumed constant for a given engine and given engine speed)

i = derived indicated horsepower

F = fuel consumption, lb per hr

v = volumetric efficiency (expressed as a ratio)

ρ = density of ambient atmosphere, lb per cu ft

Q = volume displaced by pistons on intake stroke, cu ft per hr

Then

$$i = eF \quad \ldots [2]$$

Substituting in Equation [2] the value of e from Equation [1], we obtain

$$i = (e_0 - af)F \quad \ldots [3]$$

Since

$$i = b + c \quad \ldots [4]$$

then from Equations [3] and [4]

$$b + c = (e_0 - af)F \quad \ldots [5]$$

If a series of tests is made at constant speed, at selected loads from no load to full load, and at a given set of ambient conditions, the value of b, F, and f will be obtained at each test condition. If we select from the data one set of values b_1, F_1, and f_1 then from Equation [5]

$$b_1 + c = (e_0 - af_1)F_1 \quad \ldots [6]$$

Subtracting Equation [6] from [5] and assuming that c is constant at a given speed and is independent of load

$$b - b_1 = e_0(F - F_1) - a(fF - f_1F_1) \quad \ldots [7]$$

Dividing Equation [7] by $(F - F_1)$ we obtain

$$\frac{b - b_1}{F - F_1} = e_0 - a\,\frac{fF - f_1F_1}{F - F_1} \quad \ldots [8]$$

Remembering that b, F, and f can be any set of values in a series of tests at constant speed but at different loads and that b_1, F_1, and f_1 are one particular set of values chosen arbitrarily, then Equation [8] shows that if $\frac{b - b_1}{F - F_1}$ is plotted against $\frac{fF - f_1F_1}{F - F_1}$, a straight line should result and it will be possible to evaluate the constants e_0 and a. It is apparent therefore that Equation [8] offers a method for determining the relation between derived indicated efficiency and fuel : air ratio by measuring only brake horsepower, fuel : air ratio, and fuel consumption in a series of tests at constant speed but at different loads.

To determine the applicability of the foregoing method of

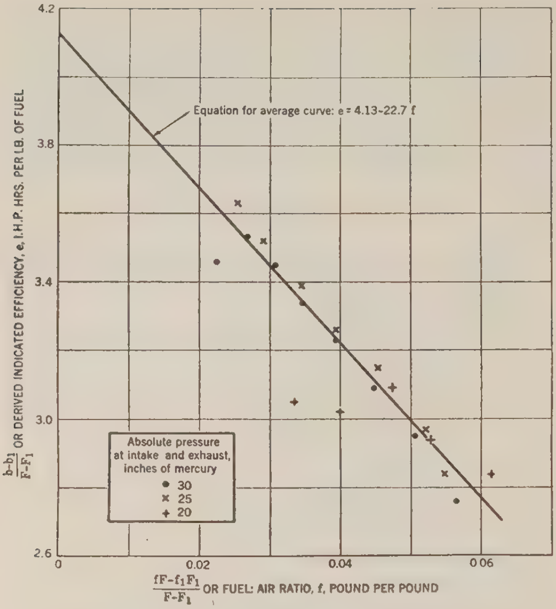

Fig. 13 Relation Between Derived Indicated Efficiency and Fuel : Air Ratio for Engine A at 600 Rpm

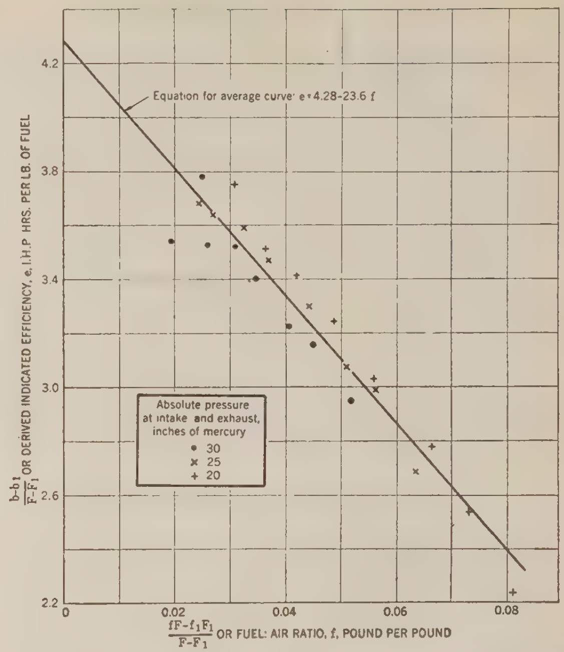

Fig. 14 Relation Between Derived Indicated Efficiency and Fuel : Air Ratio for Engine A at 1000 Rpm

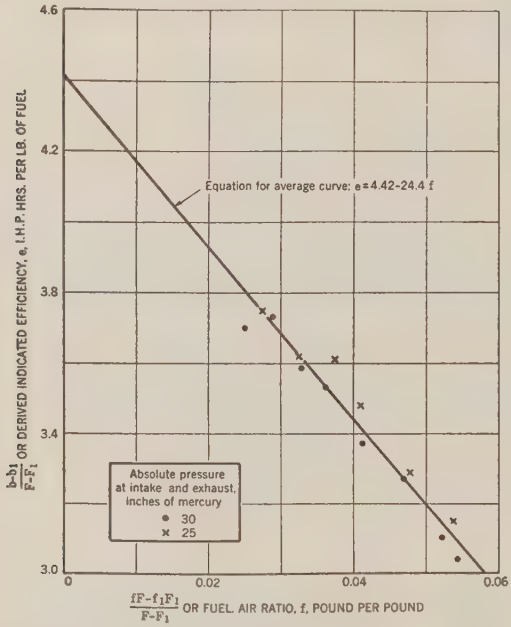

Fig. 15 Relation Between Derived Indicated Efficiency and Fuel : Air Ratio for Engine A at 1400 Rpm

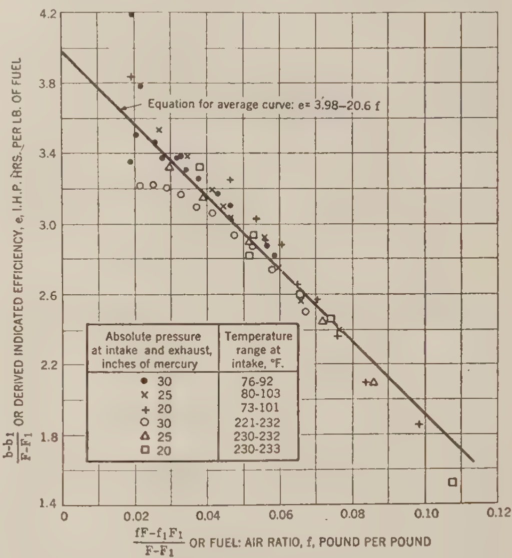

Fig. 16 Relation Between Derived Indicated Efficiency and Fuel : Air Ratio for Engine B at 1000 Rpm

analysis, the values of $\frac{b - b_1}{F - F_1}$ and $\frac{fF - f_1F_1}{F - F_1}$ were computed from the test results and are plotted in Figs. 13, 14, 15, 16, and 17. In making the computation, the test at minimum load for each constant ambient condition was arbitrarily chosen for determination of the value of b_1, F_1, and f_1.

Figs. 13, 14, and 15, respectively, represent test results obtained with engine A at 600, 1000, and 1400 rpm. It will be observed that at a given speed the results obtained at all ambient conditions can be represented with good precision by a linear relation. The same conclusion is indicated by Fig. 16 which represents data obtained in tests with engine B at 1000 rpm, and by Fig. 17 which represents data obtained in tests with engine C at 600 rpm. The average deviation from the average curve is approximately 0.08 ihp-hr per lb of fuel. Thus the average curve represents derived indicated efficiency with an average relative error that

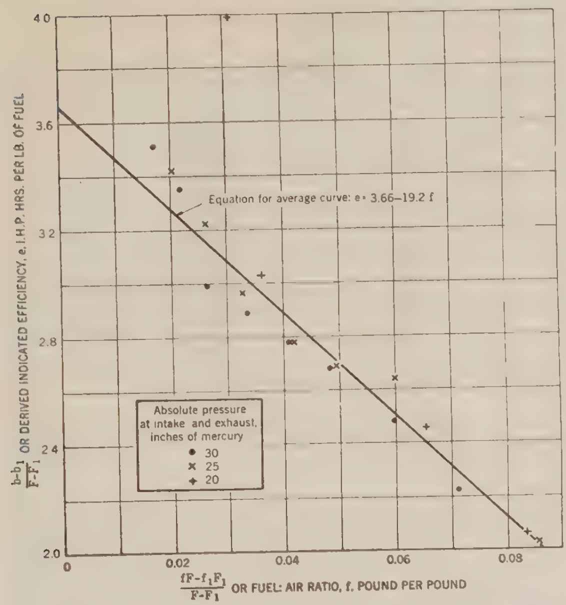

FIG. 17 RELATION BETWEEN DERIVED INDICATED EFFICIENCY AND FUEL : AIR RATIO FOR ENGINE C AT 600 RPM

ranges from approximately 2 to 3 per cent. It will be observed that individual points are dispersed at the lowest values of $\frac{fF - f_1F_1}{F - F_1}$ (see Fig. 16). This occurs because the different $b - b_1$ cannot be determined precisely when this difference is low. For example, an absolute error of 0.2 hp represents a relative error of 10 per cent when $b - b_1$ is of the order of 2 hp.

The values of the parameters e_0 and a were determined by the method of averages (7) and are given in Table 4. In determining the average points the data for each engine speed were divided into two groups above and below an arbitrarily selected value of $\frac{fF - f_1F_1}{F - F_1}$. The values selected were in the middle of the range of this term and were as follows: 0.04 for tests of engine A; 0.05 for tests of engine B; and 0.045 for tests of engine C. Table 4 also shows values of $\frac{e_0}{a}$ as these are required in a subsequent section of the report.

TABLE 4 VALUES OF e_0 AND a IN RELATION $e = e_0 - af$

Engine	Engine speed, rpm	e_0	a	$\frac{e_0}{a}$
A	600	4.13	22.7	0.182
A	1000	4.28	23.6	0.181
A	1400	4.42	24.4	0.181
B	1000	3.98	20.6	0.194
C	600	3.66	19.2	0.191

It will be observed that the values of e_0 and a appear to be affected by engine speed and type of engine. Therefore each set of values applies only to a particular engine and a particular engine speed. However, from the limited data available it appears that $\frac{e_0}{a}$ is approximately the same for the three engines tested and may not be affected by engine speed. If this is true for a number of engines, it would be possible to develop one set of correction factors applicable to all engines.

In connection with the apparent constancy of $\frac{e_0}{a}$, it is of interest to note that $\frac{e_0}{2a}$ is the fuel : air ratio at which both derived indicated horsepower and brake horsepower pass through a maximum. This can be shown by substituting for F its value $fQv\rho$ in Equation [3] or [5]; taking the derivative of i or b with respect to f; equating the derivative to zero and solving for f. Since the fuel : air ratio at which maximum horsepower is obtained is a function chiefly of the thermodynamic properties of the working fluid, it is not surprising that $\frac{e_0}{a}$ which is twice this ratio should be approximately the same for different engines. It should be pointed out that Equation [1] only approximates derived indicated efficiency at fuel : air ratios on the rich side and therefore $\frac{e_0}{2a}$ probably is not the exact fuel : air ratio that results in maximum power. Nevertheless $\frac{e_0}{2a}$ is an approximation of this particular fuel : air ratio and therefore it is not unreasonable to expect that it has the same general property of being determined principally by thermodynamic considerations and of being more or less independent of engine design.

From the foregoing it is apparent that we have a relation that is independent of ambient conditions and that can therefore be used in the development of rational procedures for correcting power output to standard ambient conditions. It is not claimed that the value of e obtained from Equation [1] is the true indicated efficiency, and for that reason it has been called derived indicated efficiency. Although the value of e is somewhat empirical, nevertheless we can operate algebraically with Equation [1] since it represents actual test data with reasonable precision.

The evaluation of the parameters e_0 and a permits evaluation of the parameter c in Equation [5]. It will be remembered that c is designated as "derived friction horsepower." Solving Equation [5] for c we obtain

$$c = (e_0 - af)F - b \qquad [9]$$

Since the values of b, F, and f are known for any test condition it is possible to determine c.

The values of c obtained using the values of e_0 and a for the engine and engine speed in question (see Table 4) are given in Table 5. Values of $\frac{c}{Q}$ are also given as they are required in a subsequent section of the report.

It will be observed that the average values of derived friction horsepower given in Table 5 are always greater than the friction horsepower determined in motoring tests (except for engine B at 30 inches absolute pressure and normal temperature). In general, the value of c tends to decrease as the ambient pressure is reduced. This same tendency was observed in the friction horsepower determined in motoring tests. The maximum and minimum values of c for each series of tests are given in Table 5 and indicate that at a given engine speed c is "constant" within reasonable limits. The computed values of c did not appear to be related to fuel : air ratio.

METHODS OF CORRECTING POWER-OUTPUT DETERMINATIONS TO STANDARD AMBIENT CONDITIONS

Derivation of Relations. Starting with Equation [1] which is independent of ambient conditions, we can derive expressions for converting measurements of power output to standard ambient conditions. Several expressions can be derived depending upon which parameters are retained. One such expression is derived in this section and two other forms of the correction equation are given.

If the subscript s denotes standard ambient conditions and the

TABLE 5 VALUES OF DERIVED FRICTION HORSEPOWER, c

Engine	Engine speed, rpm	Absolute pressure at intake and exhaust, in. of mercury	Temperature range at intake, F	Values of c Maximum	Minimum	Average	Friction horsepower from motoring test	$\frac{c}{Q}$
A	600	30	80–94	7.8	7.1	7.3	6.6	0.00224
A	600	25	71–76	7.2	6.8	6.9	6.0	0.00212
A	600	20	70–74	7.1	6.4	6.8	5.3	0.00209
A	1000	30	74–97	14.0	12.6	13.2	11.2	0.00247
A	1000	25	70–96	13.1	11.6	12.0	10.0	0.00223
A	1000	20	71–91	13.3	11.5	11.9	9.1	0.00220
A	1400	30	90–105	20.6	19.5	19.8	15.4	0.00261
A	1400	25	76–80	18.9	18.4	18.7	14.2	0.00246
B	1000	30	76–92	7.7	6.9	7.3	7.4	0.00186
B	1000	25	80–103	7.8	6.7	7.3	6.8	0.00186
B	1000	20	73–101	7.8	6.6	7.1	6.1	0.00181
B	1000	30	221–232	8.3	7.3	7.7	7.2	0.00196
B	1000	25	230–232	7.5	7.2	7.3[a]	6.8	0.00186
B	1000	20	230–233	7.2	6.4	6.8[a]	6.2	0.00173
C	600	30	102–112	10.9	9.2	10.2[a]	8.7	0.00146
C	600	25	77–92	10.1	7.5	9.3[a]	8.2	0.00132
C	600	20	85–97	10.6	8.8	9.8[a]	7.7	0.00140

[a] Values of c in tests at fuel:air ratios greater than the chemically correct value not included in average.

subscript t denotes ambient conditions at the time of a test, then from Equation [3]

$$i_s = (e_0 - af_s)F_s \quad [10]$$

and

$$i_t = (e_0 - af_t)F_t \quad [11]$$

In converting power output to standard ambient conditions the rate of fuel consumption must be the same at the standard condition as at the test condition; therefore

$$F_s = F_t \quad [12]$$

and equating the values of F_s and F_t determined from Equations [10] and [11], we obtain

$$\frac{i_s}{e_0 - af_s} = \frac{i_t}{e_0 - af_t} \quad [13]$$

or

$$\frac{i_s}{i_t} = \frac{e_0 - af_s}{e_0 - af_t} \quad [14]$$

Subtracting 1 from both sides of Equation [14], we obtain upon simplification

$$\frac{i_s}{i_t} = 1 + \frac{1 - \frac{f_s}{f_t}}{\frac{e_0}{af_t} - 1} \quad [15]$$

Since

$$f = \frac{F}{Qv\rho} \quad [16]$$

then when $F_s = F_t$ we obtain

$$\frac{f_s}{f_t} = \frac{v_t\rho_t}{v_s\rho_s} \quad [17]$$

Substituting for $\frac{f_s}{f_t}$ in Equation [15] its value from Equation [17], we obtain

$$\frac{i_s}{i_t} = 1 + \frac{1 - \frac{v_t\rho_t}{v_s\rho_s}}{\frac{e_0}{af_t} - 1} \quad [18]$$

Without introducing a significant error it can be assumed that $v_t \doteq v_s$. The basis for this assumption is discussed in a subsequent section of this report. With this approximation Equation [18] becomes

$$\frac{i_s}{i_t} \doteq 1 + \frac{1 - \frac{\rho_t}{\rho_s}}{\frac{e_0}{af_t} - 1} = R \quad [19]$$

Equation [19] gives a relation for correcting indicated horsepower to standard ambient conditions. However, in practice the principal interest is in the correction of measurements of brake horsepower to standard ambient conditions. This may be done by substituting for i_s and i_t the values obtained from Equation [4] which gives

$$\frac{b_s + c_s}{b_t + c_t} = R \quad [20]$$

Equation [20] is not in a convenient form unless the value of c_s is known but from Equation [20] we may obtain the relation

$$\frac{b_s}{b_t} = R + \frac{c_tR - c_s}{b_t} \quad [21]$$

The value of c is not affected to a great degree by changes in ambient conditions. Therefore, as a close approximation we can assume $c_s \doteq c_t$ and we obtain

$$\frac{b_s}{b_t} \doteq R + \frac{c_t}{b_t}(R - 1) \quad [22]$$

Substituting in Equation [22] the value of R from Equation [19] and rearranging we obtain

$$\frac{b_s}{b_t} \doteq 1 + \frac{\left(1 - \frac{\rho_t}{\rho_s}\right)\left(\frac{c_t}{b_t} + 1\right)}{\frac{e_0}{af_t} - 1} \quad [23]$$

When e_0 and a are known Equation [23] can be used to correct to standard ambient conditions measurements of power output made at any ambient condition. It is necessary to know the fuel : air ratio at the test condition and the value of c_t. This latter factor can be computed from Equation [9] or it can be determined in a motoring test. A relatively large error in c_t has comparatively little effect on the value of the ratio $\frac{b_s}{b_t}$.

By retaining different parameters in the correction equation other forms may be obtained. The following form contains the parameters a and c

$$\frac{b_s}{b_t} = 1 - \frac{c_s - c_t}{b_t} - \frac{aF_t^2}{Qb_t}\left(\frac{1}{v_s\rho_s} - \frac{1}{v_t\rho_t}\right) \ldots\ldots [24]$$

With the simplifying assumption made previously, Equation [24] becomes

$$\frac{b_s}{b_t} \doteq 1 - \frac{aF_t^2}{Qv_tb_t}\left(\frac{1}{\rho_s} - \frac{1}{\rho_t}\right) \doteq 1 - \frac{af_tF_t}{b_t}\left(\frac{\rho_t}{\rho_s} - 1\right) \ldots [25]$$

Another form containing the parameters e_0 and c is as follows

$$\frac{b_s}{b_t} = \frac{e_0F_t}{b_t}\left(1 - \frac{v_t\rho_t}{v_s\rho_s}\right) + \left(1 + \frac{c_t}{b_t}\right)\frac{v_t\rho_t}{v_s\rho_s} - \frac{c_s}{b_t} \ldots\ldots [26]$$

With the simplifying assumptions made previously, Equation [26] becomes

$$\frac{b_s}{b_t} \doteq \frac{e_0F_t - c_t}{b_t}\left(1 - \frac{\rho_t}{\rho_s}\right) + \frac{\rho_t}{\rho_s} \ldots\ldots\ldots\ldots [27]$$

Correction of Test Results to Standard Ambient Conditions

The test results have been corrected to standard ambient conditions of 29.92 in. of mercury and 60 F[4] by using Equation [23]. This equation has been selected for illustrative purposes, although either Equation [25] or [27] could be used. Figs. 18, 19, 20, and 21 show brake horsepower at standard ambient conditions (corrected brake horsepower) in relation to fuel consumption. It will be observed from these figures that a single curve represents the relation between corrected brake horsepower and fuel consumption regardless of the ambient conditions of the tests. In general the average deviations from the average curve are of the order of 0.5 hp or less. Part of this deviation is caused by normal experimental errors; part is caused by deviations of actual results from the average relation between derived indicated efficiency and fuel : air ratio; and part is caused by the simplifying assumptions made in deriving Equation [23]. The largest deviations of corrected brake horsepower from the average curve occurred in two tests of engine B at the highest rate of fuel consumption and at an ambient pressure of 20 in. of mercury (see Fig. 20). In these two tests the fuel:air ratios (0.0706 and 0.0840 lb per lb, respectively) were greater than the chemically correct value of 0.068. At these high fuel : air ratios combustion is incomplete and the derived indicated efficiency is not linear with fuel : air ratio (see Fig. 7). Therefore actual test results are not represented precisely by the average curve.

From the foregoing it is apparent that Equation [23] yields reasonably precise results when used to convert brake horsepower at test ambient conditions to brake horsepower at standard ambient conditions. The same order of precision of corrected brake horsepower may be obtained by using either Equation [25] or Equation [27]. It appears therefore that the principles outlined here may be useful in developing rational methods for correcting brake-horsepower determinations to standard ambient conditions. However, before such methods can be developed for universal application it will be necessary to obtain information on many more engines and to study the performance of each engine more intensively than was possible in the Bureau of Mines investigation. As mentioned previously, this investigation was concerned principally with the effect of ambient conditions on exhaust-gas composition, and performance data were obtained only incidentally.

The principal reason for choosing Equation [23] for illustrative purposes is that it can be put into a form convenient for graphical representation, which enables us to obtain a general idea of the magnitude of correction factors for power output of Diesel engines over a wide range of conditions. The value of $1 + \frac{c_t}{b_t}$ can be expressed in terms of fuel : air ratio and known parameters by utilizing Equations [5] and [16], which gives

$$1 + \frac{c_t}{b_t} = \frac{1}{1 - \dfrac{c_t}{(e_0 - af_t)Qv_t\rho_tf_t}} \ldots\ldots [28]$$

[4] This represents a commonly used engineering standard condition. The A.S.M.E. Test Code for Internal Combustion Engines, September, 1943, refers gas volumes to a standard condition of 29.92 in. of mercury and 68 F. The maximum difference between horsepower corrected to either condition is less than 0.9 per cent.

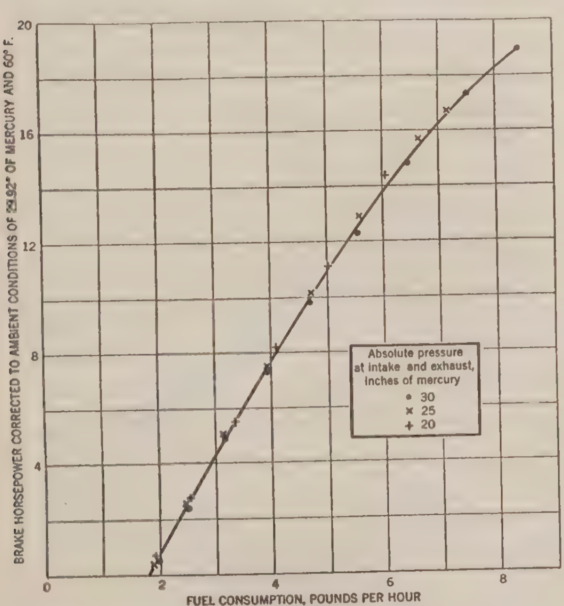

Fig. 18 Relation Between Corrected Brake Horsepower and Fuel Consumption for Engine A at 600 Rpm

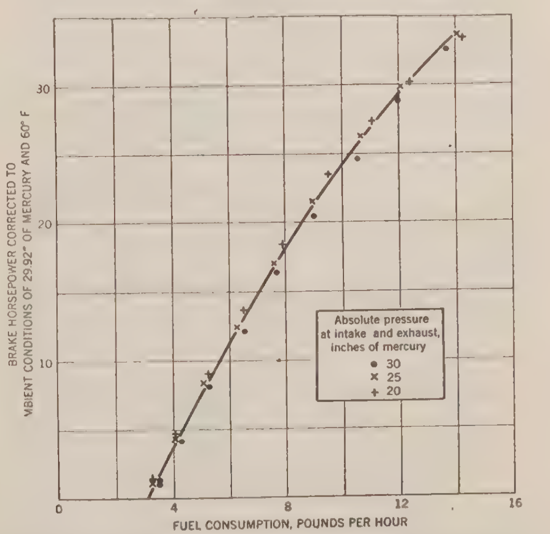

Fig. 19 Relation Between Corrected Brake Horsepower and Fuel Consumption for Engine A at 1000 Rpm

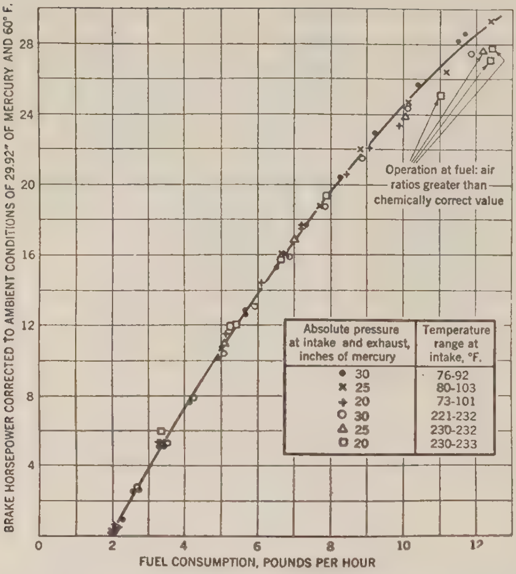

FIG. 20 RELATION BETWEEN CORRECTED BRAKE HORSEPOWER AND FUEL CONSUMPTION FOR ENGINE B AT 1000 RPM

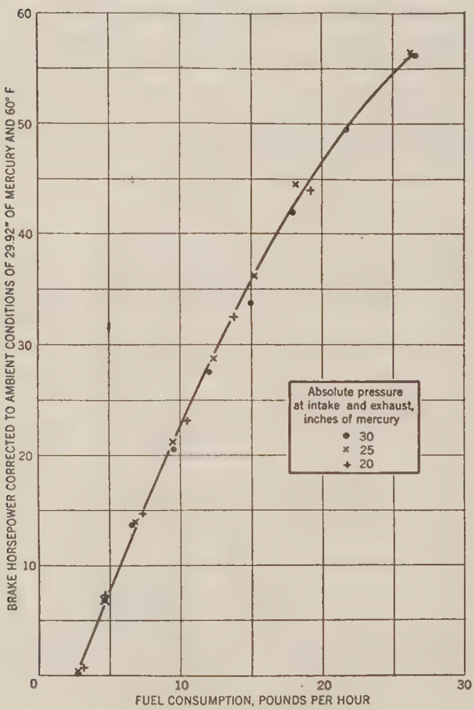

FIG. 21 RELATION BETWEEN CORRECTED BRAKE HORSEPOWER AND FUEL CONSUMPTION FOR ENGINE C AT 600 RPM

Substituting in Equation [23] the value of $1 + \frac{c_t}{b_t}$ from Equation [28], we obtain upon simplification

$$\frac{b_s}{b_t} \doteq 1 + \frac{1 - \frac{\rho_t}{\rho_s}}{\frac{e_0}{af_t} - \frac{c_t}{Qaf_t^2 v_t \rho_t} - 1} \qquad [29]$$

Equation [29] shows that $\frac{b_s}{b_t}$ can be expressed in terms of $\frac{\rho_t}{\rho_s}$ and f_t when the values of e_0, a, c_t, Q, and v_t are known. If we use the data for engine B and assume $e_0 = 3.98$, $a = 20.6$; $c_t = 7.2$; $Q = 3930$; and $v_t = 0.85$ (range of values of v_t for all test conditions at normal temperatures was 0.88 to 0.82) we obtain

$$\frac{b_s}{b_t} = 1 + \frac{1 - \frac{\rho_t}{\rho_s}}{\frac{0.194}{f_t} - \frac{0.000105}{f_t^2 \rho_t} - 1} \qquad [30]$$

Equation [30] is shown graphically in Fig. 22 for an assumed ρ_s of 0.0765 lb per cu ft (corresponding to an absolute pressure of 29.92 in. of mercury and a temperature of 60 F). From this figure we may obtain an excellent idea of the approximate magnitude of the correction factor for brake horsepower throughout a wide range of possible operating conditions. It is emphasized that Fig. 22 applies only to one engine and one engine speed, and until further information is developed each engine and engine speed should be considered as an individual case in preparing charts similar to Fig. 22. However, from the data available it appears that correction factors for any engine and engine speed will be within a few per cent of the factors shown in Fig. 22, except at low fuel : air ratios. The reason for this is that the values of $\frac{e_0}{a}$ and a do not vary greatly with type of engine and engine speed. Furthermore the value of $\frac{c_t}{Q}$ is not affected greatly by engine speed within the range studied because both c_t and Q are a function

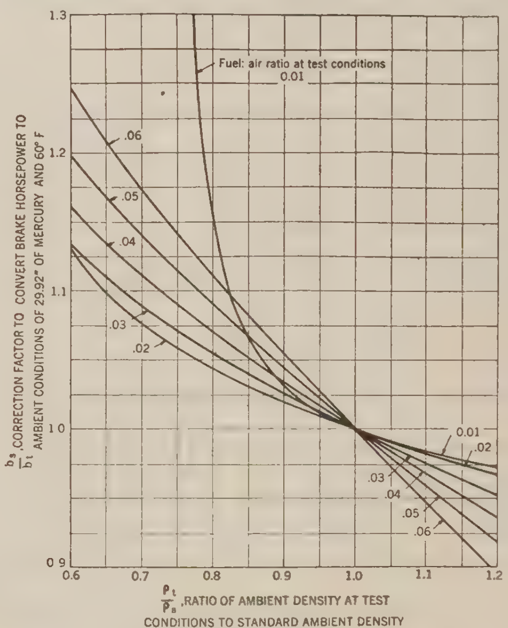

FIG. 22 RELATION BETWEEN $\frac{b_s}{b_t}$, $\frac{\rho_s}{\rho_t}$, AND f_t FOR ENGINE B AT 1000 RPM

(For assumed standard ambient conditions of 29.92 in. of mercury and 60 F.)

of engine speed. The value of $\frac{c_t}{Q}$ is affected by type of engine; however, a study of Equation [29] shows that comparatively large variations in $\frac{c_t}{Q}$ have relatively little effect on $\frac{b_s}{b_t}$ at the higher fuel : air ratios.

The most desirable form of the correction equation can be determined only when additional data are available. If further tests show that $\frac{e_0}{a}$ is approximately constant for a number of engines then it may be possible to use Equation [23] with an average value of $\frac{e_0}{a}$ for all engines except when extreme precision is required. The value of a may not vary widely from engine to engine, in which case Equation [25] might be more desirable.

The conditions under which correction factors may have a significant effect on measurements of power output of Diesel engines will depend principally upon the precision of the measurement of power output, on the ratio of the ambient density during the test to the standard ambient density, and on the fuel:air ratio at which the engine is operated during the test. In this connection it is interesting to note that the A.S.M.E. Test Code for Internal-Combustion Engines, September, 1943, permits a maximum deviation from specified operating conditions of ±3 per cent of the absolute barometric pressure specified and of ±5 per cent of the equivalent absolute temperature specified. Correction factors for power output are not required within these limits. With these permissible deviations it would be possible for ambient density to deviate ±8 per cent from the specified value. From Equation [30] it is evident that this deviation in ambient density would cause a deviation in specified brake horsepower of ±3.5 per cent in the case of engine B operated at a fuel:air ratio of 0.05 lb per lb. It appears, therefore, that in some instances consideration might be given to the application of correction factors for power output even though the test ambient conditions are within the permissible limits specified by the A.S.M.E. Test Code.

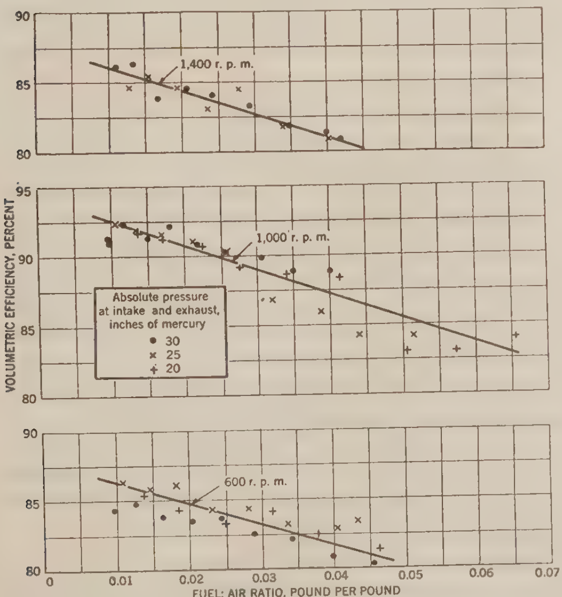

Fig. 23 Relation Between Volumetric Efficiency and Fuel: Air Ratio in Tests of Engine A

Effect of Ambient Conditions on Volumetric Efficiency

In deriving Equation [23], one of the simplifying assumptions was that volumetric efficiency was not affected significantly by changes in ambient conditions. It is of interest therefore to present the evidence supporting this assumption.

Figs. 23, 24, and 25 show the relation between volumetric efficiency and fuel:air ratio. It will be observed that volumetric efficiency decreases slightly as fuel : air ratio increases. The relation is apparently independent of ambient pressure within the range studied, as the deviations of volumetric efficiencies at different ambient pressures are generally within the limits of experimental error (approximately 2 to 3 per cent).

Fig. 24 shows a significant increase in volumetric efficiency as the temperature is increased. This apparent effect is a result of the method of heating the air and probably would not be observed if other more suitable arrangements were provided for heating the intake air. In the tests at elevated temperatures the intake air was heated electrically just before it entered the intake manifold. This method was used to avoid insulation of the intake surge tank. The addition of heat to the intake air results in an expansion in volume and therefore the work done during this expansion, which occurs at substantially constant pressure, must be considered. Under the test conditions a relatively large mass of intake air was on the upstream side of the air heater and a relatively small mass was on the downstream side. It would appear therefore that most of the work resulting from expansion during

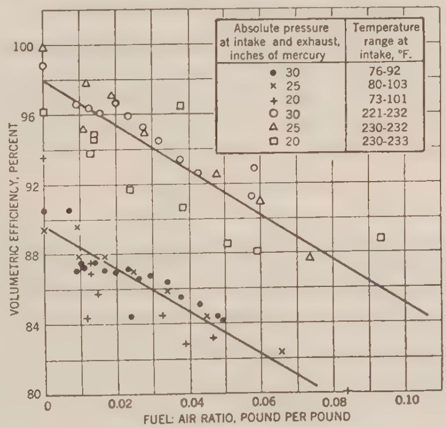

Fig. 24 Relation Between Volumetric Efficiency and Fuel: Air Ratio in Tests of Engine B at 1000 Rpm

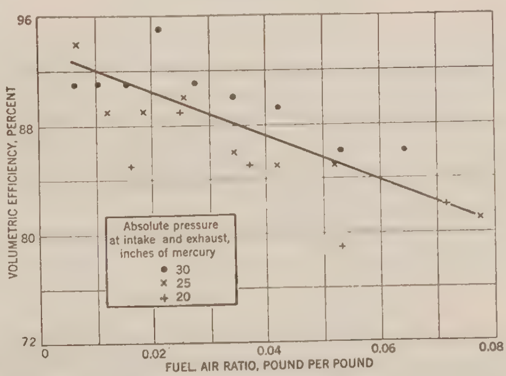

Fig. 25 Relation Between Volumetric Efficiency and Fuel: Air Ratio in Tests of Engine C at 600 Rpm

heating would be done on the relatively low mass of air downstream from the heater. It might be expected therefore that volumetric efficiency would be improved. It is believed that the foregoing explanation of the apparent effect of temperature is correct but additional experimental evidence will be required before it can be stated definitely that ambient temperature has no significant effect on volumetric efficiency.

In a Diesel engine operated at constant speed, changes in ambient conditions result in changes in fuel : air ratio if the throttle setting (rate of fuel consumption) is maintained constant. Accordingly, volumetric efficiency will be affected slightly by ambient conditions. If it is assumed that the ratio of ambient density during the tests to standard ambient density is as low as 0.6 the volumetric efficiencies under the two conditions do not differ by more than 3 per cent in any of the engines tested. If this difference were taken into consideration in Equation [23] the correction factor would be approximately 1 per cent greater at the high rates of fuel consumption.

If it is desired, volumetric efficiency can be related to fuel : air ratio by an equation of the following form

$$v = v_0 - df \quad [31]$$

in which v_0 and d are constant for a given engine and engine speed. This equation could then be used to determine the ratio of $\frac{v_s}{v_t}$. However, for the ordinary range of ambient conditions encountered it is doubtful whether this refinement is necessary.

Predicting Engine Performance by Algebraic Relations

The equations developed in preceding sections may be used to predict engine performance at any given ambient condition and rate of fuel consumption. From Equations [5] and [16] we obtain

$$b = \left(e_0 - \frac{aF}{Qv\rho}\right) F - c \quad [32]$$

In applying Equation [32] an average value of v may be assumed without introducing a significant error. If desired, v can be eliminated from Equation [32] by the use of Equation [31] but the resulting relation is much more inconvenient to handle.

For purposes of illustration, Equation [32] will be applied to the results of tests of engine B at 1000 rpm. The values of the parameters are as follows: $e_0 = 3.98$; $a = 20.6$; $Q = 3930$; $v = 0.85$ (range 0.82 to 0.88); $c = 7.2$; and Equation [32] becomes

$$b = 3.98\,F - 0.00640\,\frac{F^2}{\rho} - 7.2 \quad [33]$$

A comparison of observed values of brake horsepower and values calculated from Equation [33] is given in Table 6. This comparison shows that the equation can be used to predict the brake horsepower of engine B in its normal operating range with a maximum error of 0.7 hp. The error is greater when the engine is operated on the rich side (see Table 6, test 91) because under these conditions a linear relation between indicated efficiency and fuel : air ratio no longer holds.

If desired, Equation [33] can be represented graphically to show the relation between b, F, and ρ.

TABLE 6 COMPARISON OF BRAKE HORSEPOWER OF ENGINE B, CALCULATED FROM EQUATION [33], WITH OBSERVED BRAKE HORSEPOWER

Nominal ambient pressure, in. Hg, abs: 30			25			20		
Test no.	Calc. bhp	Obs. bhp	Test no.	Calc. bhp	Obs. bhp	Test no.	Calc. bhp	Obs. bhp
47	0.4	0.3	97	0.2	0.2	87	0.4	0.4
57	5.3	5.1	76	5.0	5.1	88	4.4	5.0
58	10.2	10.1	77	9.9	10.1	89	9.6	10.2
59	14.9	15.0	78	14.7	15.0	90	14.4	15.1
54	19.5	20.0	79	19.4	20.1	119	19.2	18.8
63	26.9	27.5	80	25.5	25.2	91	21.5	19.6

Summary and Conclusions

This report presents the results of performance tests on three different commercial Diesel engines at different ambient conditions. Operation at different ambient conditions was simulated by controlling conditions in surge tanks connected to the intake and exhaust of the engines. The tests were limited in scope because the principal interest in the investigation was the effect of ambient conditions on composition of the exhaust gases produced by the engine.

The performance data were analyzed, and a method was developed for deriving a relation between so-called derived indicated efficiency and fuel:air ratio that is independent of ambient conditions. The existence of such a relation would be predicted from theoretical thermodynamic considerations. The report presents a method for deriving the relation between indicated efficiency and fuel:air ratio from determination of brake horsepower, fuel consumption and fuel:air ratio at different loads throughout the operating range at a given engine speed.

The relation between derived indicated efficiency and fuel:air ratio is used as the basis for developing rational procedures for correcting to standard ambient conditions brake-horsepower determinations made at ambient conditions prevailing during the tests. The methods developed were applied to the results of tests at different ambient conditions and it was shown that the corrected brake horsepower at a given rate of fuel consumption was independent of the ambient conditions during the tests. Thus the methods developed appear to be suitable for determination of correction factors at least for the three engines on which data were obtained.

The relation between derived indicated efficiency and fuel:air ratio may also be used as the basis for algebraic expressions which permit determination of the brake horsepower of an engine at any rate of fuel consumption and at any ambient density when all the parameters in the equation are known. Brake horsepowers computed from the derived equation are in very good agreement with brake horsepowers determined in tests at a wide range of ambient conditions.

The data and analysis presented in this report have shown that rational procedures can be developed for correcting brake-horsepower determinations to standard ambient conditions. However, only three engines were tested and these engines were not studied as intensively as would be required in the development of generally applicable procedures for determining corrected brake horsepower. Further work may show that the parameters in the correction equation are not significantly different for different engines, in which case a single equation may be used for all engines. If this is not true, the correction equation will differ for different engines. There are indications also that these parameters may be affected by engine speed.

In an extensive study of methods for determining corrected brake horsepower it would be highly desirable to determine indicated horsepower directly and from this to determine whether ambient conditions have a significant effect on the relation between indicated efficiency and fuel:air ratio. If this were done it would not be necessary to derive an empirical relation from data on brake horsepower, fuel consumption, and fuel : air ratio as outlined in this paper.

Direct determination of indicated horsepower would also permit determination of the effect of ambient conditions on friction horsepower and of the effect of engine load on friction horsepower. If this fundamental information were available, more precise

methods could be developed for determining corrected brake horsepower.

Surge tanks connected to the intake and exhaust were used in the present investigation because of their simplicity and availability. However, in an extended investigation it probably would be advisable to test engines in a chamber in which the desired ambient conditions are maintained.

Acknowledgments

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Water Injection in a Spark-Ignition Engine

By W. P. GREEN[1] and C. A. SHREEVE, JR.[1]

The present war witnessed a revival of interest in the use of water as a coolant and detonation suppressor inside the engine cylinder. Aircraft engines when operated with water injection have been boosted in output above normal sea-level capacity and have functioned satisfactorily. This paper gives a review of past tests on the use of water as a coolant and detonation suppressor in gasoline engines. It presents new data regarding the use of water in very high compression ratio engines to promote increased part-load efficiencies with low octane fuels.

THE introduction of water into the mixture, entering the cylinder of a spark-ignition engine, was tried as early as 1866, and was common on carbureted engines burning kerosene a few decades ago. Its action was not fully understood but its purpose was to prevent detonation and to promote smooth combustion. In 1935 Ford L. Prescott reported tests on supercharged aircraft engines in which as much as 2.83 lb of water were added per lb of fuel. No loss in thermal efficiency and an increase in power output were noted. M. S. Kuhring has reported tests in which water was injected into the mixture at the supercharger, with results similar to those obtained by Prescott. Kuhring noted cooling effects on the spark plugs, cylinder heads, and valves when the water was injected.

Colwell, Cummings, and Anderson have presented results of recent work on the injection of water, and water-alcohol into the mixture of truck engines. Data are given to show no increase in wear or other harmful effects on engine life as a result of using water in the combustion chamber.

Steinitz has pointed out that the use of water with its attendant equipment complicates the engine and increases weight. Also there is the possibility of the water freezing unless mixed with an antifreeze solution.

Past research has established the following facts in regard to use of water injection as a detonation suppressor:

1 There is no loss of thermal efficiency with low rates of water injection.

2 Cooling of the engine is not so important a problem at high outputs.

3 Supercharged engines will give higher outputs due to cooling of the mixture by the injected water. The higher manifold pressures increase maximum cylinder pressures and engine output. Detonation is suppressed by the water.

4 Space and weight requirements for carrying the water may be excessive unless condensing equipment is used to secure water from the exhaust gases.

5 There is danger of the water freezing unless mixed with an antifreeze solution.

Purpose of Present Tests

In the light of previously published information the authors felt that tests conducted over a wide range of compression ratios and rates of water injection would accomplish the following:

1 Add valuable data as to quantity of water needed to prevent detonation at a number of load conditions and compression ratios.

2 Determine whether worthwhile gains could be made by operating an overcompressed engine without water at low loads, with detonation suppressed in the high load range by water injection.

3 Indicate effect of injected water on such items as exhaust temperature and heat loss to cooling water.

Test Equipment. A single-cylinder water-cooled N.A.C.A. Universal test engine, located in the mechanical-engineering laboratories of the University of Maryland, was used for all tests, Fig. 1. This engine has a 5-in. bore and 7-in. stroke. Its

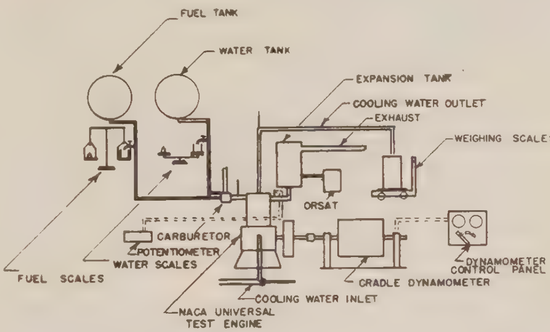

Fig. 1 Schematic Layout of Test Equipment

compression ratio may be continuously varied from 4 to 1 to as high as 14 to 1. The engine is equipped with a valve-in-head combustion chamber, with two intake and two exhaust valves. A magneto ignition system was used with a single spark plug located at the top of the combustion chamber. Engine output and friction horsepower were measured with a Sprague-type electric dynamometer.

Water was fed from a $^1/_2$-gal beaker, mounted on a weighing scale, to a small jet located in the carburetor venturi. Flow of water was controlled manually.

A standard fuel-weighing set-up was used to measure fuel-consumption rates. Mixture temperatures were measured with a thermometer inserted in the intake manifold 10 in. from the intake-valve port. Mixtures were checked during each test with an Orsat apparatus. Samples of exhaust gas analyzed were taken from an expansion tank located 4 ft from the engine-exhaust-valve port. Exhaust-gas temperatures were measured by an unshielded thermocouple located in the engine exhaust and read by a potentiometer. Exhaust temperatures were corrected for radiation and recovery losses as suggested by King (1)[2] and Hottel (2).

Cooling water from laboratory mains entered a recirculating system actuated by a pump on the engine. Its temperature was measured at the inlet to the recirculating system. After passing through the engine cylinder and head to a thermometer well at the outlet, the water was caught in a tank and weighed. An RCA piezoelectric indicator was used to take cards and to check detonation visually.

[1] Associate Professor of Mechanical Engineering, University of Maryland, College Park, Md. Mem. A.S.M.E.

Contributed by the Oil and Gas Power Division and presented at the Fall Meeting, Cincinnati, Ohio, October 2–3, 1945, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Test Procedure. It was decided to operate the engine on one type of fuel at four different throttle openings, thus varying the output between 30 per cent load and full load. At each throttle position tests were run at a number of compression ratios. The first run at each throttle opening was made at a low enough compression ratio so that no detonation occurred under normal operating conditions. No water was injected on this run. Succeeding runs were made at higher compression ratios with just enough water being injected to suppress detonation. Detonation was detected by ear and by the oscillograph on some tests.

Table 1 shows items which were maintained constant on all runs.

TABLE 1 ITEMS WHICH WERE CONSTANT DURING TEST RUNS

Fuel used	gasoline
Octane no. of fuel	68
Revolutions per minute	1000
Spark advance, deg Btdc	25
Air-fuel ratio (approx)	14:1
Cooling-water temperature in cylinder head and cylinder wall, deg F	180 ± 5
Oil temperature, deg F	125 ± 5
Oil pressure, psi	42
Exhaust pressure	atmospheric
Inlet air, deg F	60 ± 5
Mixture temperature, deg F	65 ± 3
Barometer reading, in. Hg	30 ± 0.10

The engine was operated for approximately $2^1/_2$ hrs, until readings indicated that all operating conditions were constant. Several test runs of 15 minutes were conducted at each compression ratio. Whenever the compression ratio was changed at least 30 minutes were allowed for conditions to stabilize before new test readings were taken.

Primary test data taken in addition to readings of standard conditions were as follows:

Compression ratio	Dynamometer load
Duration of run	Cooling-water temperature at inlet
Pounds of fuel consumed	Cooling-water temperature at outlet
Pounds of injection water	Pounds of cooling water
Average rpm	Exhaust temperature
Orsat analysis	

Results. Results computed from the data are plotted in Figs. 2, 3, 4, 5, and 6. Each figure shows the engine performance at one throttle opening for various compression ratios and throttle openings.

It is apparent from a study of these figures that as the load is increased, water injection must start at lower compression ratios in order to prevent detonation. If, however, effective compression ratios are considered, it is necessary to start water injection at compression ratios between 4.2 and 5.1 to 1 for all tests on this engine.

Indicated horsepower, brake horsepower, and thermal efficiency show marked gains with increase in compression ratio, especially in the case of throttle opening No. 1. At compression ratios larger than 11:1 there is no appreciable gain in efficiency and power due to the action of the injected water.

Exhaust temperatures and heat to cooling water decrease rapidly with increased compression ratios and water rates.

Fig. 6 shows a typical mechanical-efficiency curve for this engine. It was found that the variations in throttle opening and compression ratio had little effect on this curve. All points fell within ± 2 per cent of the curve.

Quantities and Effects of Injection Water

Fig. 7 shows a family of curves indicating the pounds of water required per pound of fuel to prevent detonation for various horsepower and compression ratios on this engine. It will be noted that higher outputs and compression ratios required higher rates of water injection. In connection with these curves, it should be noted that in the high-load and compression-ratio ranges the engine did not detonate although it ran roughly and occasionally misfired. It is possible that misfiring could have been prevented by use of other types of spark plugs, although this was not investigated.

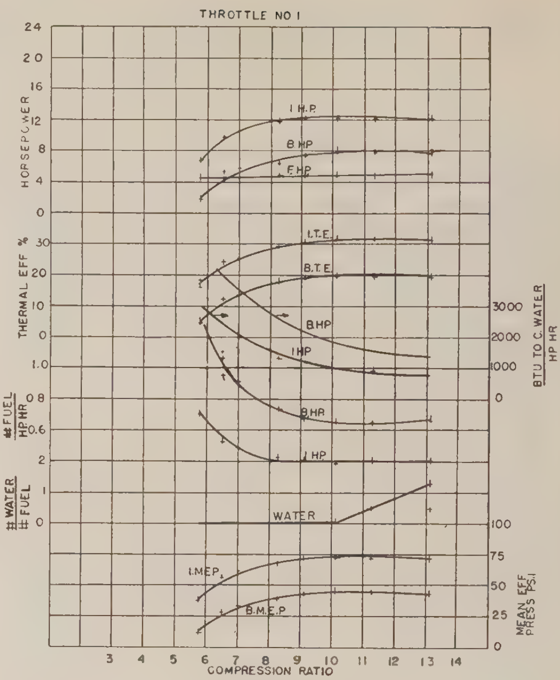

Fig. 2

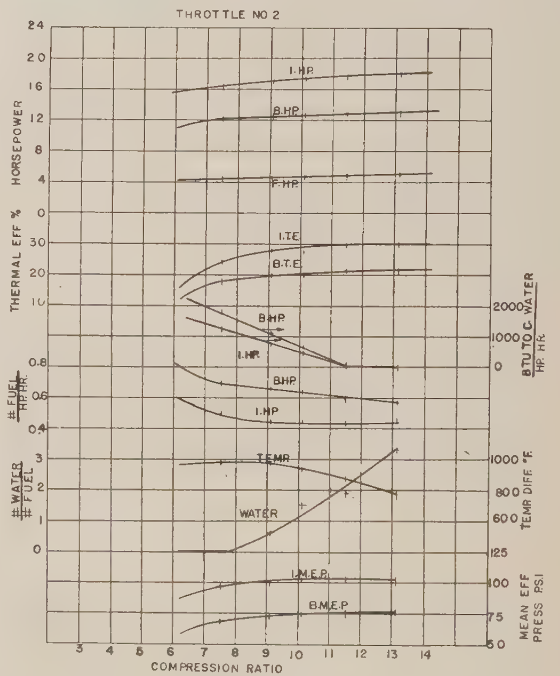

Fig. 3

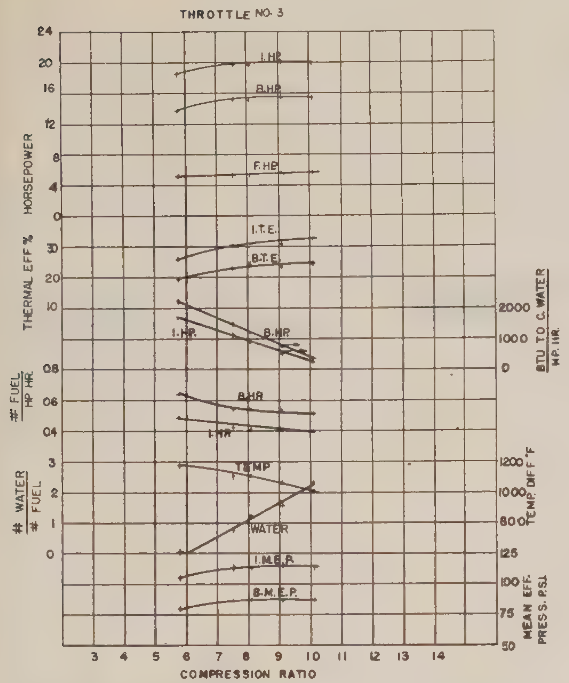

FIG. 4

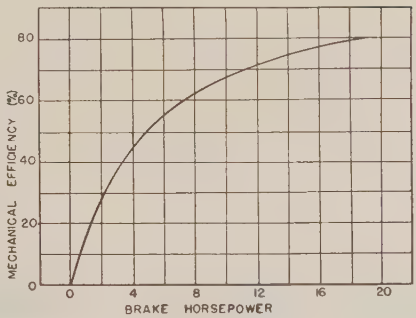

FIG. 6

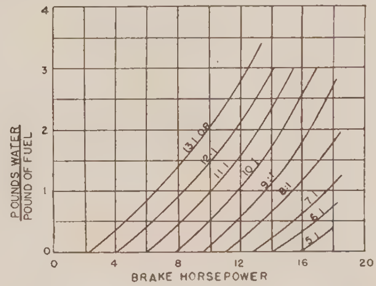

FIG. 7

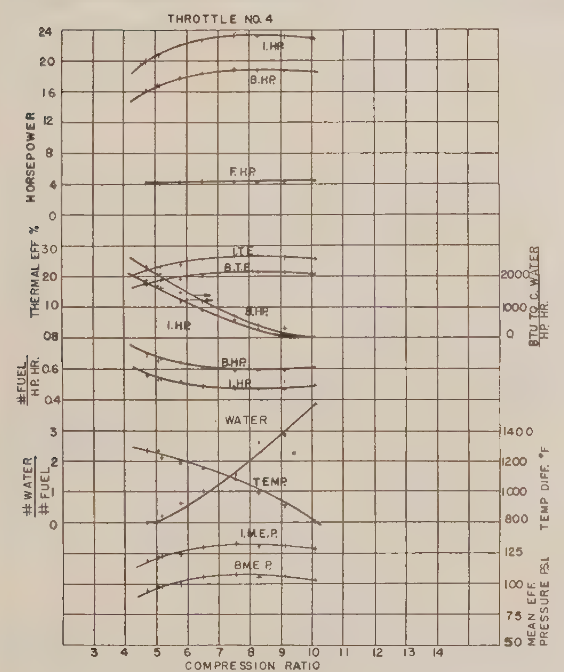

FIG. 5

Fig. 8 shows a set of typical pressure-time-indicator cards for this engine at full throttle and with various compression ratios, using a slightly different gasoline from that of the preceding tests.

At the 4.8:1 compression ratio it was not necessary to add water to prevent detonation. At 5.2:1 compression ratio slight detonation was evidenced by the sharp peak at the top of the card. This was remedied on the next card by the addition of just enough water to prevent detonation. It will be noted that the first effect of the water was to lower the maximum pressure.

As the compression ratio was increased it was necessary to increase the water-injection rates to prevent detonation. It will be noted that the effect of high rates of water injection was to decrease the slope of the expansion line to a marked degree.

The 9:1 compression-ratio card has an excess of water and definitely shows the effect of the water vapor on the expansion line.

Fig. 9 shows the effect of the addition of water on the cooling loss (Btu/bhphr). This curve indicates that the heat loss to cooling water is largely dependent on the water-injection rate. It also shows that for higher rates of water injection the cooling problem is reduced to a point where it is possible that no external cooling other than that done by radiation and the lubricating oil will be required.

Fig. 10 shows the effect of brake horsepower and compression ratio on brake specific fuel consumption. Water was added where necessary to prevent detonation in the amounts indicated in Fig. 7. At the lower brake horsepowers there is a marked difference between the specific fuel consumption at the various compression ratios, while at the higher outputs the water required to suppress detonation lessens the gains due to higher compression ratios.

Fig. 11 shows a comparison between the fuel required for the various compression ratios in per cent of fuel required for 6:1 compression ratio. At the lower loads the 11:1 compression ratio will save about 50 per cent of the fuel required at the 6:1 compression ratio. As the load increases the fuel economy due to the higher compression ratio decreases until at full load the amount of injected water may actually make the higher ratio slightly less economical.

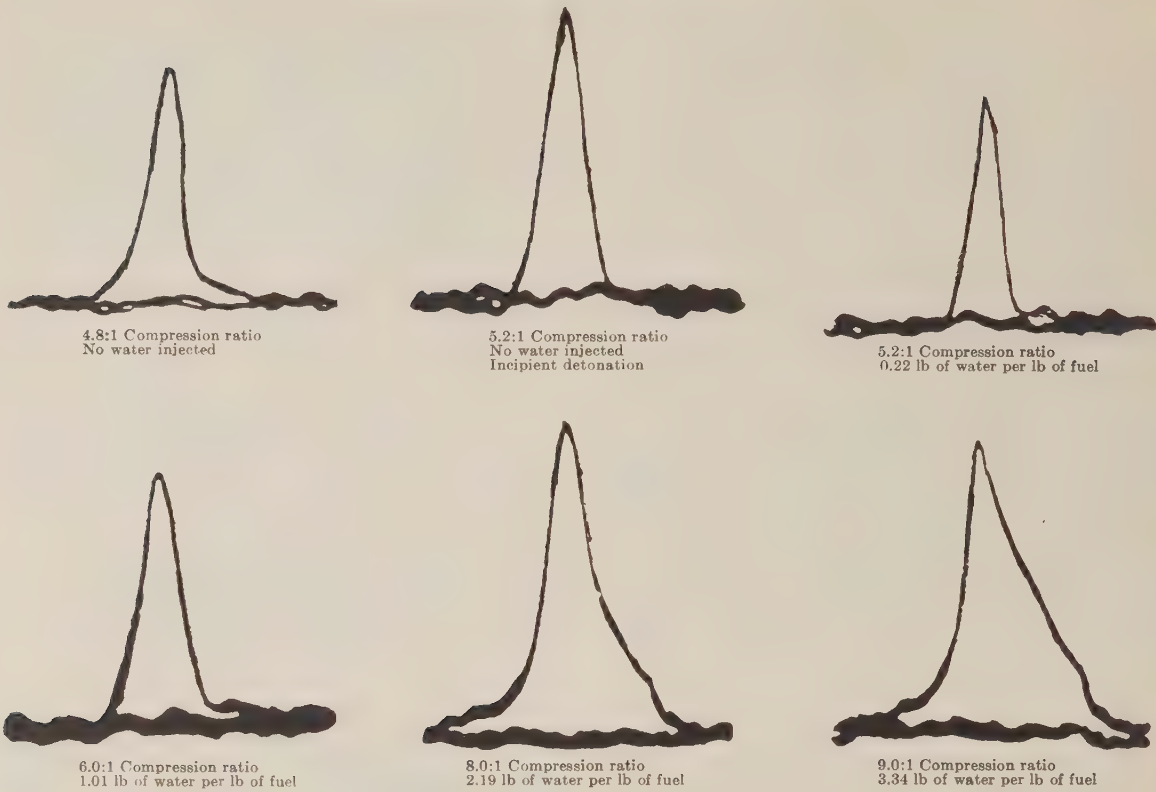

Fig. 8 Indicator Cards of N.A.C.A. 5-In. × 7-In. Engine, 1000 Rpm, Tested at the Mechanical Engineering Laboratory, University of Maryland

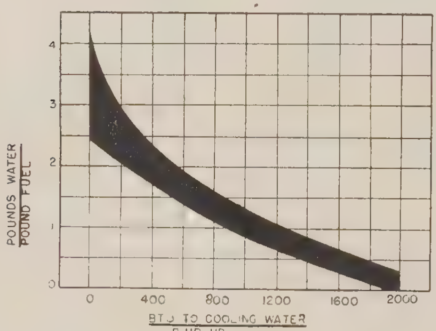

Fig. 9

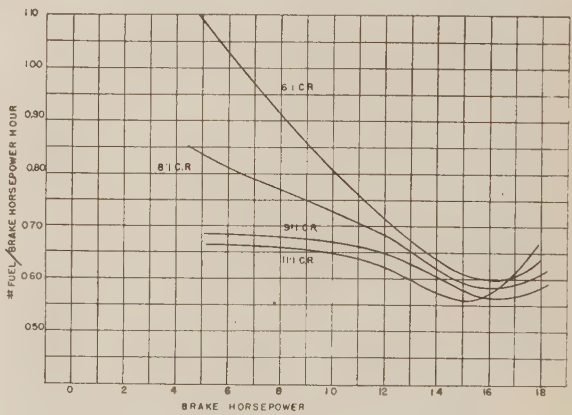

Fig. 10

Conclusions

The injection of water into the engine cylinder decreases the maximum pressure and tends to decrease the slope of the expansion line with little or no change in thermal efficiency.

Savings shown by the curves in Fig. 11 indicate the desirability of the use of water injection in engines that will operate the greater portion of the time in the low-load range, with only occasional use at full power.

Exhaust temperatures are reduced by water injection although a portion (or all) of the normal heat transferred to the cooling water now appears in the exhaust.

Acknowledgment

The authors wish to acknowledge the assistance of A. E. Seigel and N. C. Eckhardt, senior mechanical engineering students, in setting up equipment and running preliminary tests.

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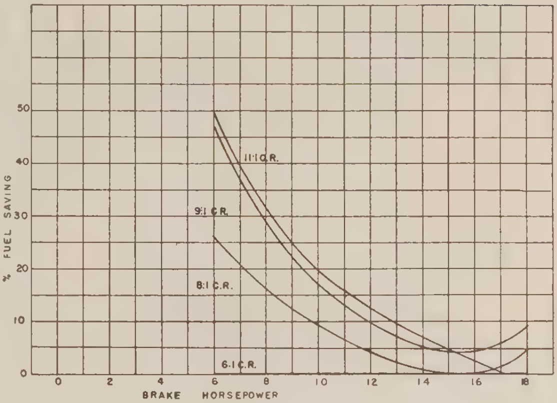

Fig. 11

(Owing to travel emergency conditions existing when this paper was presented, discussion will be accepted until August 16, 1946)

Relation Between the Impact and Flexural Tests for Molded Plastics

By L. E. WELCH[1,2] and H. M. QUACKENBOS, JR.[1]

A theoretical treatment of the phenomena of impact is presented which indicates that rupture under dynamic stress-strain conditions results from the same basic considerations as those governing static or flexural failure. This thesis is proved experimentally by reducing the problem to one of stress rather than the usual method of energy evaluation. When this is done with the aid of newly developed high-speed electronic techniques, it is conclusively demonstrated that impact loading is merely a special case of static loading whereby the stress-strain conditions are identical except for a slight elevation in strength under dynamic conditions. It is shown that the existing methods of impact-strength evaluation such as the standard Izod and Charpy tests lead to erroneous results since the machines in use are apparently of too light a construction, and the technique is oversimplified. It is suggested that the static flexural-test technique when properly carried out to evaluate the energy required for fracture and the notch sensitivity, is sufficient to predict the service characteristics of essentially brittle materials under conditions of both static and impact loading.

Nomenclature

The following nomenclature is used in the paper:

A, B = constants in Equation [16]
b = width of specimen
E = modulus of elasticity of specimen
g = gravitational acceleration
h = depth of specimen (taken outside notched area where bar was notched)
I = moment of cross section of specimen
K, K_M, K_S = spring constant of system, machine and specimen, respectively
l = distance from plane of loading to center of gage for cantilever
L = span of specimen, or length in Equation [4]
M = mass of impacting element
P = force exerted in either static or dynamic loading
t = time after impact
t_1 = total time duration of nonfracture impact
u = mass per unit length of specimen
U = strain energy in specimen under flexural or impact loading
U_0 = kinetic energy of falling (impacting) element
U_S, U_M = energy stored in specimen spring and machine spring, respectively
w = frequency of natural vibration of specimen
v_0 = velocity of impacting element at moment of impact
ϵ = maximum strain in specimen
σ = maximum stress in specimen
y = deflection of specimen in bending test
y_M, y_S = deflection of machine spring and specimen spring, respectively, at any time during impact
y_M', y'_S = maximum values of y_M and y_S during impact

Introduction

Three tests have been generally accepted for evaluating the mechanical properties of rigid plastics. These are the tests for tensile, flexural, and impact properties. Of these the Izod and Charpy impact tests on a notched bar are the most valuable because they differentiate between materials that show similar tensile and flexural results. For example, with the phenolic-resin molding materials with which this paper is largely concerned, the flexural strength usually falls in the range 7000 to 15,000 psi, while the tensile strength displays a smaller variation. The type of filler (wood flour, fabric, mineral, etc.) is more responsible than the type of resin for these changes. The Izod impact strength, however, may vary from 0.15 ft-lb per notched specimen for the wood flour filled to 2.5 ft-lb and higher for the fabric filled. Moreover, the impact evaluation is usually a fair forecast of the toughness of a molded object, especially where internal corners exist. But there are many instances where the agreement between impact tests and practical experience breaks down completely, and so an examination of the Charpy and Izod tests for rigid plastics was undertaken.

A brief description of the flexural and impact tests can well precede a review of the literature. In the standard flexural test a molded specimen, $5 \times {}^1/_2 \times {}^1/_2$ in., is loaded on a 4-in. span. The load-deflection curve is often linear to failure and if ductility is exhibited, it is small. The usual modulus of elasticity (hereinafter referred to as simply modulus) for phenolic molding materials ranges from $1\text{–}4 \times 10^6$ psi. The Charpy and Izod machines are similar to those used for testing metals, except that they are of much lighter construction to accommodate the relatively brittle plastic materials. The standard Izod machine weighs approximately 120 lb. In each test the standard specimen is ${}^1/_2 \times {}^1/_2$ in. with a milled notch 0.100 in. deep, and with a radius of 0.010 in. at the bottom. The Izod specimen is a cantilever $1^1/_4$ in. long struck 0.866 in. above the point of clamping. The Charpy specimen is supported on a 4-in. span and is struck on the side opposite the centrally disposed notch.

The impact strength, that is, the energy loss of the pendulum, is usually considered to be composed of three main parts:

1 Actual strain energy to break the specimen.

2 Machine losses; vibration and damping in the base of the machine.

3 Energy lost by the pendulum in projecting the broken end (Izod) or ends (Charpy) of the specimen.

Item 3 has been the subject of several recent papers (1, 2, 3, 4, 5).[3] The "end effect" was eliminated either by lowering the

[1] Research and Development Laboratories, Bakelite Corporation, Bloomfield, N. J.

[2] East Orange, N. J.

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

pendulum in the orthodox tests until it possessed just enough energy to fracture the specimen, or by using a drop-weight machine in which the specimen is struck blows of increasing severity until fracture is caused. The energy associated with the end effect varies considerably with the filler. It is a small or negligible proportion of the measured impact strength with the tougher materials having a filler of fiber. With the more brittle wood-flour type the proportion may be 60 per cent and as high as 85 per cent with a high-density lead-filled composition. Consequently, quite a different comparison between the common molding materials is given when the end effect is eliminated.

With the energy of projection disposed of, the residual impact energy is then the sum of items 1 and 2, that is, the strain energy plus the machine losses. Most opinions place the machine losses as nearly zero (2, 3, 6). This leaves the residual impact energy as strain energy, and one could reasonably expect it to agree with the energy in a flexural test. Recent tension impact tests of metals (7, 8) have demonstrated that the same stress-strain behavior often occurs in both dynamic and static loading. Early work (9) on wood in long-span flexure shows the same agreement. A paper by Hazen (10) points out that the static flexural energy is a better criterion of service toughness than the standard Izod impact value. Yet the residual impact energy is about 25 per cent higher than the corresponding flexural energy in the Charpy test and is 3 or 4 times as high in the Izod test. The Izod flexural energy for the cantilever of 0.866 in. was calculated in the orthodox way, from the ratio of the spans, as 0.866/4 in. times the energy at a flexural span of 4 in., with a 32 per cent increase to allow for shear.

These discrepancies between the static and dynamic energies suggest the following possibilities: (*a*) The stress-strain relations under impact loading are different from those under static loading; (*b*) machine losses are appreciable.

To resolve these questions the stress-strain diagram in impact has been determined in the Izod and Charpy machines and compared with that for static loading at the same spans. The strain energy in impact absorbed by the specimen can be calculated from the relation between stress and strain. The difference between this energy and that applied by the pendulum, with the end effect eliminated, is taken as machine losses. As machine losses were appreciable, their variation with span was established in a drop-weight type of impact machine.

As to the stress-strain diagram in impact, the strain was measured with a resistance-wire strain gage cemented to the specimen. The voltage change across the gage on straining was amplified and passed to an oscillograph where the displacement of the oscillograph spot then became a measure of the strain. The measurement of dynamic strain in this fashion is an established technique in a number of laboratories (8, 11, 12). The determination of dynamic stress requires a delicate method (8) that could be avoided here because most of the materials used possess a linear stress-strain relation to failure. The dynamic modulus was calculated from the frequency of natural vibration and assumed equal to the modulus present under impact loading. The dynamic stress was derived as the product of this modulus and the measured strain.

Experimental Procedure

Experiments covered the range of common, phenolic molding materials having a variety of fillers and included work on cast iron and steel. All bars had a rectangular cross section. The molding materials were either molded as individual specimens or cut from molded plates 8 in. × 10 in. The laminated specimens were cut from plates. The cut surfaces were smoothed by grinding. The cast iron was cast as a single plate, 12 × 12 × $^3/_8$ in., and bars were cut therefrom 6 × $^3/_8$ × $^3/_8$ in. and finished by grinding. The single steel specimen used was 6 × $^1/_2$ × $^1/_2$ in. from cold-rolled stock. Some of the plastic specimens having a section $^1/_2$ × $^1/_2$ in. contained standard notches, 0.100 in. deep, 0.010 in. radius, milled in the center of one face. Table 1 summarizes the materials.

TABLE 1 LIST OF MATERIALS USED

Material	Description
BM-021	Phenolic resin + wood flour
BM-120	Phenolic resin + wood flour
BM-262	Phenolic resin + mica
BM-1914	Phenolic resin + kraft paper
BM-3510	Phenolic resin + fabric
BM-6260	Phenolic resin + wood flour + cotton floc
XM-9131	Pure phenolic resin — no filler
BM-13080	Phenolic resin + wood flour + cotton floc
BM-16468	Phenolic resin + fabric
Steel	
Cast iron	

The plastic specimens were conditioned 2 days at 50 C and were kept until tested in the laboratory maintained at 25 C, 50 per cent relative humidity.

The following tests were performed: To remove directional effects the same face (e.g., a top molded or cast face) was always in tension.

1 Flexural test on a span of 4 in., according to A.S.T.M. D790–44T: Load-deflection curves were recorded with a Templin stress-strain recorder which magnified the deflection 40 times. Spans other than 4 in. were used occasionally. The measured deflection includes deformation of the machine parts and indentation into the specimen of the edges for loading and supporting. A load-deflection curve was run at zero span (supporting edges together) to give a correction to the measured deflection. For the common thermosetting materials in the form of a bar of $^1/_2$ × $^1/_2$ in. cross section this correction is about 5 per cent at a span of 4 in.

With deflection corrected, modulus for an unnotched bar was calculated from the slope of the load-deflection curve according to

$$E = \frac{PL^3}{4ybh^3}\left[1 + \frac{3h^2}{L^2}\right] \quad [1]$$

and flexural energy from the area under the load-deflection curve.

Modulus was also measured by loading an unnotched bar having a strain gage cemented to the tension face of the specimen. The gage (Baldwin Southwark type SR-4, A-5, length $^1/_2$ in.) was connected to an SR-4 strain meter in the usual way. The center of the gage was vertically below the loading column. The observed strain for a given load must be multiplied by $L/(L - {}^1/_4)$ to give the maximum strain in the plane of loading. Load was converted to stress by

$$\sigma = 1.5\,\frac{PL}{bh^2} \quad [2]$$

and modulus calculated from the slope of the curve of stress versus maximum strain.

2 Static cantilever test: Modulus was determined with resistance-wire strain gages for a span of 0.866 in. If the center of the gage was distance l from the plane of loading, the stress by which the observed strain was divided to give the modulus was calculated from

$$\sigma = \frac{6Pl}{bh^2} \quad [3]$$

A notch, when present, was located on the tension side with its "vee" in the plane formed by the faces of the gripping members and a "modulus" was calculated as if the notch were absent.

3 Dynamic modulus: The frequency of natural vibration

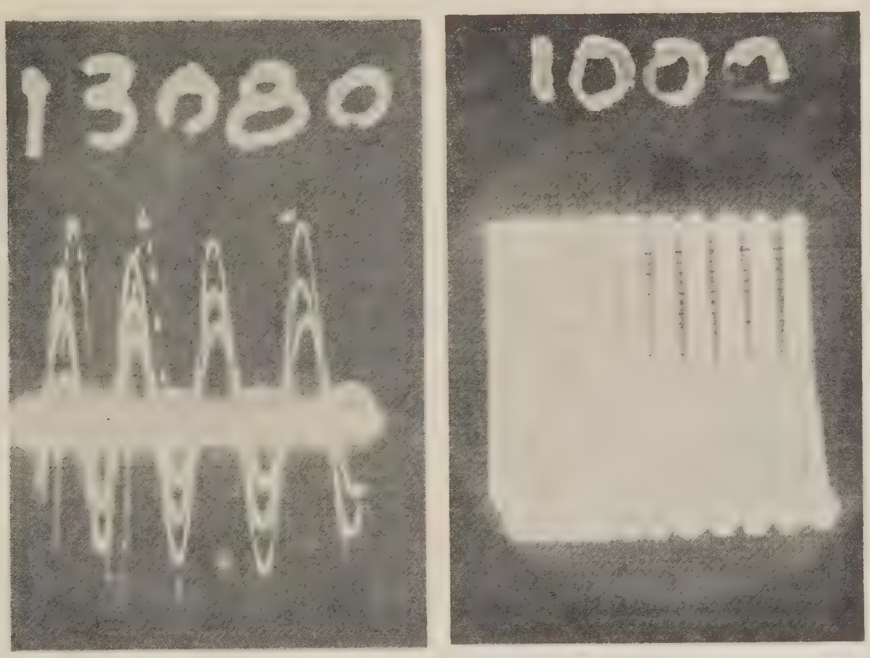

FIG. 1 FREQUENCY OF NATURAL VIBRATION FOR BM-13080, $^1/_2 \times ^1/_4$ IN. AND 1000 CYCLES PER SEC CALIBRATION

was determined mainly for bars having a cross section $^1/_2 \times ^1/_4$ in. An SR-4 A-5 gage was cemented to the central area of the specimen. The gage was connected into the dynamic-strain circuit described in the Appendix. With the oscillograph spot spread into a line, the vibrations of the bar when held lightly at one end and struck at the other cause a wave to appear on the screen of the oscillograph, Fig. 1. This wave is photographed and compared with a wave of 1000 cycles per sec induced by an oscillator. From the natural frequency w, modulus E is (13)

$$w = \frac{22.4}{L^2}\sqrt{\frac{EI}{u}} \qquad [4]$$

4 Measurement of applied impact energy: The Izod and Charpy machines were fitted with movable spring releases to allow the pendulum to be dropped from any desired height. The heights used were such that the impact velocity varied between 2 and 11.4 fps. The specimen was either given a light blow, insufficient to break, or a blow such that the swing-through after fracture was small. The energy applied to the specimen in either case is calculated as in the Appendix. The Izod machine is a standard type manufactured according to the specifications of the Bell Telephone Laboratories. The Charpy instrument was of lighter weight and was made in the bakelite machine shop.

In the drop-weight tester, Fig. 2, a grooved weight released magnetically slides down vertical stainless-steel wires. The base is a steel plate set in a concrete block, $18 \times 12 \times 4$ in. The energy is the product of the height and the weight with an allowance for friction established in a calibration with an electrical timer. The energy of fracture was determined by an increment method. The height was raised 10 per cent each time from an initial height adjusted so that 2 to 4 blows were needed for fracture.

5 Measurement of impact strain and time: The distribution of strain was established with A-8 gages which were good

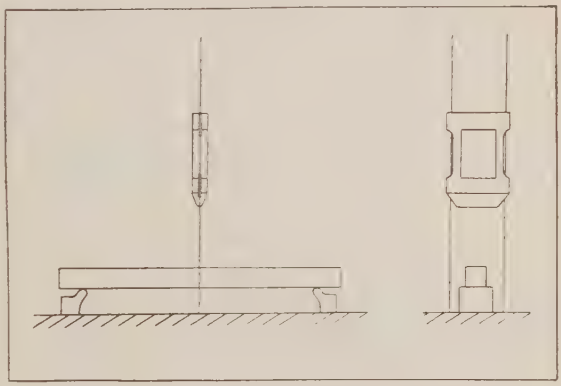

FIG. 2 THE DROP-WEIGHT IMPACT MACHINE, SHOWING WEIGHT SLIDING DOWN WIRES

indicators of essentially point conditions by virtue of their extremely small gage length of $^1/_8$ in. The A-8 gage was also used to detect any compression of the bar against the pendulum and against the supports. Fig. 3 illustrates the disposition of the gages for these experiments.

Once the strain was found to be distributed linearly, subsequent experiments were carried out with A-5 gages which are $^1/_2$ in. long. The maximum strain was calculated from the observed strain at a certain position assuming linear distribution.

With most experiments in which the bar was struck lightly the oscillograph beam of electrons was narrowed to a spot and its displacement was observed by eye as a mean of several blows. Photographs confirmed the precision of this visual method. When photographs were taken the circuit was modified so that the spot was traveling across the face of the oscillograph at a constant rate. The impact then appears as a loop with strain in

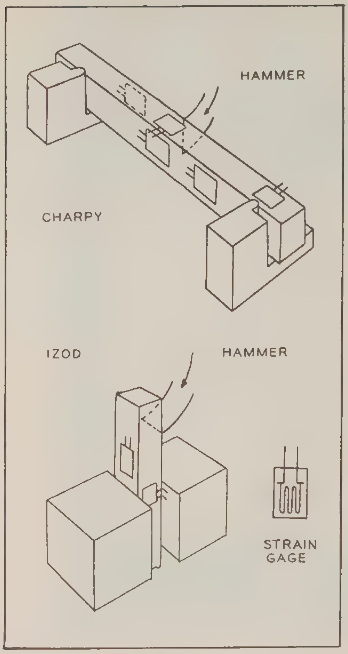

Fig. 3 Disposition of Strain Gages on Charpy and Izod Specimens

vertical units and time in horizontal units. The calibration of the vertical and horizontal units is described in the Appendix. Where the bar was broken photographs were taken with a Contax.

In any one set of tests all bars of one material were from the same batch but different batches were frequently used for different tests. This explains slight variations in modulus from test to test.

Discussion of Results

In analyzing the response of a bar to impact loading it is natural to compare with conditions under static loading. The criteria of comparison are modulus and distribution of strain.

Dynamic Modulus. The dynamic modulus is in most instances the same as the flexural modulus, Table 2. Later when

TABLE 2 DYNAMIC AND STATIC MODULI; 4 IN. SPAN

Material	Flexural modulus psi $\times 10^{-6}$	Dynamic modulus psi $\times 10^{-6}$
BM-120	1.25	1.20
BM-262[a]	4.66	4.55
BM-1914	1.53	1.69
BM-3510	1.47	1.80
BM-6260	1.13	1.14
XM-9131	0.98	1.04
BM-13080	1.15	1.15
BM-16468	1.33	1.46
Steel	29.5	(29.5)
Cast iron	10.5	(10.5)

[a] Specimen, $^1/_2 \times {}^1/_2$ in. All others $^1/_2 \times {}^1/_4$ in.
Note: Dynamic modulus assumed equal to static modulus for steel and cast iron.

calculations were made of impact strain energy in the specimen, the modulus used was the flexural modulus, except for BM-3510, BM-16468, and BM-1914, where the flexural moduli were increased 20 per cent, 10 per cent, and 10 per cent, respectively. Two simplifications are implied when calculations on this basis are made for specimens of various dimensions in the Charpy and Izod tests and for notched bars in the latter. These are as follows:

(*a*) While nearly all the results of Table 2 are for bars having a section $^1/_2 \times {}^1/_4$ in., they are assumed to hold for all dimensions.

(*b*) The static modulus for notched and unnotched cantilever beams (0.866-in. span) is the same as the flexural modulus (4-in. span). Actually the former (notched) tends to be a few per cent lower, and the latter (unnotched) the same degree higher, as established by strain gages.

Deformation Under Static and Dynamic Loading. Fig. 4 illustrates the variation with distance of the dynamic surface strain for a beam tested in the Charpy machine. In general this strain increases from zero at the supports to a maximum at the center, the point of striking, just as it would under similar flexural loading. This was true for several values of the span length and several specimens of different dimensions struck blows insufficient to cause fracture.

There were anomalous effects but these were small enough to be ignored: (*a*) The hammer knife-edge indents into the specimen causing a low compression strain at the point of impact, Fig. 4; (*b*) at the supports and at the striker a local compression wave travels between the surface being hit and the opposite face; (*c*) at the supports the bending strain is not quite zero.

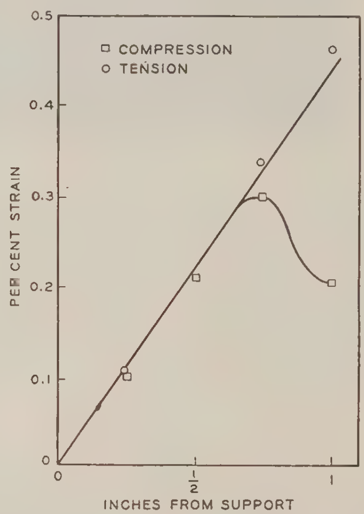

Fig. 4 Impact Strain Versus Distance From Support in Charpy Machine
(BM-021, Span = 2 in., b = 0.5 in., h = 0.5 in.)

Similar anomalies were observed in the unnotched Izod specimen, but otherwise the surface strain increased in linear fashion from the plane of striking to the plane of clamping.

Taken in conjunction with the agreement between the static and the dynamic modulus these findings permit the conclusion that strain, stress, and load build up from zero in the same way in both tests.

Energy Absorbed by the Impact Specimen. In these circumstances the impact strain energy can be computed from the measured impact strain by exactly the same equations that hold for static loading. The strain energy absorbed by a rectangular beam in reaching a maximum strain in flexure ϵ, at the surface, in the plane of loading is given by

$$U = \frac{1}{216} bhLE\,\epsilon^2 \left(1 + 3\frac{h^2}{L^2}\right) \qquad [5]$$

This is a standard textbook formula (14). The factor in parentheses allows for shear energy and is predicated on a value of

TABLE 3 ENERGY ABSORPTION IN CHARPY MACHINE FOR UNNOTCHED BARS STRUCK WITH BLOW INSUFFICIENT TO FRACTURE

Material	b in.	h in.	Span in.	Dynamic modulus, E, psi $\times 10^{-6}$	Measured absorption, per cent	Calculated absorption,[a] per cent
BM-021	0.50	0.50	$2^1/_{16}$	1.25	36	36
BM-262	0.50	0.50	$2^1/_{16}$	4.55	16	13
			$4^1/_8$		81	52
BM-3510	0.50	0.50	$2^1/_{16}$	1.90	34	27
			$4^1/_8$		72	73
Cast iron	0.375	0.375	$2^1/_{16}$	10.5	43	16
			$4^1/_4$		91	58
BM-120, BM-1914, BM-3510, BM-6260, XM-9131, BM-13080, BM-16468	0.50	0.25	$4^1/_4$	See Table 2	102 (mean)	96–97

[a] Calculated absorption based on BM-021 as explained in text.
NOTE: Energy applied varied from 0.1 to 0.22 ft-lb; corresponding velocities of impact being 2 and 3 fps. All bars unnotched.

0.3 for Poisson's ratio, which is true for most plastics (15). The formula for a cantilever is similar (14)

$$U = \frac{1}{216} bhLE\,\epsilon^2 \left(1 + \frac{h^2}{L^2}\right) \quad [6]$$

where ϵ is now the strain at the grip.

A sample calculation for the Izod machine will illustrate the method. Material BM-021, $b = 0.50$ in., $h = 0.50$ in., no notch. Observed strain for gage with center 19/64 in. from grip = 3000 microinches per in. Maximum strain at grip = 3000 × 0.866/(0.866 — 19/64) = 4560 microinches per in.; $E = 1.25 \times 10^6$ psi. Hence energy absorbed by specimen U, from Equation [6] with L = 0.866 in. is 0.034 ft-lb. Energy applied by hammer = 0.131 ft-lb. Per cent measured absorption by specimen = 0.034/0.131 × 100 = 26 per cent. Measured absorptions for the Charpy machine are arrived at in the same way by means of Equation [5].

Measured absorptions calculated by these methods are listed in Tables 3 and 4 for the two impact machines. The significance of calculated absorption will be discussed later.

When the specimen does not break there are, of course, no end effects, and these have been eliminated from the applied energy when it does break. It is seen that generally not all the applied energy is absorbed by the specimen, the remainder presumably being lost to the machine. According to the figures presented, the per cent absorption varies from material to material in both machines and is especially erratic for specimens broken in the Izod test. Further, in the Izod test there is a considerable variation in energy absorbed from one specimen to another, sometimes as high as ±18 per cent. This in spite of the fact that the percentage absorption for one specimen struck lightly varies only ±6 per cent. Changes in clamping pressure seemed responsible to only a small extent.

In view of these findings, the energy lost by the pendulum, corrected for the end effect, has no absolute or comparative significance in either of the impact tests. The energy absorbed gives the correct answer, but even this is unsatisfactory in the Izod test because of its variability. A new test to replace the Izod test is suggested later in this paper.

There are certain disagreements in Table 4, as yet unexplained, between the energy absorption at fracture and that under a light blow. This is so when the filler is fabric, kraft, or floc.

Theory of Energy Absorption. It has been established that the energy absorbed by a specimen is only a part of that applied by the impact hammer. A reasonable assumption is that the remainder is lost by absorption in the impact machine. The system comprising the machine and specimen may then be compared to two elastic springs and the energy absorption should follow a certain pattern. An ideal system is illustrated in Fig. 5

TABLE 4 ENERGY ABSORPTION IN IZOD MACHINE

Material	b	h	Modulus, E psi $\times 10^{-6}$	Measured energy absorption, per cent	Calculated energy absorption,[a] per cent
BLOW INSUFFICIENT TO BREAK[b]					
Unnotched					
BM-021	0.50	0.50	1.25	26	26[a]
BM-13080	0.50	0.50	1.07	23	29
BM-3510	0.50	0.50	1.90	26	19
BM-262	0.50	0.50	4.7	21	9
Steel	0.50	0.50	29.5	12	1.5
Cast iron	0.375	0.375	10.5	36	11
BM-3510, BM-13080, BM-6260	0.49	0.25	Table 2	33–36	62–72
Notched					
BM-1914	0.5	0.5	1.69	21	21
BM-3510	0.5	0.5	1.90	25	19
BM-13080	0.5	0.5	1.15	25	29
BLOW TO BREAK[c]					
Unnotched					
BM-021	0.50	0.50	1.25	25	26
BM-262	0.50	0.50	4.7	17	9
BM-1914	0.50	0.50	1.69	24	21
BM-13080	0.50	0.50	1.07	24	29
BM-3510	0.50	0.50	1.90	12[d]	19
Notched					
BM-021	0.50	0.50	1.25	22	26
BM-1914	0.50	0.50	1.69	12	21
BM-13080	0.50	0.50	1.07	14	29

[a] Calculated absorption based on BM-021 as explained in text.
[b] Applied energy varied from 0.098 to 0.39 ft-lb, and velocity of impact from 2.5 to 5 fps.
[c] Velocity of impact varied from 7 to 11.3 fps, except that it was 3 to 4 fps from BM-262.
[d] This is the only material broken in Table 4 that exhibits any ductility near fracture. A gage outside the region, near the clamp, of high strain will not reflect this slight ductility so that the energy absorption is low.

in which a falling weight (the pendulum) drops onto a pan attached to the springs representing the machine and the specimen. An analysis can be made on the following assumptions: (*a*) No damping in any components; (*b*) no loss of energy from the system, e.g., through the machine spring to the rigid foundation to which it is attached; (*c*) both springs weightless and elastic; (*d*) displacement of the springs during impact is small compared with the height of fall of the weight.

At any time during impact the force P, exerted by the falling weight is the same for each spring, and with spring deflections y_M and y_S and spring constants K_M and K_S is given as

$$P = K_M y_M = K_S y_S = K(y_M + y_S) \quad [7]$$

where K = spring constant for total system.

When the strain in the specimen reaches a maximum all the energy U_0, of the pendulum is given up to the springs, i.e.

$$U_0 = U_S + U_M = {}^1/_2 K_M y_M'^2 + {}^1/_2 K_S y_S'^2 = {}^1/_2 K\,(y_M' + y_S')^2 \quad [8]$$

where the primes denote maximum deflections. This is true for nonfracture, and for fracture where end effects are eliminated. Therefore from Equations [8] and [7]

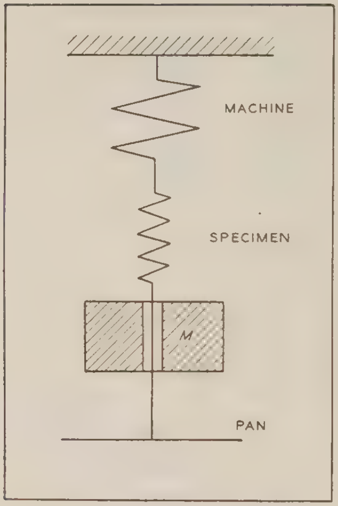

Fig. 5 Ideal Impact System

$$\frac{U_S}{U_0} = \frac{K_S y_S'^2}{K\,[y'_M + y'_S]^2} = \frac{K_S}{K} \cdot \frac{K^2}{K_S^2} = \frac{K}{K_S} \quad [9]$$

Now from Equation [7] also

$$K = \frac{K_S K_M}{K_M + K_S}$$

and on substitution in Equation [9]

$$\frac{U_S}{U_0} = \frac{K_M}{K_M + K_S} \quad [10]$$

Note that U_S/U_0 is the ratio of the energy absorbed by the specimen to that applied by the hammer.

By definition, the spring constant of the specimen K_S is equal to P/y_S and for the beam (Charpy) type of specimen the ratio is given by Equation [1]

$$y_S = \frac{PL^3}{4Ebh^3}\left(1 + \frac{3h^2}{L^2}\right) \quad [1]$$

or

$$K_S = \frac{P}{y_S} = \frac{4Ebh^3}{L^3\left(1 + 3\,\frac{h^2}{L^2}\right)} \quad [11]$$

The formula for K_S for the Izod specimen is similar so that

$$K_S = \frac{Ebh^3}{4L^3\left(1 + \frac{h^2}{L^2}\right)} \quad [12]$$

Now let these equations be applied in practice. For the Charpy machine $U_S/U_0 = 0.36$ for BM-021 when $L = 2^1/_{16}$ in., $b = 0.50$ in., $h = 0.50$ in., $E = 1.25 \times 10^6$. Hence K_S for BM-021 in this instance is 3.01×10^4 lb per in.; K_M can then be calculated from Equation [11] as 1.69×10^4 lb per in.

With these facts it is possible to predict the absorption that will be exhibited by any other bar of any dimensions and modulus and at any span. The procedure is to calculate K_S from Equation [11] and then compute the absorption U_S/U_0, from the relationship.

$$\frac{U_S}{U_0} = \frac{1.69 \times 10^4}{1.69 \times 10^4 + K_S}$$

Table 3 illustrates a fair agreement between calculated and measured absorptions. The correlation is poorest with the materials of higher modulus, BM-262 and cast iron.

With the Izod machine $U_S/U_0 = 0.26$ for BM-021 unnotched having $b = 0.50$ in., $h = 0.50$ in., $E = 1.25 \times 10^6$ psi, $L = 0.866$ in. K_S calculated from Equation [12] is 2.26×10^4 lb per in., so that K_M from Equation [10] is 0.80×10^4 lb per in. The energy absorption can be predicted in a manner already outlined for the Charpy machine. There is generally a wider difference between the actual and the predicted values than in the Charpy machine, especially when the specimen constant K_S, is high (BM-262, steel and cast iron) or low ($^1/_2 \times {}^1/_4$-in. materials).

The analogy of the system to two springs emphasizes that the low energy absorption is not some mysterious attribute of impact testing but arises simply because the machine constant K_M, is too small. Low energy absorption can equally well be observed in a flexural test but the usual testing machine is massive enough to make energy absorption practically 100 per cent.

The results for steel and cast iron are good only for small impact loadings in the elastic range, and the figures are no criterion of tests carried to ductile yielding or to failure. From the spring analogy one might conclude that in the ductile region the energy absorption would be nearly 100 per cent because there the specimen constant K_S, is low in keeping with the effective "modulus" in the ductile range.

Impact Energy Versus Span in a Drop-Weight Machine. From Equation [10] the energy of the falling weight U_0 is

$$U_0 = U_S\left(1 + \frac{K_S}{K_M}\right) \quad [10]$$

When U_0 is just sufficient to break the specimen at a given strain ϵ, the energy U_S, taken by the specimen, is

$$U_S = \frac{1}{216}\,bhLE\,\epsilon^2\left(1 + 3\,\frac{h^2}{L^2}\right) \quad [5]$$

and as

$$K_S = \frac{4Ebh^3}{L^3\left(1 + 3\,\frac{h^2}{L^2}\right)} \quad [11]$$

$$U_0 = \frac{1}{216}\,bhLE\,\epsilon^2\left(1 + 3\,\frac{h^2}{L^2}\right)\left[1 + \frac{4Ebh^3}{L^3K_M\left(1 + 3\,\frac{h^2}{L^2}\right)}\right] \quad [13]$$

If the breaking strain ϵ, is considered constant it is easy to demonstrate that a plot of U_0 versus span L goes through a minimum either by plotting values or differentiating a simplified expression with $h^2/L^2 = 0$. In the latter instance the span-depth ratio at the minimum L/h is

$$\frac{L}{h} = \sqrt[3]{\frac{Eb}{K_M}} \quad [14]$$

The effect of the depth of specimen h on the location of the minimum is elegantly illustrated by Fig. 6. For span-depth ratios higher than the minimum value the energy absorption soon reaches 100 per cent, as required by Equations [10] and [11], and the dynamic curve becomes a straight line. The absorption for the static curve is, of course, always close to 100 per cent.

In its linear region the dynamic curve lies well above the static,

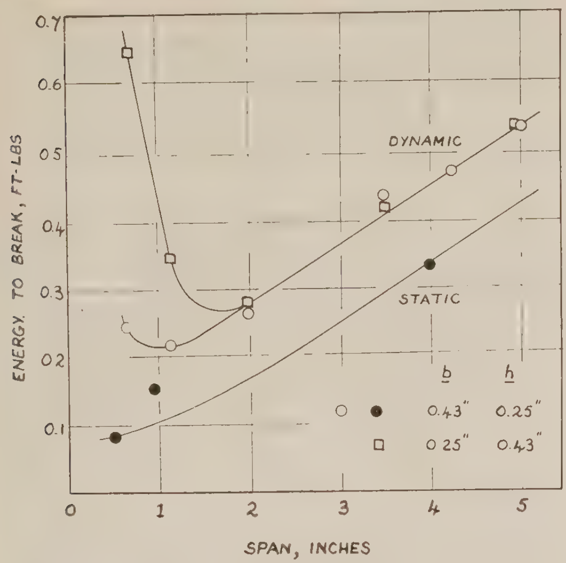

Fig. 6 Effect of Span on Energy to Break for BM-13080
(Dynamic test in drop-weight machine; static test in Amsler 60,000-lb testing machine.)

suggesting that the breaking stress is considerably elevated in impact loading. Measurement of dynamic strain in the drop-weight machine did not confirm this suggestion. In fact dynamic and static breaking stresses were equal and the energy absorption reached not 100 per cent but only a steady 70 per cent. Presumably the loss is in damping in the rather "dead" concrete base.

The existence of damping losses distorts this analogy of the impact system to two elastic springs but probably not too seriously if such losses remain constant with span.

Further Conclusions From Spring Analogy. The connection between displacement y, and time t, for the two springs is described by

$$M\frac{d^2y}{dt^2} + Ky = 0 \quad [15]$$

the solution of which is

$$y = A \sin\sqrt{\frac{K}{M}}\cdot t + B \cos\sqrt{\frac{K}{M}}\cdot t \quad [16]$$

Time of impact t_1, from the moment the pendulum strikes the specimen until the moment at which it leaves the specimen on the rebound for a blow insufficient to fracture is, from Equation [16]

$$t_1 = \pi\sqrt{\frac{M}{K}} \quad [17]$$

This time, as required by Equation [17] and established by experiment, is independent of the velocity and energy of impact provided that the specimen does not fracture. When the Izod hammer weighs 1 lb and $K_S = 2.26 \times 10^4$ lb per in. (BM-021, $b = 0.5$ in., $h = 0.5$ in., $E = 1.25 \times 10^6$ psi) and K_M (Izod) $= 0.80 \times 10^4$ lb per in.

$K = K_SK_M/(K_S + K_M)$ (from Equation [7]) $= 0.59 \times 10^4$ lb per in.

Hence $t_1 = 2.1$ millisec.

Strain-time photographs establish a mean for t_1 of 1.7 millisec for BM-021 and similar materials like BM-1914 and BM-13080. Fig. 7 shows sample photographs. The strain-time curves are often of a faintly saw-tooth outline, showing that the hammer momentarily loses contact with the specimen, which then starts to return to the initial undeformed position until the hammer overtakes it again. The slight oscillations dying away, in Fig. 7,

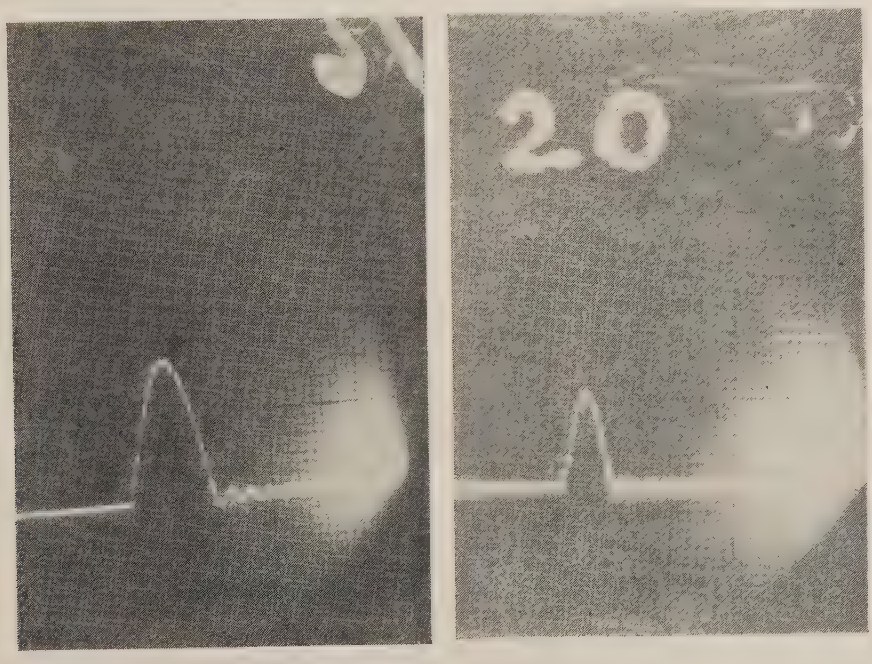

Fig. 7 Strain-Time Relations for Nonfracture Impact in Izod Machine
(BM-13080, left, $1/2 \times 1/2$ in. notched, and cast iron, right, $3/8 \times 3/8$ in. unnotched.)

arise because after the hammer has rebounded the specimen continues to vibrate feebly at its natural frequency.

When impact is violent enough to cause fracture the time of impact evidently must fall as the velocity of the striker is raised more and more. This is confirmed by photographs which resemble Fig. 7 except that the trace breaks off when the peak is reached.

When the limiting conditions: $t = 0$, $y = 0$, $dy/dx = v_0$ are applied to Equation [16] to evaluate the constants, it becomes

$$y = v_0\sqrt{\frac{M}{K}}\sin\sqrt{\frac{K}{M}}\cdot t \qquad [18]$$

The maximum value of y is when

$$\sin\sqrt{K/M}\,t = 1 \text{ or } \sqrt{K/M}\,t = \pi/2$$

then

$$y = v_0\sqrt{\frac{M}{K}} \qquad [19]$$

The force P_0 at this point is given by

$$P_0 = Ky = v_0\sqrt{MK} = v_0 M\frac{\pi}{t_1} \qquad [20]$$

where t_1 is the time of total nonfracture impact. Fig. 8 demonstrates that P_0 is linear in $1/t_1$; P_0 was calculated from readings of strain by the orthodox formula.

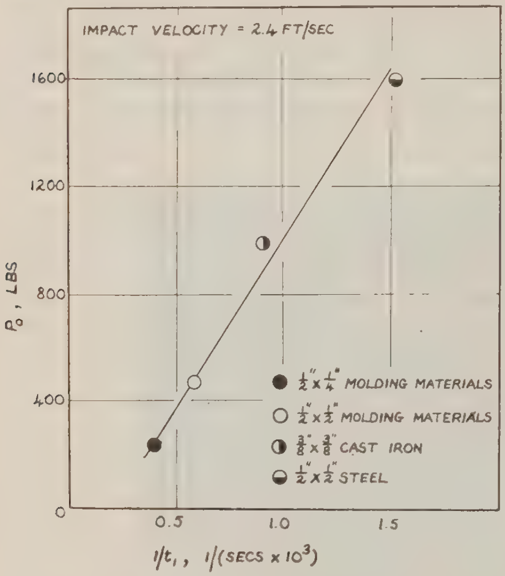

Fig. 8 Maximum Force in Pounds P_0, Developed During Impact Against Specimen as Function of Total Time of Nonfracture Impact for Constant Hammer Energy in Izod Machine

Limitations to Spring Analogy. The comparison of the impact system to two springs has allowed a successful interpretation of the experimental results in the following respects: (*a*) The effect of span, modulus, and dimensions on the per cent energy absorption in the Charpy machine; (*b*) the time of impact and its relation to maximum force; (*c*) the existence of a minimum in the curve of impact energy versus span in the drop-weight machine.

Deficiencies in the theory have appeared in the differences between results and predictions for the materials of higher modulus in both machines and for all materials in the Izod tester.

It is possible that these defects are connected with the assumption that the energy applied by the striker, divided between the specimen and the machine, is completely recovered on rebound, that is, that

$$U_0 = U_S + U_M$$

Actually this is never so with the Izod machine. The energy recovered, judged by the rebound of the pendulum, is as low as 45 per cent for a steel specimen. The energy not accounted for is probably lost by damping in the machine and its foundations. The magnitude of damping losses has already been discussed for the drop-weight machine.

The machine losses probably depend largely upon the machine itself and not too much on its attachment to a foundation, provided that the attachment is fairly rigid. It is possible that the nature of the foundation may have a marked effect on damping losses so that the Izod value indicated by the machine may depend upon its location.

Dynamic Breaking Strain. A specimen deforms into exactly the same shape as demonstrated by the distribution of strain, whether the loading is gradual or dynamic. Further, there is a close agreement between the dynamic and static moduli. So much has been proved.

In regard to breaking strain, sufficient results are not available for a final judgment. For velocities of impact up to 11 fps, it is tentatively concluded that the dynamic and static breaking strains do not differ by more than 10 per cent for the common phenolic molding materials. If this conclusion is firmly established it will follow that impact loading in the orthodox machines is, when properly analyzed, practically no different from flexural loading. This possible consequence opens up attractive vistas of simplification. The energy absorbed by the specimen in impact is incomparably more significant than the orthodox Izod and Charpy values. However, there are the objections that the end effect is not easily eliminated and the use of electronic strain equipment is undesirable for general testing. The simple alternative is to use the area under the flexural curve as the real impact energy. The flexural bar may be notched or unnotched according to the geometry of the molded object to which the results are to be applied.

The use of flexural energy brings out a very important point which has been concealed by the false results of the Izod test. Most common molding materials possess nearly the same linear load-deflection curve in flexure with nearly equal values of modulus of elasticity and of flexural strength. The unnotched impact strength is then almost constant. When there is a notch the impact strength is decided by the notch sensitivity, which is a measure of the ability of a material to withstand the weakening effect of stress concentration. Here the materials begin to show a wide range of real impact strength simply because notch sensitivity varies so much with the filler used (16).

Consequently, the "high-impact" materials, which have high Izod values, are of special value only when the molded object contains sharp notches or their equivalents; and their superiority over other molding materials almost vanishes when notches are eliminated by careful design and when they are not formed during service.

From considerations of notch sensitivity, the standard notched-bar impact tests, faulty as they are, have a much greater power of differentiating between molding materials than the orthodox tensile and flexural tests for which the specimen is not notched.

The effect of not only a notch but of modulus of elasticity, flexural strength, and ductility on resistance to impact is brought out clearly by the use of flexural energy.

Conclusions

The following conclusions are based on molded thermosetting plastics and certain thermoplastics:

Stress-Strain Diagram. The static and dynamic stress-strain diagrams are essentially identical, differing primarily only in the magnitude of breaking stress.

Strength. The dynamic breaking stress or strength is slightly elevated over the static flexural strength by approximately 10 per cent, resulting from the very high rate of loading. This is in agreement with previous data on the rate-of-loading effect as determined by other investigators.

Modulus of Elasticity. No perceptible variation in the modulus of elasticity is encountered under dynamic stressing compared with static loading.

Energy Distribution. In the static flexural test the energy absorption by the specimen is virtually 100 per cent. In the standard impact tests the energy absorption by the specimen varies from 10 to 50 per cent of the indicated energy loss of the pendulum. The remainder of the energy is lost in the machine and in kinetic energy of the broken parts of the specimen.

Notch Sensitivity. The ability of the standard notched-bar impact tests to differentiate between materials of like tensile- and flexural-strength properties is primarily due to the large variation in notch sensitivity in these materials.

Static Flexural Test. Static-flexural-test techniques employed to evaluate the energy of fracture and notch sensitivity for molded thermosetting plastics are concluded to be sufficient for predicting the service characteristics of such materials under both static and impact loading.

Theory. The theoretical concepts developed are not restricted to the testing of plastics but may be applied to the testing of metals and all other materials of construction.

Appendix

Izod and Charpy Energies at Low Swings. When the pendulum is released from a lower point than that for which the machine is calibrated, the energy is deduced from the scale reading (position to which pointer is carried by pendulum) for a free swing without any specimen in the vise. For example, for a 2-ft-lb machine having a velocity of impact of 11.3 fps:

Free swing, usual position, pointer carried = 0.060 ft-lb
Above repeated, pointer carried from 0.060 = 0.032
Therefore windage correction, full swing = 0.032
Windage correction, half-swing = 0.016
From lowered position, free swing, pointer carried = 1.381

For this reading after a specimen broke, correction to be subtracted is 0.036, according to calibration by manufacturer.

It can be shown, then, that energy at impact for free swing of 1.381 is

$$1.984 - 1.381 + 0.036 = 0.639 \text{ ft-lb}$$

$$\text{Velocity}^2 \text{ at impact} = \frac{0.639}{1.984} \times 11.3^2$$

or

Velocity = 6.4 fps

If specimen breaks when struck from lowered position and swing-through after fracture is to 1.945, then impact strength, i.e.

$$\begin{aligned}\text{Energy loss of pendulum} &= 1.945 - \text{correction for } 1.945 - (1.381 - 0.036)\\ &= 0.577 \text{ ft-lb}\end{aligned}$$

Dynamic-Strain Circuit. The entire strain circuit is illustrated in Fig. 9. Normally the measuring strain gage G is switched in, and the calibrating circuit is out. When the specimen is subjected to an impact strain the condenser allows only the voltage change to pass to the amplifier and to the oscillograph. With the strains measured, varying from 0.1 to 1 per cent, the extra amplifier A, is needed to supplement that already embodied in the oscillograph (Dumont, Model 208). The amplifier A is essentially identical to that of the second and third stages of the Y-axis amplifier of Model 208. Its chief virtue is the possession of a constant amplitude gain over a wide frequency range (60 to 10,000 cycles per sec).

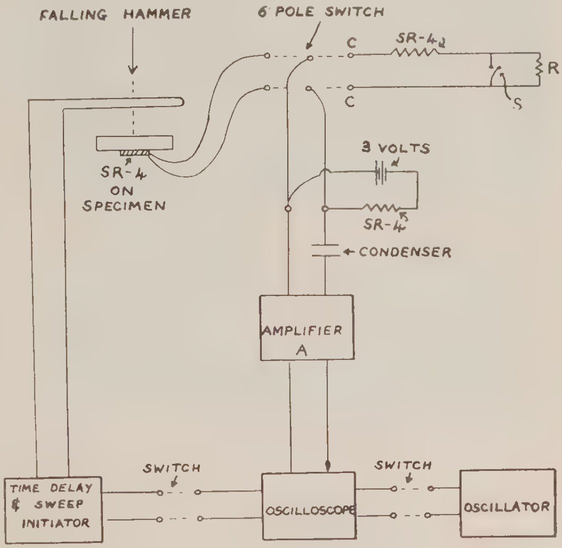

Fig. 9 Dynamic-Strain Circuit

When dynamic strain only is to be measured, the basic circuit, alone, comprising gages, amplifier, and oscillograph, is used. The oscillograph beam is narrowed to a spot, and the displacement on impact is estimated by eye. The screen is calibrated frequently by switching out the measuring gage and bringing in the calibrating circuit. When the microswitch S (normally closed), is opened by hand, an additional small resistance R is suddenly introduced and displaces the spot of the oscillograph. The value of this resistance R in terms of strain is determined as the difference in two readings when the calibrating circuit is attached to an SR-4 strain meter with the microswitch held steady closed and steady open.

Two additional circuits are added when strain-time relations are to be established, i.e., the oscillator circuit and the time-delay and sweep-initiating circuit. The latter device operates in the following fashion:

Initially the spot is off the screen. During its fall the hammer breaks a circuit which, after an adjusted time delay, causes the spot to sweep across the screen at a known rate. The sweep is in the center of the screen when the hammer striking the specimen superimposes a vertical strain displacement on the horizontal travel. The result is a record of the type shown in Fig. 7. The single horizontal sweep across the screen is secured by means of a

multivibrator circuit. With proper measures the velocity of sweep across the screen can be made constant throughout the travel. The time-delay device is a modified multivibrator in which one of the plate-grid connections is purely resistive. The tube containing this plate becomes nonconducting when the circuit receives a pulse on being broken by the falling hammer. Conduction through this tube to the sweep initiator is not re-established until there is a leak of charge from the grid to ground. The rate of leak can be varied by changing the capacity of the capacitive grid-plate connection.

After the strain-time relations are determined the oscillator circuit is brought in to determine the sweep rate. A sinusoidal vibration of 1000 cycles per sec is superimposed on the sweep across the screen so that a photograph gives a calibration of the horizontal axis in terms of time. The oscillator is of the resistance-capacity tuned feedback type.

When natural frequencies of vibration were being measured the external sweep initiator was eliminated and replaced by the built-in sweep circuit of the Dumont oscilloscope. This allowed a record of the type shown in Fig. 1 when the specimen was struck, and a calibration of the X-axis in terms of time when the oscillator was brought in.

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Data From Activity-Recording Instruments Applied to Textile Machinery

By C. S. PARSONS,[1] NEW BEDFORD, MASS.

The author advocates the adoption of activity recorders in the analysis of production machinery operations widely throughout the textile industry, as a means of achieving improved performance and lowered costs. After describing the several types of equipment and their applications to the study of textile-machine operations, the paper explains the procedure followed in analyzing the data recorded.

Use of Machinery-Activity Recorders

THE textile industry in general has not recognized the advantages of obtaining graphically data regarding machinery operations. There are four explanations for this as follows:

1 Low costs of labor and machinery compared with other industries.

2 In general, many productive units per machine operator.

3 Much engineering has been done by consultants, and the installation of this equipment for limited observations is expensive.

4 Nature of the machine does not readily lend itself to adapting the instruments.

With increased manufacturing costs, the use of machine-activity recorders becomes of much greater importance than formerly.

Wages have risen to new peaks and reduced costs must be realized through the more efficient use of manpower. Prices for machinery are also higher today, due not only to the increased cost of building the equipment but also because of the higher speeds, resulting in the necessity for better machining and closer tolerances.

The description which follows will show methods for analysis of machinery operations that will assist in reducing production costs.

Examples of Application

The use of recorders in studying weaving will serve as an illustration for this discussion. However, for mills that are not primarily interested in weaving, three of many other possible applications will be mentioned:

1 For cloth-finishing processes, the analysis of operation time will serve as a record of machinery utilization, as well as uniformity of cycle times for control of quality.

2 Another example of an indirect installation is that of the standard makes of regain controls, Fig. 1, which record the moisture remaining in the yarn after drying. When the sizing machine is stopped the instrument registers zero. As a result, data are available for the frequency of machine stops, time lost by the machine at the beginning of the work day, cleaning, or other reasons. In this operation the machine investment and wages for labor are comparatively high and the machinery efficiency is low due to fixed setup times.

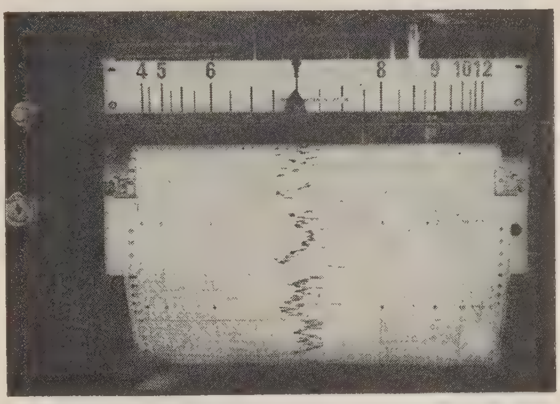

Fig. 1 Type of Regain Controller

3 Roving frames serve as an interesting example since many mills are installing new long-draft machinery to reduce the number of operations. The greatest causes of lost efficiency are end breakage, doffing, and creeling. In conjunction with conventional time studies, graphic recorders assist in the analysis of roving-frame operations for frequency of occurrence and time spent by the operator on the machine, as well as the time that the frame is not receiving the attention of the operator. This last down time includes the time when more than one of the tender's frames are stopped simultaneously.

Description of Recorders

Recording wattmeters, Fig. 2, may be used for obtaining data on one or more machines according to the electrical drive. However, their use is somewhat limited due to variable conditions in a group of machines and interpretation of average data.

Before considering the advantages of the use of instruments for loom studies, a brief description will be given of the design of recorders that are intended for machine activity only.

There are two common types of recorders and their selection will depend on the following factors:

(*a*) Permanency of the installation of the machine or instrument.

(*b*) Flexibility for use on other machines.

(*c*) Accuracy with which charts can be read.

(*d*) Ease in comparison of records from various machines to be studied.

(*e*) Availability of charts during study.

(*f*) Original cost of recorders per machine to be studied.

(*g*) Installation cost of instruments.

(*h*) Maintenance costs of charts.

For simplicity, the two types for looms will be identified as mechanical and electrical.

The mechanical type is clamped or bolted to the loom and any mechanical motion that will move or swing the pendulum located within the device will also move the stylus and make a record on the chart. In use, the recorder case is closed so that

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[1] Superintendent, Hathaway Manufacturing Company. Mem. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

the stylus marks the chart only during the time when the pendulum is put in motion by the working activity of the machinery to which the recorder has been attached. The circular charts used on this device are 4 in. or 6 in. diam and are designed for 1, 3, or 7-day operation.

The electrical type (Fig. 3 for example) is manufactured by at least two different companies. It is beyond the present purposes to make comparisons of the differences in design. The following description will be limited, therefore, to one maker's design, Fig. 4. This particular instrument is built for blocks of twenty machines and records their production on a continuous running chart with the records of each machine spaced about $^1/_4$ in. apart. The electrical type may have a mechanical clock mechanism, but the operation of the machine is transmitted to the recorder by means of an electrical switch connected to the shipper mechanism or photronic equipment.

When the productive machine is in operation an electromagnet raises the pen about 0.1 in. above the base line. When the machine is stopped, the electric current stops and the pen returns to its normal position. The chart is fed through the machine so that previously printed horizontal lines spaced 0.1 in. apart normally designate 1 or 2 min according to the type selected.

Application to a Loom

On a loom with an electrical stop motion the current is shut off when the loom stops. This switch is then used for operating the

Fig. 2 Model A.W. Switchboard Graphic Instrument

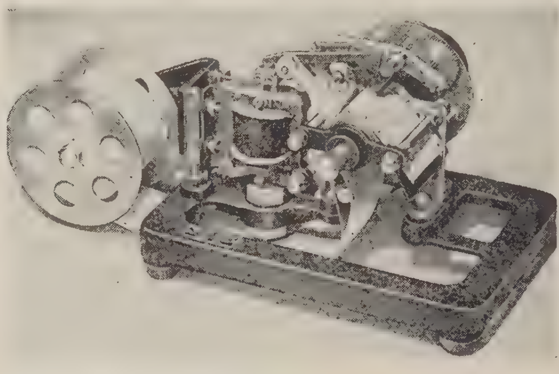


Fig. 3 One Type of Electrical Recorder

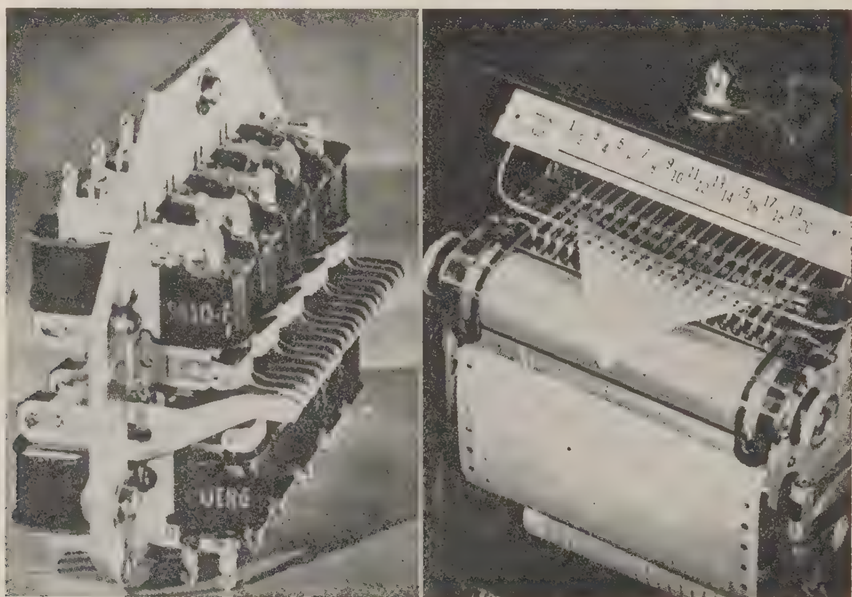


Fig. 4 Twenty-Pen Machine-Activity Recorder

(All of the electromagnets of a twenty-pen operation recorder are mounted on a single plate, but each is removable as a unit. Tilting the scale plate automatically lifts all twenty pens of the operation recorder to allow easy threading of the chart.)

pen of the recorder for indicating the stoppage for all causes. However, it is possible to cut into the wiring of the electrical stop motion and indicate on another pen the warp stoppage. A device could be arranged so that the filling-detector mechanism would actuate a third pen. Thus on one third the number of looms as might be otherwise studied, an analysis could be made of total stoppage, filling and warp stoppage, and by deduction, all other causes. The records would then conform with current practice of observing three major classifications of loom stoppage.

Explanation of Chart

One of the first impressions obtained in observing a chart such as Fig. 5, is the great variation from loom to loom even on the same cloth construction. This is readily apparent to those who are not acquainted with time-study analysis. The activity of the looms is more effectively shown by graphs than by figures, as is true of graphic presentation of such data.

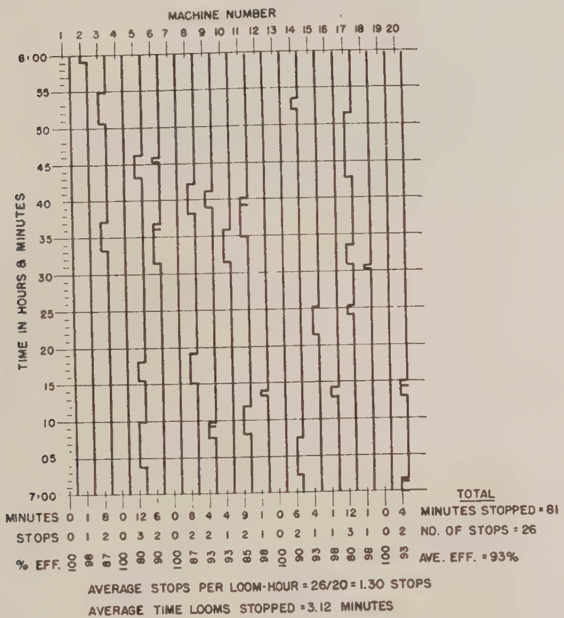


Fig. 5 Chart Made From Data Obtained With Electrical Recorder

Some engineers record operations in the sequence of their occurrence, which requires considerable clerical work in the summation of the data to follow the activity of any one particular machine. In contrast, some time-study observers record their findings directly according to the machine, irrespective of actual sequence. As a result, in a 4-hr study, considerable trouble may occur in a brief period, for example, in $^1/_2$ hr, and for the remaining seven eighths or $3^1/_2$ hr of the test the performance might be normal. Such instances are clearly seen by the use of graphic recorders.

Further examination of Fig. 5 shows that the machine numbers are across the line on the top of the page. One hour has been taken to illustrate this chart. There are 60 horizontal lines with 12 heavy lines representing 5-min intervals, as shown on the extreme left-hand margin. It will be seen that loom No. 3 was stopped for 4 min from 7:33 to 7:37.

Beneath the chart are the total stops for each loom for the hour and the total number of minutes stopped.

The efficiency of each loom can be readily calculated from the minutes of lost production.

By adding across each minute line, as shown in the right-hand column, data may be obtained on the number of looms stopped during each minute.

This twenty-pen machine-activity recorder, Fig. 4, makes graphic records of the production of twenty machines at one time. By suitable electrical connections to the starting or knockoff mechanism of the machine it is possible to determine the following:

1 Number of stops.
2 Duration of stops.
3 Frequency of stops for any period.
4 The number of machines stopped at any given period.
5 The amount of down time for that period.

This machine will give a graphical record of the quantitative analysis of the production and, if customary time studies are made in conjunction with it, there should be an accurate analysis which could not be obtained by other means. It is recognized that a complete analysis of any problem should contain quantitative and qualitative studies. For the work in textile mills the cause of machine stoppage is of secondary importance if the frequency can be accurately obtained, since if there is not a large amount of machine stoppage, the causes are not of great importance. Frequently it may be necessary to make a special analysis of the production of the machinery in order to determine the cause of machinery being stopped. This device will not give an accurate unit time that the operator spends in starting the machine in all cases, inasmuch as the machine will be stopped automatically in most cases and as a result, the down time of the machine will include:

(*a*) Time spent by the operator arriving at the machine.
(*b*) Time spent by the operator working to get the machine in operation.
(*c*) Actually starting the machine.

In most cases, such as weaving, the last two factors are very closely related and for most practical purposes can be combined.

Weekly Variations Per Machine

In order to determine the operation of one style a chart has been made from data obtained with the electrical recorder. In Fig. 6 is shown the analysis of the stops per loom-hour from ten looms operating 81 hr in 1 week. Loom No. 11 had an average of 3.04 stops per loom per hr, and an efficiency of 77.1 per cent. Loom No. 17 showed much less stoppage and operated at 90 per cent efficiency.

The test was operated for only 5 hr on the first shift Monday, 8 hr on the second shift that day. On Tuesday the results of 7 and 8-hr shifts are shown. It is interesting to observe that loom No. 12 operated at more nearly normal efficiency and although it ran poorly Thursday morning, it ran well the last of the week.

Variations Per Machine

Fig. 7 shows a probability curve of the variation in stops per hour. From this graph it is apparent that 21 per cent of the stops per loom per hour will be the average for the total test. Likewise, $7^1/_2$ per cent will vary approximately 30 per cent below or above the average and 10 per cent of the stoppage will be approximately 25 per cent higher or lower than average.

Similar data were obtained and analyzed for an entirely different cloth construction. In this case 32 per cent of the stoppages were average instead of 21 per cent. Approximately 8 per cent of the recorded stoppages varied 30 per cent above or below the mean.

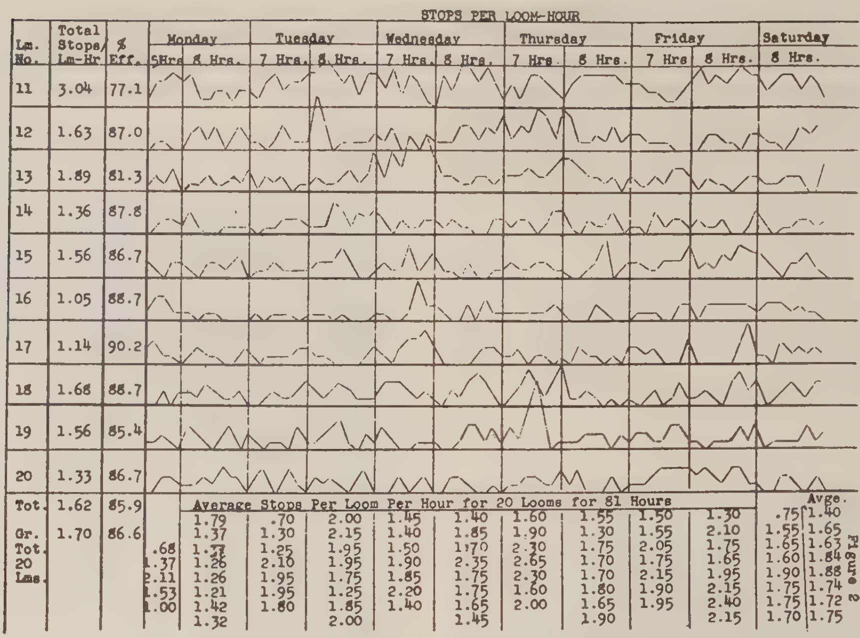


Fig. 6 Stops per Loom per Hour for Each Day

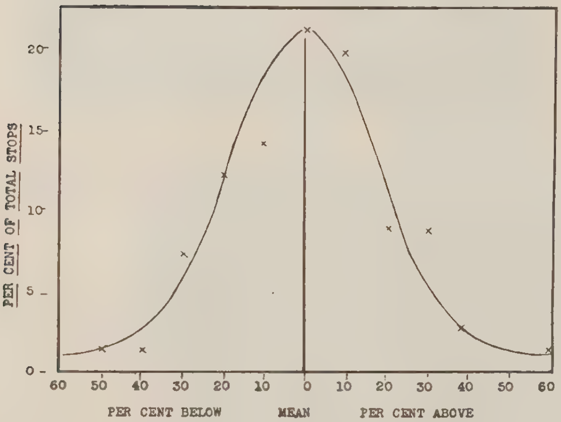


Fig. 7 Variation in Stops per Loom per Hour
(Average 20 looms; continuous testing for 81 hr.)

These data are of interest in selecting a representative test when conventional time studies are made.

Stops Per Hour Versus Unit Times

Fig. 8 shows the relationship between stops per loom-hour and weaving efficiency from a 50-hr study of twenty looms. Unfortunately, this relationship is not always recognized and the values do not closely correspond since (1) the elemental time values will vary according to the nature of the stop, and (2) the weaver is not always immediately available to start the loom.

Stops Per Weaver and Production Efficiency

If the recording instruments are to be used for weavers' work assignment, it will be interesting to examine Fig. 9, an analysis of stops per weaver per hour. In the upper half of this chart is

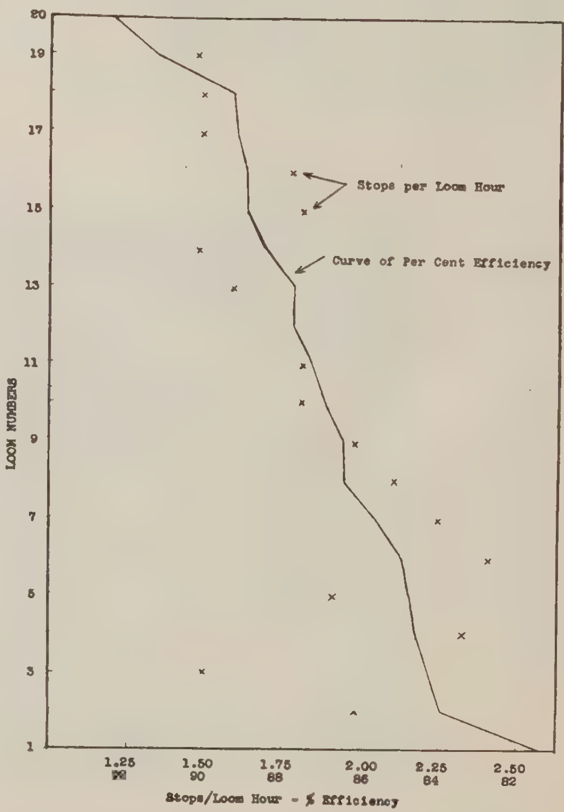


Fig. 8 Graph Showing Relation of Per Cent Efficiency to Stops per Loom per Hr

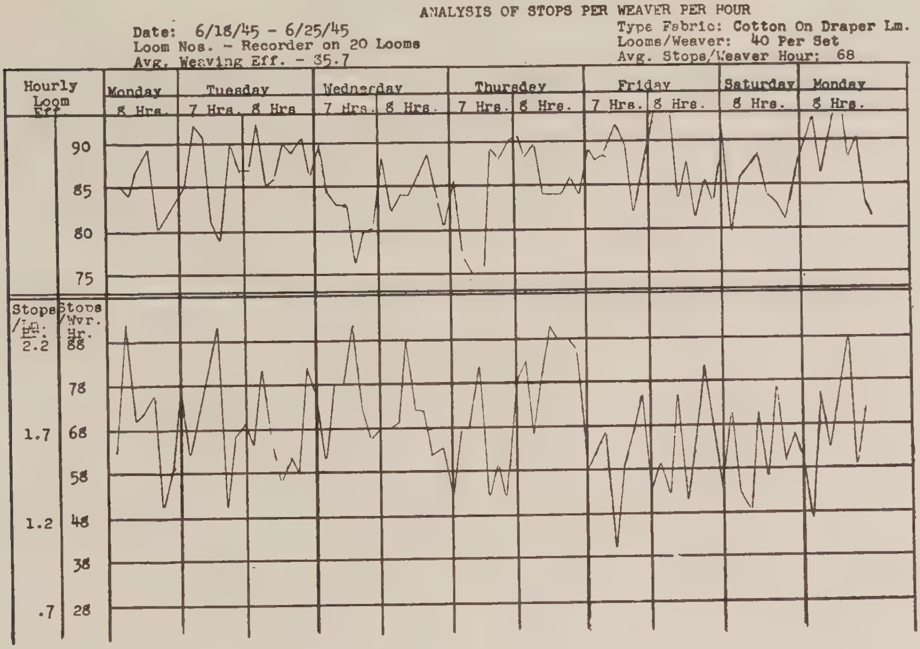


FIG. 9 ANALYSIS OF STOPS PER WEAVER PER HR

the hourly fluctuation in weaving efficiency of twenty looms for two shifts operating one week.

In the lower half of this figure is the variation in stops per loom-hour. The best shift performance was on the first shift Friday. No third shift was in operation and yet the looms ran well the first hour in spite of their being stopped. It is apparent that a test for only a few hours on one day probably would not give representative data.

Machine Interference

Published material on time studies contains very little regarding the problems resulting from multiple machine jobs. As was previously mentioned, the number of looms stopped at one time is a limiting factor of machine efficiency. An automatic loom has stop motions which operate when a yarn breaks or certain mechanical parts are not in correct adjustment. When all looms are in operation the weavers' duties are limited to inspection and a few minor tasks. Therefore the measurement of the amount of time when a varying number of looms are stopped is of great importance. To obtain these data by personal observation is difficult when a large number of looms is tended by one weaver. However, the electrical recorder makes an ideal record which not only makes the data easily obtainable but also shows graphically when abnormally poor conditions occur.

Interference Time

Interference time, as previously mentioned, occurs when more than one machine is stopped and the operator is able to work on only one machine at a time. For example, assume an operator is assigned four machines; at certain periods all machines may be running; then one stops, and he works on that machine. Then a second machine may stop before the other machine has been put back in operation and interference time occurs. The amount of lost time for this reason will depend on the following:

1 The machine setup time.
2 Number of machines per operator.
3 Average time to start one machine.
4 Frequency with which the machine stops.
5 The number of other employees that may assist in the operation of the same machines and the amount of their time.

In order to measure interference time, formulas have been set up by several engineers, but as yet there is none that has been generally accepted. This is unquestionably due to the difficulty in obtaining the data for proving the formulas. The present purpose is to show that with the use of activity recorders the desired information can be readily obtained, and that unless sufficient data are accumulated, erroneous deductions may result. In order to appreciate the importance of interference time it should be mentioned that in a recent weavers' arbitration an allowance in earnings of $7^1/_2$ per cent was claimed.

Example of Interference

An actual example of loom-stoppage tests to illustrate the importance of the problem is given in Fig. 10. Conventional time studies were made simultaneously with the activity recorder during 4-hr periods. Two of these tests were chosen showing the effect of the weaver on the production. In the case of weaver A, 88 per cent efficiency was obtained, and weaver B, working on the same job on the second shift, obtained 95 per cent efficiency. Each of the first four curves represents one hour's work of weaver A.

These graphical records analyze the number of looms stopped at $^1/_2$-min intervals. In the top curve, for instance, it is seen that at the start of the test all looms were running; $^1/_2$ min later one was stopped; and at the end of 1 min three were stopped. Four looms had stopped in the first $1^1/_2$ min of the test. At the end of 5 min all looms were again in operation. After 35 min all looms were running for approximately 5 min.

It is apparent that the four lower curves representing weaver B show much better weaving efficiency, since there is less area enclosed by the curves above the base line.

The summary of these tests is shown in Fig. 11. These illustrate that when weaver A ran the looms for 26.5 per cent of the

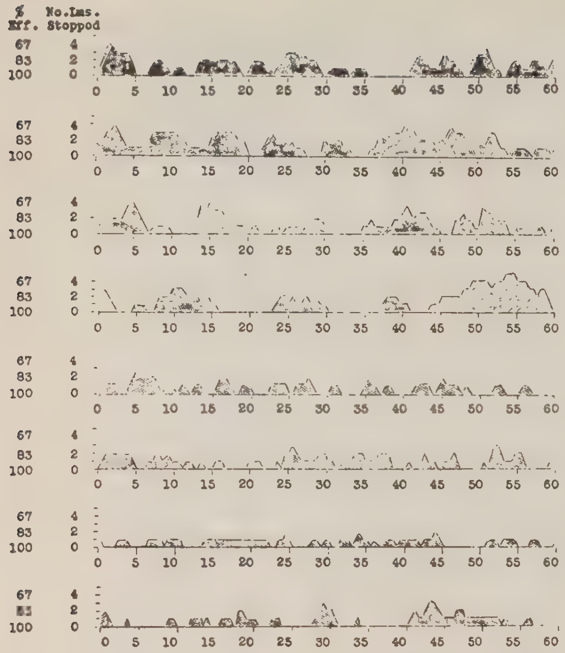


Fig. 10 Analysis of Looms Stopped and Weaving Efficiencies
(Results of eight 1-hr tests. Observations made at 1/2-min intervals.)

480 observations, no loom was stopped, and 28.3 per cent of this time one loom was stopped. In contrast, weaver B with the higher efficiency, had all looms in operation 48.1 per cent of the time, and only one loom stopped 40 per cent of the time.

It is believed that these data can be used as a basis for determining certain qualifications of the weaver, such as skill, effort, etc.

It should be emphasized that the number of looms stopped at one time does not necessarily indicate the running qualities of the job, but how the job was operated. The data below Fig. 11 are significant, namely, 7 per cent less production, 40 per cent greater unit time, resulting in much greater time lost and the looms not in operation. The interference time in the first case is 5.2 per cent and 1.2 per cent as shown for weaver B.

The number of looms that should be assigned to a weaver is governed by the frequency of loom stoppage and time spent working on such duties. In addition there must be adequate allowances for patroling the job, miscellaneous tasks, and personal time. The patrol and inspection time frequently established is 20 per cent of the total weaver's time. Modern quality-control methods establish statistical procedures for analyzing the frequency of inspection. Today less inspection is required in many instances than was formerly practiced. It therefore becomes of greater importance to demonstrate that other work is not required of the weaver. The methods just described will prove that there is not an excessive number of looms stopped requiring work by the weaver.

Conventional time studies show the amount of time the operator is not occupied performing his regular weaving duties to keep his machines in operation. When it is found that additional work can be performed, the total available time may be divided by the number of such time-study readings to obtain an average unit time available for other work. This elemental time used in conjunction with data regarding the average number of machines stopped at one time, may be of assistance in establishing the correct work assignment.

The data obtained from this recording device accumulate so quickly that the need for modern statistical analysis of test data soon becomes apparent. By means of such analysis standard deviation of data can be obtained. In addition, the adequacy of test observations can be calculated. This will unquestionably result in establishing new standards for the length of time that observations are necessary. Frequently, substantial savings in the cost of time-study observations will result.

Conclusions

It is apparent that the following information is readily available from the charts:

1 Frequency of stoppage per hour or any other desired period.

2 Duration of time for each stoppage.

3 Machine efficiency of any recorded unit or machines for any time period of the test.

4 Comparison of performance of similar machines.

5 Comparison of the performance with similar products.

6 A number of machines in one operator's job can be recorded at one time on one sheet.

7 Variation in operations:

(*a*) per shift.

(*b*) comparison of shifts.

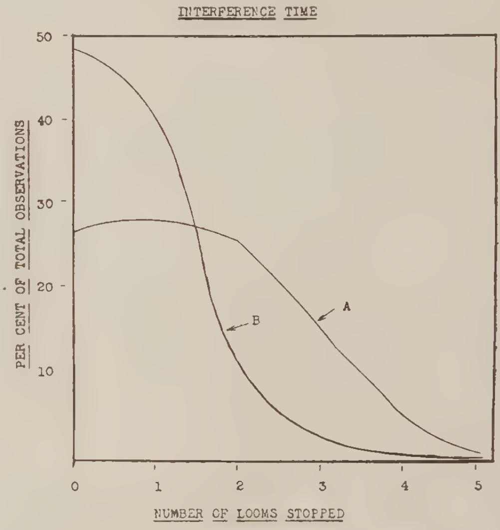
(*c*) comparison of performance on different days or even greater periods.

8 Starting and stopping time of shift.

9 Instantaneous records for immediate investigation.

10 After installation, continuous records at minimum expense.

Other data were accumulated for further analysis which frequently are not easily obtainable, such as the following:



	Weaver A	Weaver B
Weaving Efficiency	88	95
Stops per Producing Loom Hour	2.4	2.1
Average Time per Stop in Minutes	.98	.70
Per Cent of 4 Hours Weaver Spent on Stops	54%	29%
Interference Time (Waiting for Weaver)	5.2%	1.2%
Average Number of Looms Stopped When Weaver was Available	1.1	.34

Fig. 11 Interference Time

1 Weekly variations per machine.
2 Variations for similar machines.
3 Stops per operator and production efficiency.
4 Machine interference.

Adoption of activity recorders in the analysis of production machinery operations will result in improved performance, which will automatically reduce costs. The data obtained from these instruments will serve for work-load assignments based on adequate evidence of frequencies of occurrence.

Discussion

N. M. Mitchell.[2] The paper obviously has as its objective the pointing out of benefits which may be obtained through intelligent use of recording instruments.

The author discusses his activities in connection with the use of the Esterline-Angus unit which has proved so useful to him in his own mill-engineering work. He obviously recognizes that the data which can be collected by means of this equipment can be misleading and actually dangerous if used either after insufficient testing to cover truly representative periods, or without due consideration of the practical elements, such as types of operatives, types of fabrics, conditions of machinery, and surrounding conditions.

The crying need in the textile industry today from an engineering standpoint is the determination of the true value of interferences of all types in direct-labor operating activities. The type of instruments described by the author comes more nearly to providing a means for the correct determination of interference than any equipment which has been presented for consideration to date. The fact that the type of equipment referred to is costly and somewhat cumbersome should not deter engineers from purchasing and using it, because its improvement and ultimate meeting of the real need for the industry can come only through reasonably wide distribution and use by engineers in the various mills.

There is definite need that the engineering profession recognize the fact that publication of information collected in a casual manner can be of far more damage to a mill than would be the case if the subject were to be left untouched. The use of recording instruments in connection with work-load studies or machine studies will unquestionably expand and become general throughout the field, providing its early use is not abused, or the information reported as a result of its use is not based upon incomplete or short-period testing.

This paper, covering as it does the use of recording instruments, furnishes a great deal of valuable background information. It is interesting to note the state of operational affairs from the standpoint of loom stoppage, recorded in the chart Fig. 5. This indicates that the loom stoppages at given intervals throughout the study ranged from zero to four. It would be extremely difficult to establish this fact without this type of recording instrument. Analysis of this information can serve as a means of evaluating interference and its influence on a weaver or operative of any type of multiple-machine operations. Without accurate means of measuring consecutive and respective stoppages in machinery in multiple-load activities, it would be extremely difficult to evaluate interference and properly analyze the situation.

The writer believes there is great need to devote intensive study to work-load interference in order to prevent the subject becoming one of mystery and possible misuse in respect to establishing work loads and piece rates in the textile industry. There is a possibility that the subject may have a harmful psychological influence on mill operatives unless it is treated intelligently and resolved into matter-of-fact common-sense usage by mill management.

The use of recording instruments will unquestionably bring about the revelation of substandard mechanical conditions which otherwise might pass unnoticed if these conditions had to depend only on observations of work-load students, without the use of a means for multiple coverage of a job.

Present-day labor-management conditions make it necessary to establish accurate true facts relative to all phases of machine operations as well as that for the human being. The writer is of the strong opinion that the equipment described in the paper and used along the lines obviously applied in the collection of the information which was submitted, is very much worth while and its use should be encouraged if possible in all mills.

A. Palmer.[3] Many mills are faced with problems relating to work assignment that can be solved only by scientific methods. In many cases time studies and loom-stoppage studies are satisfactory. In others they are not, either because they are resented by the employees or because they do not give the kind of information that is needed.

We have been asked by mills recently how loom-stoppage data can be obtained without taking time studies. Our reply has been to refer them to this paper.

As the author points out, the time-study data that many of us have been taking show the average performance of a set of looms. Unless we analyze the data very carefully, we have no knowledge of the performance of each individual loom. The recording instruments used by the author give him not only the average picture, but also a clear one of the performance of each individual machine.

Is it possible to use these recording instruments to show the causes for loom stoppage? If these instruments can be connected to differentiate warp breaks, filling breaks, and miscellaneous stoppages from each other, some very desirable information can be obtained.

One problem that, in so far as the writer knows, no one has been able to solve satisfactorily is the question of loom interference. Years ago Mr. McElroy and various others tried to set up work-assignment formulas that took this element into consideration. They did exceptionally well with the information that they had at their disposal, but they will probably agree, at least in some measure, with the writer's feeling that there still is work to be done in this connection by someone who is familiar with higher mathematics, particularly with the theory of probability.

In this connection it seems to me that from the records taken by instruments such as the one the author has been using, much essential basic data could be obtained. Put into the proper hands, these figures should enable someone who knows how to supply the necessary mathematical theory to develop a formula that would be useful, at least in the case covered by the particular records that were used

Much work of this nature has been done by Dr. Thornton C. Fry of the Bell Telephone Laboratories. He has supplied some of the mathematical procedures developed for the use of the telephone companies to the problem of assigning work loads in industry. To the layman his computations appear to be very complicated, not only because of the fundamental mathematical processes, but also because of the large number of variables that he includes.

In our opinion, Dr. Fry could help the textile industry if that industry supplied him with data similar to that which has been accumulated by the author. It is believed also that if the textile industry indicated to Dr. Fry that a simplified process was re-

[2] President, Barnes Textile Associates, Inc., Boston, Mass. Mem. A.S.M.E.

[3] Vice-President, Crompton & Knowles Loom Works. Worcester, Mass.

quired, he might be able to give us a sufficiently complete answer. If we know the average length of time during which looms stop; if we know the frequency of the occurrences and any other data that are pertinent to the loom-stoppage question, isn't it enough to determine what the best number of machines per operative is to keep the operator fully occupied but not overburdened, and to give the maximum production per loom?

If the writer understands Dr. Fry's work, he has included many considerations such as the weavers' wages, overhead expenses, and items of that nature, all of which are necessary but which seem to complicate the problem at the outset. If we can find out, on the basis of the loom-stoppage data, the number of looms that an operator should run to give the maximum production per loom, then the cost items automatically take care of themselves. It is therefore the writer's suggestion that as soon as the author, Mr. McElroy, and others who are doing this work, have sufficient data, they consult Dr. Fry to see what he can do with them.

A. N. Sheldon.[4] The advantages accruing from the information provided by the use of activity-recording instruments seems quite obvious from the paper. The utility of its general adoption would appear to justify the cost if the data derived therefrom become the basis of a careful and continuing analysis; the records are accurate, impersonal, visual, and permanent, for guidance of both management and operatives.

There is one deduction that appears possible from the data in Fig. 5, and that is that for this particular example there were too many looms per weaver; out of 20 looms there were only 5, (looms 1, 4, 7, 13, and 19) that operated continuously throughout the hour covered by the record, and the remaining 15 looms stopped 26 times and were unproductive for a total of 81 min. Consequently the efficiency of these 15 looms was only 90 per cent; the stops per loom per hour were 1.7, and the average duration of each stop was 5.4 min. Expressed in another way, out of these 20 looms, 1.30 looms were stopped for the entire hour. For a group of 1000 looms this is equivalent to 65 looms continuously idle, if the record for 1 hr is typical of all other hours. Since the capital invested in a single loom might be $1500 (including the cost of loom, building, air conditioning, power and light wiring, heating, etc.) there would be approximately over $100,000 of invested capital totally and always unproductive.

Fig. 5 also shows that there were only 12 min in the hour, or 20 per cent of the time, when all 20 looms were running simultaneously, which together with the average duration of stoppage of 5.4 min per loom, is strong circumstantial evidence that the weaver's job for this particular situation was too extended, with consequent loss in production from the looms and unproductive effort by the weaver.

Whether or not the foregoing conclusion is correct, the activity recorder presents a reliable device for investigating its validity, and as indicated by the chart in Fig. 5, there is nothing gained by increasing the looms per weaver beyond the point where loss of production, arising from "interference time" becomes the dominant factor. Incidentally, it does not seem consistent to penalize the loom-operative efficiency with interference time just because the weaver is not present to run it.

[4] F. P. Sheldon and Son, Engineers, Architects, Appraisers, Providence, R. I. Mem. A.S.M.E.

Discussion

In 1945, when there was a ban on national meetings, some papers originally scheduled for these meetings were presented before local groups. In the case of these papers the Committee on publications suspended its rule, which requires simultaneous publication of paper and discussion, and accepted discussion based on the published paper.

Heat Transfer in the Locomotive Boiler[1]

F. P. Huston.[2] The author presents in this paper a procedure for determining the output of any given quality of steam from conventional locomotive boilers for given rates of heat release and gas production.

The first part of the computation deals with the transfer of heat through radiation to the firebox and combustion surfaces. The second part deals with the transfer by convection through the walls of the tubes, flues, and superheater pipes. The author shows that the results of the computations are in reasonably close agreement with test data.

The variations between the maximum evaporation estimated by present methods and the maximum obtained on test, as reported by the late C. A. Brandt,[3] on ten locomotives range from 47 per cent more evaporation obtained on test over the estimated maximum to 14 per cent less. This wide spread is apparently due to the rule-of-thumb methods now in use, as exemplified by the 55-lb rule, dating back to 1912, as the result of the "Coatesville tests," or Baldwin's 80-lb rule. Either of these figures will apply at some low to moderate firing rate, but many locomotives are used today at near the maximum steaming capacity where the evaporation from the firebox surfaces may approach a rate of 125 lb of water per sq ft per hr.

The author, in presenting the formulas derived from the generally accepted fourth-power equation for the transfer by radiation in the firebox and the double logarithmic equation for transfer by convection in the flues, provides a means to compute the expected performance of the present-day conventional boiler to greater accuracy and enables a better proportioning to be made between the firebox and the flue areas. His contribution should prove of great value in studying the influences of changes which obviously must be made in the combustion of the fuel and features of design to enable steam-locomotive boilers to meet the requirements imposed by the modern trend toward higher steam pressures, greater demands, higher superheat temperatures, and better fuel economy.

H. S. Vincent.[4] In the opinion of the writer the author's Fig. 5 would be more useful had it been constructed to show the temperature of the combustion gases at the rear flue sheet instead of a mythical equilibrium temperature. With such data the heat absorbed through the walls of the firebox would be given by the expression

$$H_a - H_t = h_f \quad [1]$$

where

H_a = available heat of combustion
H_t = heat in gases at rear tube sheet
h_f = heat absorbed through walls of firebox.

The use of the author's Equation [1] for radiation from combustion gases cannot be defended, as has been shown conclusively by Schack (1),[5] Hottel (2), and other investigators. Equation [1] of the paper can apply only to radiation from solid fuel where the total radiations are normal to the cold surface. To give correct results the equation must include a factor allowing for the average angle through which the radiating surface "sees" the absorbing surface. Hottel (3) has given data for evaluating such a factor in cubical furnaces as well as in other types of heat exchangers. In so far as the writer is aware nothing of this character has been worked out for the locomotive furnace.

About 10 years ago the writer published a series of articles (4) on the general subject of heat transfer in the locomotive boiler with the particular purpose of determining the proportion of available heat taken up by its various components, viz., firebox, tubes, and superheater. The data there given were derived from 62 laboratory tests of nine different types of locomotive boilers. The method used in determining heat transfer through the walls of the firebox is given by the writer's Equation [1]. The principal chance of error in using this method is from excess air introduced through the fire door or above the fire. With automatic-stoker-firing, this contingency does not often arise.

It will be observed from the author's Fig. 3, especially in the series C, that there is excessive fluctuation in the measured temperature of the gases in the firebox in relation to the rate of firing. The reason for this abnormal variation in recorded temperature is that only one temperature determination is made for each complete test whereas for the temperature in the smokebox, as well as for all other data, frequent determinations are made. In any calculation based upon the firebox temperature it is evident that the results will be more logical and accurate if a mean of the observed firebox temperatures is taken rather than those actually recorded. This was the method pursued by the writer in the articles referred to.

The author has given as criteria his Table 4 and Fig. 5, for estimating the heat transfer through the firebox walls for any given locomotive. Using these data, the writer has prepared Table 1 of this discussion, for the locomotive series C, for which the author includes four tests. The writer's Table 1 includes eleven tests of this locomotive, only two of which coincide with tests cited by the author.

The data for columns 4 and 5 of Table 1 are taken from the articles cited and are based upon the Fry (5) method of calculating these quantities. Column 6 gives the equilibrium temperatures, as established from the author's Fig. 5, although they do not agree with those shown in his Table 3, column 4. Column 7 of Table 1 is the so-called "radiation factor," taken from the author's Table 4, at the appropriate temperature. Column 8 of Table 1 gives the heat absorbed through the firebox walls in per

[1] By Lawford H. Fry, published in the February, 1946, issue of Trans. A.S.M.E., vol. 68, pp. 107–113.

[2] In Charge of Railroad Development, Development and Research Division, The International Nickel Company, Inc., New York, N. Y. Mem. A.S.M.E.

[3] "The Locomotive Boiler," by C. A. Brandt, Trans. A.S.M.E., vol. 62, 1940, pp. 379–419.

[4] Consulting Engineer, East Harwich, Mass. Mem. A.S.M.E.

[5] Numbers in parentheses refer to the Bibliography at the end of this discussion.

TABLE 1 EXPERIMENTAL DATA AND COMPUTED VALUES FOR PERCENTAGE OF AVAILABLE HEAT TRANSMITTED THROUGH THE FIREBOX WALLS OF LOCOMOTIVE BOILER SERIES C

Series 1	Test no. 2	Firing rate lb dcf per sfg per hr 3	Heat available for absorption btu per sq ft of firebox hs 4	Mixed gas lb per hr per sq ft of firebox hs 5	Equilibrium temperature, author's Fig. 5 6	Radiation factor, author's Table 4 7	Heat absorbed through firebox walls in per cent of heat available for absorption: Author's method 8	Writer's method 9	Difference in columns 8 and 9, per cent 10
C	532	67.3	119200	157.5	1810	41600	34.9	48.3	13.4
	539	80.5	136500	185.1	1850	44700	32.7	43.9	11.2
	511	94.4	167200	210.5	1970	54900	32.9	45.8	12.9
	520	105.4	180100	213.0	2040	61700	34.2	46.8	12.6
	521	108.8	178600	212.5	2040	61700	34.5	45.9	11.4
	530	112.8	194200	241.5	2040	61700	31.7	42.6	10.9
	519	122.0	202000	247.1	2060	63700	31.5	42.3	10.8
	526	128.1	204100	230.0	2130	71200	34.8	46.0	11.2
	531	146.9	224000	272.5	2115	69550	31.1	38.8	7.7
	527	174.6	267000	325.0	2160	74600	27.9	35.8	7.9
	523	231.2	336500	386.0	2300	92000	27.3	30.1	2.8

NOTE: Dcf = dry coal fired; sfg = square feet of grate area; hs = heating surface.

cent of the heat available for absorption, using the author's method of calculation. Column 9 gives data similar to those of column 8, but calculated in accordance with the writer's Equation [1]. Column 10 shows the difference in percentage between columns 8 and 9. This varies from about 3 to 13 per cent and indicates to the writer that firebox absorption, as determined by the author's formulations, is too low by this amount.

In the present state of the art it seems inadvisable to attempt to show the exact method by which the available heat reaches the firebox walls. This determination must await the patient work of investigators who are equipped to take all of the complex elements of the problem into consideration. A very great need today is for many more laboratory tests of the locomotive with the results made available to all who can use them. It seems most fitting that such tests should be made under the auspices of the Federal Government and could well be conducted by the Bureau of Standards. It can be said without contradiction that our present knowledge of the thermodynamics of the steam locomotive is largely due to the generosity of the Pennsylvania Railroad in making available records of tests conducted at their plant in Altoona.

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1 "The Influence of Radiation in Coal-Fired Furnaces, Etc.," by W. J. Wohlenberg and E. L. Lindseth, Trans. A.S.M.E., vol. 48, 1926, Appendix 5, p. 922.

2 "The Radiation of Furnace Gases," by H. C. Hottel and R. B. Egbert, Trans. A.S.M.E., vol. 63, 1941, p. 297.

"Radiant Heat Transmission From Water Vapor," by H. C. Hottel and R. B. Egbert, Trans. A.I.C.E., vol. 38, 1942, p. 531.

3 "Heat Transmission," by W. H. McAdams, first edition, McGraw-Hill Book Publishing Company, Inc., New York, N. Y., 1933, p. 50.

4 "Heat Transmission in Locomotive Boilers," by H. S. Vincent, *Railway Mechanical Engineer*, vol. 109, May, 1935, p. 180.

5 "A Study of the Locomotive Boiler," by L. H. Fry, Simmons-Boardman Publishing Corporation, New York, N. Y., 1924.

C. J. SURDY.[6] In more recent years the so-called "80 lb rule" has been quite generally used by locomotive builders in calculating the evaporative values of the boiler direct-heating surfaces. The presence of these new and higher values has been mentioned by both Johnson and Brandt but Mr. Fry in his paper has clearly and pointedly attributed this to high rates of combustion. Table 3 (columns 3 and 6) of the paper offers abundant proof of the existence of a 55 lb-125 lb evaporative range per sq ft of direct heating surface, depending upon combustion rates.

The present work may be viewed as an extension of the author's treatise "A Study of the Locomotive Boiler" (Simmons-Boardman Publishing Corporation, New York, N. Y., 1924) in which tables 5, 9, 10, 11, 12, 13, and 14 show exhaustive test results forming the original premise of the conclusions set forth there and expanded in the present paper. In this earlier treatise the steam produced at the various rates of combustion is equated in terms of the total heating surface of the locomotive firebox. However, it is clear from this study and the present paper that a definite relationship exists between rate of combustion and evaporation per square foot of heating surface.

To locomotive design engineers this paper presents an authoritative exposition of the factors which make the modern high-capacity steam locomotive possible. The author should be complimented for this further contribution to his studies of the locomotive boiler.

[6] Assistant to President, The Standard Stoker Company, Inc. Mem. A.S.M.E.

AUTHOR'S CLOSURE

The author appreciates the comments made by Messrs. Surdy and Huston.

Mr. Vincent's discussion contains several points of interest. The author welcomes comparison between the present paper and Mr. Vincent's extensive and valuable study of heat transfer in locomotive boilers (4) published in 1935 which deserves to be better known. Mr. Vincent took the results obtained in nine series of locomotive tests and by noting the rate of heat release in the firebox, the weight of the gases of combustion, and the temperature at which they left the firebox, he computed the rate at which heat was taken up in the firebox. From this the total heat taken up by the boiler could be separated with considerable precision into its three components, heat absorbed in firebox, heat absorbed in flues and tubes, and heat absorbed in superheater. In his discussion of the present paper, Mr. Vincent points out that his values for heat taken up in the firebox are higher than those given in the paper. This is true and further consideration of the problem leads to the conclusion that a modification should be made of the statement in the paper that "radiation at the equilibrium temperature corresponds to the heat actually taken up by the firebox." Some correction should be made for the fact that the equilibrium temperature is higher than the temperature at which the gases leave the firebox. Mr. Vincent's figures, which confirm some of the author's earlier figures (5), show that the heat actually taken up by the firebox is larger than that computed as radiated by Equation [1] of the paper by about 12 per cent at low rates and by about 5 per cent at high rates of operation.

It might be possible to arrange a corrective formula to give better correlation between rate of heat transfer in the firebox and the radiation at the equilibrium temperature. It is however not highly important to determine the exact rate of heat transfer in the smokebox. Radiation at the equilibrium temperature gives a reasonably good qualitative picture of the distribution of heat absorption between the firebox and the flue bundle, and for the present attention is directed to the fact that the method as a whole gives a remarkably exact procedure for estimating the over-all output of locomotive boiler. Mr. Vincent in a private

communication says that he has made a number of checks of the method and has found no case in which it deviates from the measured results by more than one per cent.

Rather than strive for too high a degree of idealistic perfection, the author prefers, as stated in the preamble of the paper, to work with an admittedly empiric method that produces useful practical results.

NOTE: Computed equilibrium temperature for Test 527 Series C should be 2120 not 2320.

Rate of Temperature Change in Short-Length Round Timbers[1]

MAX JAKOB.[2] In the paper under discussion one finds sentences like the following: "Each curve....was prepared by plotting the temperatures for each heating period for the logs of different diameters." These logs are not logarithms, but real logs. Unfit for mathematical treatment as they may seem to be, the paper subjects them to such treatment. It contains and illustrates the solution of the differential equation for simultaneous axial and radial thermal conduction in short cylindrical pieces of timber which are suddenly heated and then kept at constant temperature all over the surface.

Strange enough, almost the same solution as needed and used here in conditioning timber logs was previously needed and derived for a similar practical purpose, namely, glass cooling, by Williamson and Adams.[3] The author's solution also includes the case of different thermal conductivities in axial and radial directions as occurs in timbers; this, however, causes only a slight change in the main equation.

The paper contains a great number of useful graphs, representing the temperature distribution in logs, 4 and 8 ft long, of different diameters for different time intervals. These graphs can easily be converted to be used under different conditions.

Two items in this very valuable paper are to be regretted: The author does not use the letter symbols for heat flow as recommended by the American Standards Association.[4] The American Standards symbol for thermal diffusivity, for instance, is α. The author, however, uses h^2 and q^2 for the diffusivities in the radial and axial direction, respectively, which is particularly undesirable because the American Standards employ h for surface coefficient of heat transfer and q for heat-flow rate. For time the author uses T which the Standards reserve for temperature on absolute scale as generally adopted.

The other objection is that the author, though knowing very well that diffusivity is a property of the substance in which conduction occurs, proceeds as though it were different for heating the timber in steam or water. His statement, "The diffusivity values determined from experimental tests may be considered as values which apply for the heating medium used," is likely to confuse readers who are not quite familiar with the concept of "diffusivity." It is significant that the author speaks of a "diffusivity factor for radial heating" instead of, "diffusivity factor for radial direction." What he actually has in mind is that the surface coefficient which we call h, is generally greater for condensing steam than for water in contact with the surface, and that increasing h accelerates the propagation of temperature in a solid as does an increase of the diffusivity α. The effect of an increase of h may therefore be approximated by raising the value of α. Then, however, one should call this an "apparent" or "equivalent" diffusivity and express it by a different symbol, for instance, α_e. Both lines in the author's Fig. 1 obviously represent equivalent rather than true diffusivities, since the author neglects the temperature difference between the heating medium and the surface of the timber samples.

[1] By J. D. MacLean, published in the January, 1946, issue of Trans. A.S.M.E., vol. 68, pp. 1–16.

[2] Consultant in Heat Research, Armour Research Foundation, and Research Professor of Mechanical Engineering, Illinois Institute of Technology, Chicago, Ill. Nonresident Research Professor of Heat Transfer, Purdue University. Mem. A.S.M.E.

[3] "Temperature Distribution in Solids During Heating or Cooling," by E. D. Williamson and L. H. Adams, *Physical Review*, vol. 14, 1919, pp. 99–114.

[4] American Standard Letter Symbols for Heat and Thermodynamics Including Heat Flow," A.S.A., Z10.4-1943, published by A.S.M.E.

Temperature-Time Distribution in Rectangular Bars[1]

G. M. DUSINBERRE.[2] The authors suggest the use of their curves in the calculation of cooling rates during quenching of steel. There are two difficulties here, as follows:

The first lies in the way the results are plotted, log of the temperature-change ratio against a function involving time directly. This is a widely used convention. It has the advantage, in plotting, that the lower end of each curve approaches a straight line. However, this region will ordinarily be of little interest. The authors wisely give a better chart in the same space by omitting values of T_x less than 0.02, which values are shown on many similar charts.

Consider the cooling of steel from 1600 F to 100 F. The most important changes occur around 1300 F, or $T_x = 0.8$. The logarithmic plotting constricts this part of the chart. Therefore a uniform scale of T_x would be an improvement. A further improvement would be to use $\sqrt{a\theta/w}$ as abscissa to expand the time scale in the region of practical importance.

The second difficulty lies in the use of constant values of the thermal properties (k, c_p, and h) and neglect of the latent heat of phase change. These simplifications are unavoidable under the conventional mathematical treatment, although not under a numerical treatment.

Take an experimental temperature-time curve for a steel slab or billet. There is a definite "flat" in the critical range not shown on curves such as the authors'. If, by choice of mean values of the properties, we attempt to fit a theoretical curve to this, we will find that it must be in error, perhaps 10 to 15 per cent in position at some point and even more in regard to slope. This makes any question of two- or three-figure "accuracy" rather academic.

The fact is that calculus is not adequate for this problem. If we have to get solutions by calculation rather than experiment, then the writer feels that we should use the less esthetic but more accurate finite-step methods.

AUTHORS' CLOSURE

The authors appreciate Commander Dusinberre's amplifying statements regarding the mathematical method of solution. The log plot for T_x function has become a generally adopted method of plotting this type of data. The authors realize that it is sometimes desirable to plot the results in a different way from the one they chose; hence they have included the tabular results shown in Tables 1 through 5, which may be used in plotting the curves in any desired manner.

This method of solution involving the solution of the partial

[1] By W. M. Rohsenow, M. A. Aronstein, and A. C. Frank, published in the February, 1946, issue of Trans. A.S.M.E., vol. 68, pp. 135–141.

[2] Department of Mechanical Engineering, Virginia Polytechnic Institute, Blacksburg, Va. Mem. A.S.M.E.

differential equation is limited to cases in which the thermal properties (k, Cp, and h) are essentially constant and in which there are no phase changes and no latent heat. This method may be applied to the steel slab mentioned by Commander Dusinberre provided the temperature range does not include the critical point.

Measurements of Temperatures in Metal Cutting[1]

K. J. Trigger.[2] The authors present interesting results of average chip temperatures versus cutting speed, Fig. 2 of their paper. The reasoning for the observed constancy of chip temperature at the higher range of speeds appears logical. The results do, however, emphasize the desirability of determining the temperature at the tool-chip interface, or contacting surface, since, as pointed out, this is where tool failure has its inception.

The most direct approach to the tool-chip interface temperature is probably by means of the tool-work thermocouple, which is, in reality, composed of many thermocouples in parallel so that the emf measured is the average emf of all the points of contact (thermocouples) of the tool and chip. A temperature so determined represents therefore the average temperature at the tool-chip interface, and in this discussion is referred to as the cutting temperature. The writer has been investigating cutting temperatures utilizing the tool-work thermocouple principle and using a steel-cutting grade of cemented carbide as the tool and annealed NE8640 as the workpiece.

Since the authors used carbide tools on steel it would seem that the results of the cutting-temperature tests should apply quite directly, and sample curves are submitted to supplement the authors' statements concerning the temperature of the tool.

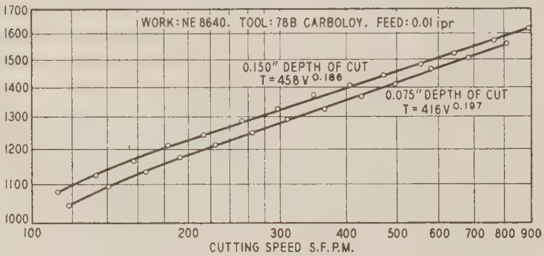

Fig. 1 Cutting Speed-Cutting Temperature Relationship

Fig. 1 of this discussion shows the effect of cutting speed upon cutting temperature for two depths of cut at constant feed. Many similar curves have been obtained, all of the same general shape.

It is generally observed that the curves droop at low speeds and/or light cuts. Examination of the chip under such conditions reveals some scuffing or roughness on the separating surface coincident with the presence of a significant built-up edge on the tool. The effect of the built-up edge would be to remove some of the thermocouples from the separating surface, and thus lower the indicated (average) emf. Supporting this belief is the fact that when the chip shows no scuffing the points lie on the straight-line portion of the curve. Such conditions have been observed in many tests to date.

Within the range of conditions herein reported, the writer believes that cutting temperature is essentially a power function of cutting speed, and is expressed by the relationship $T = CV^n$, where C is a constant and n the exponent. This contention receives support from the older and more common relationship for tool life and cutting speed, i.e., $VT^n = C$, where V = cutting speed, sfpm; T = tool life, min; C = const; and n = exponent.

Various investigators, including the authors, report constant cutting force if cutting speed is the only variable. If, then, the tool life is shortened exponentially with increased cutting speed, it is presumably due to increased temperature at the tool surface in a similar sort of relationship, that is, exponential.

Authors' Closure

Professor Trigger's comments on the paper, "Measurements of Temperatures in Metal Cutting," are very much appreciated. Professor Trigger's method of measuring the cutting temperatures between the chip and tool by using a tool-work thermocouple appears to be a logical procedure to determine the maximum temperatures developed in metal cutting. In fact, Professors Gilbert and Boston first presented a paper before the A.S.M.E. in December, 1934, entitled, "Relation Between Cutting Force Temperature and Tool Life in Cutting Steel With Single-Point Tools." The time of this presentation coincided with the depression and the paper was never published. Another paper entitled, "Cutting Temperatures Developed by Single-Point Turning Tools," was presented to the American Society for Metals[3] by Professors Gilbert and Boston. Both of these papers showed that the cutting temperature as a function of cutting speed was practically a straight line for any shape of tool, except for speeds below approximately 25 fpm, where the curve was concave downward but above the normal line. Professor Trigger's definition that the thermocouple is composed of many thermocouples in parallel is in agreement with the authors' opinion. In fact, the concave portion of the temperature-cutting speed curve for the lower speeds is presumably due to this particular principle, in that at lower speeds the built-up edge becomes larger, and actually introduces a considerable amount of worked metal between the newly formed chip and the tool face. This is one of the objections of the tool-work thermocouple. Apparently Professor Trigger's work as shown in Fig. 1 was carried out at speeds where the built-up edge did not exist or was small in size. His lowest speeds were about 120 fpm, with tool temperatures above 1000 F.

An objection to the tool-work thermocouple is the difficulty of calibration. The authors have found that the electrical properties of all tool bits are not identical, and that the calibration must be carried out between a piece of the work being cut and that particular cutting tool doing the cutting. It is also necessary to make a new calibration for each material cut. It would be interesting to know Professor Trigger's procedure in calibrating his tool-work thermocouple.

The two-tool thermocouple in which two dissimilar tool materials form the cutting edge, eliminated the need of calibrating against the material cut. The greatest trouble was in regrinding the tool, and also the sintered carbide wore faster than the other tool material at cutting speeds below 100 fpm. This two-tool thermocouple had the advantage of being able to cut any material, the emf developed between the two tools being a direct function of the temperature at the cutting edge and independent of the material cut. This permitted the use of this two-tool thermocouple to compare cutting temperatures under standard sizes of cuts when turning steel, cast iron, nonferrous metals, etc.

Later a double tool-work thermocouple was developed in which

[1] By A. O. Schmidt, O. W. Boston, and W. W. Gilbert, Trans. A.S.M.E., vol. 68, 1946, pp. 47–49.

[2] Professor of Mechanical Engineering, College of Engineering, Department of Mechanical Engineering, University of Illinois, Urbana, Ill. Mem. A.S.M.E.

[3] Trans. American Society of Metals, vol. 33, no. 3, September, 1935, pp. 703–726.

one tool of high-speed steel, and a second tool of cast nonferrous alloy, identical in shape, were used when cutting a bar at the same speed, feed, and depth of cut. The reading on the potentiometer was independent of the material being cut, the tool grinding was simplified, and the accuracy of reading was improved.

All of these types of thermocouples gave practically straight-line relationships between the cutting speed and temperature. In this respect the authors' work does not agree with Professor Trigger's, due perhaps to the fact that our temperatures are normally below 1000 F, whereas Professor Trigger's are normally above. It would therefore appear that further work on this subject to establish the relationship between the temperature and the tool life as a function of cutting speed would be well worth while.

Summary Report on the Joint E.E.I.-A.E.I.C. Investigation of Graphitization of Piping

By S. L. HOYT,[1] R. D. WILLIAMS,[2] AND A. M. HALL[3]

Following the failure in January, 1943, of a welded joint in a high-pressure steam line at the Springdale Station of the West Penn Power Company, among the several investigations undertaken, a research program was initiated at Battelle Memorial Institute to study the fundamental causes of graphitization and the restoration of graphitized joints. The present paper is a summary of the findings to date on graphitization and includes not only work of the Institute but elsewhere. It gives the present status of significant points which have emerged in relation to manufacture and fabrication.

Introduction

THE failure in January, 1943 (1)[4] of a welded joint in a high-pressure steam line at the Springdale Station of the West Penn Power Company made the whole power industry, as well as the metallurgical world, acutely aware, for the first time, that graphite could form in steel piping operating at steam temperatures, and that its presence in certain instances could so weaken the piping as to produce a service hazard.

Among the several investigations immediately begun by various interested groups was a research program at Battelle Memorial Institute under the auspices of a Joint Subcommittee on Graphitization, initiated by the Edison Electric Institute and the Association of Edison Illuminating Companies. The program was set up to study the fundamental causes of graphitization and the restoration of graphitized joints. For new pipe, it was desired to know one or more kinds of steel which would be resistant to graphitization.

In particular, interest centered about the segregated type of graphite found in the Springdale pipe. This occurred as a continuous film across the entire cross section of the pipe, at or near that zone of a welded joint which had been heated to the Ac_1 point of the steel during welding, i.e., the low-temperature edge of the weld-heat-affected zone. Instead of being plane, this film of graphite was scalloped in conformity with the heat-affected zone and in a cross section through weld and stock, showed a characteristic "eyebrow" formation. Graphite nodules might have formed elsewhere in the pipe, but with their random distribution they were thought to be relatively harmless. The segregated streaks, however, tended to form a weak and brittle plane across the entire pipe and it was imperative that their formation be prevented.

This paper is a summary of the findings to date on graphitization investigations, both at Battelle and elsewhere. These include reports by approximately 40 operating companies on the condition of their steam lines[5] and by additional investigators, principally staff members of the power companies and of suppliers who had individual points under study.

In general, no attempt is made in this paper to distinguish between the sources of the data discussed, although this has been done in the Battelle progress reports to the Joint Subcommittee.

It is intended in this summary report to give the status to date on points that have emerged as being of greatest significance in manufacture and fabrication. Fortunately, it is possible to give reasonably good answers to some of the more important questions, but it should be recognized that the exploration of a new field is likely to raise additional questions, the existence of which had not been anticipated. That is true in the present instance and is principally responsible for the tentative position which must be taken on some of the points.

Influence of Deoxidation and Alloying Elements

Aluminum Deoxidation. Among the plain-carbon steels the high-aluminum-deoxidized types (i.e., 1.5–2 lb Al per ton) were found to be relatively susceptible to graphitization at the higher steam temperatures. Plain-carbon steels, low-aluminum-deoxidized (i.e., 0.5 lb Al per ton, or less) or straight silicon-deoxidized, appear resistant to graphitization at present steam temperatures. These findings are confirmed by both plant and laboratory investigations (2). Bead-welded high-aluminum-deoxidized plain-carbon steel showed graphitization after only 500 hr at test temperatures of 925 F and 1025 F. At the same test temperatures a low-aluminum-deoxidized plain-carbon steel contained no more than possible traces of graphite after 7700 hr.

Residual aluminum in the steel, or some factor which runs parallel with it, seems to promote graphitization very strongly. In this connection there is some evidence, though it is inconclusive, to suggest that it is residual aluminum as metallic aluminum, rather than as Al_2O_3, which is effective. This point will be again considered when the relationship between graphitization and microstructure is discussed.

Molybdenum. The presence of about 0.5 per cent molybdenum in the carbon-molybdenum steel retards graphitization to some extent. However, molybdenum was originally added to improve creep strength, and its retardation of graphitization is a secondary effect, far outweighed by the action of aluminum deoxidation as an accelerator. Thus while low-aluminum-killed or straight silicon-killed carbon-molybdenum steel was found to be very resistant to graphitization, high-aluminum-killed carbon-molybdenum steel graphitized to an appreciable extent, though not as readily as high-aluminum-killed plain-carbon steel. The graphitization of carbon-molybdenum steel and the effect of aluminum deoxidation may be seen by comparing Fig. 1 with Fig. 2.

[1] Technical Adviser, Battelle Memorial Institute, Columbus, Ohio.

[2] Welding Engineer, Battelle Memorial Institute.

[3] Research Engineer, Battelle Memorial Institute.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Joint A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperature on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[5] Of the forty stations reporting, seven had cases of severe graphitization (chain-eyebrow-type or heavily segregated nodules), two had cases of moderately severe graphitization (segregated nodules), four had cases of moderate graphitization (random nodules), three had cases of slight graphitization (a few random nodules), and 23 had found no graphite.

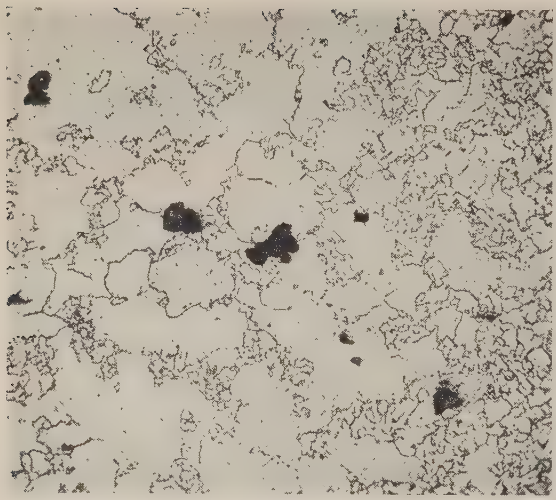

Fig. 1 High-Aluminum-Killed Carbon-Molybdenum Steel Bead-Welded and Treated 9700 Hr at 1025 F
(Random graphitization. Nital etch; ×250.)

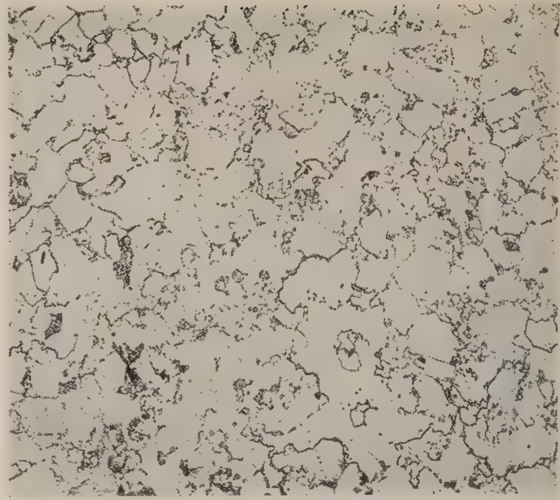

Fig. 2 Low-Aluminum-Killed Carbon-Molybdenum Steel Bead-Welded and Treated 9700 Hr at 1025 F
(No graphitization. Nital etch; ×250.)

Chromium. Examination of the chromium-molybdenum steels has developed a feature of the microscopic identification of graphite that deserves mention. Chromium is added, in this case, to produce a carbide of such stability that it will be immune to any tendency to form graphite at operating temperatures. There is considerable evidence to support this belief, and one expects the microscopic examination of heated test samples to show nothing of the nature of graphite. Actually, one observes very small particles, even in samples containing 1 per cent of chromium or more, and he is inclined to report "traces of graphite." One of these small particles is shown in Fig. 3. The occurrence of graphite is highly doubtful and the situation calls for positive identification of these particles, if not in the present samples in which the task would be relatively difficult, then in samples which have been heated long enough for them to grow into a more convenient size. Unfortunately, at this time it has not been possible to carry out the work necessary to complete the identification—they might be chromium oxide or some other unsuspected reaction or precipitation product.

With this in mind it is noted that chromium-molybdenum steels, even one containing as little as 0.25 per cent chromium and made with high-aluminum deoxidation, did not appear to graphitize in a laboratory test. After heating at 925 F and 1025 F for 5000 hr, careful metallographic examination showed so little graphite, if any, that it could not be identified with certainty. Under the same conditions, control specimens, similar in composition except that no chromium was added, graphitized after 2500 hr. Figs. 4 and 5 indicate these differences. Thus the stabilizing or retarding capacity of chromium is much greater than that of molybdenum, and is apparently sufficient to offset the accelerating tendency of aluminum.

Nickel. It has been reported that nickel in amounts up to 3.43 per cent does not increase the tendency of low-carbon molydenum steel toward graphitization in the temperature range of 950 F to 1050 F and may quite possibly decrease it.

Effect of Prior Heat-Treatment and Structure

Welding. The striking peculiarity in the graphitization problem confronting the power industry is the fact that when segregated graphite forms in pipe during service it usually occurs in a narrow zone in the pipe immediately adjacent to a weld. It is evident that the location of this zone and the degree of graphite segregation in it are largely determined by the thermal gradients imposed on the pipe during welding. Microscopic examination shows that this zone of graphite segregation is the narrow band through the wall of the pipe near the weld and following the weld contour, which has been heated approximately to the Ac_1 point of the steel by the welding operation. The influence of welding heat is shown in Figs. 6 and 7 for a high-aluminum-killed plain-carbon steel.

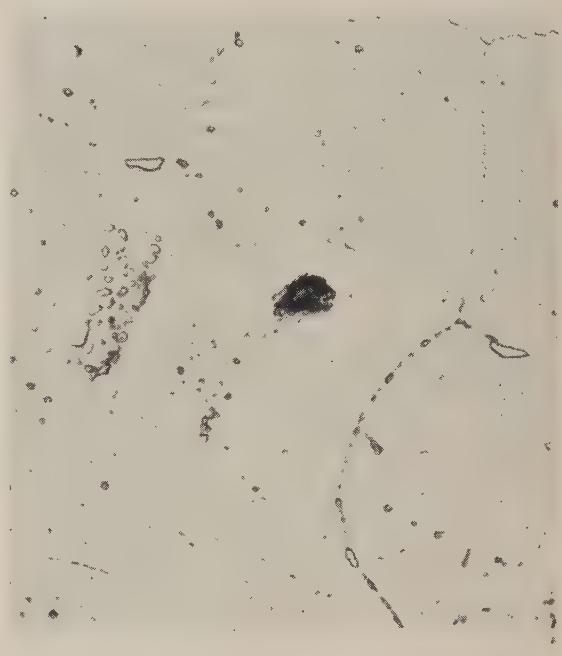

Fig. 3 Small Dark Particle in Low-Aluminum-Killed Chromium-Molybdenum Steel, 0.5 Per Cent Cr, 0.5 Per Cent Mo, Treated 2500 Hr at 1025 F; ×2000

Welding is not the only factor operating to produce segregated

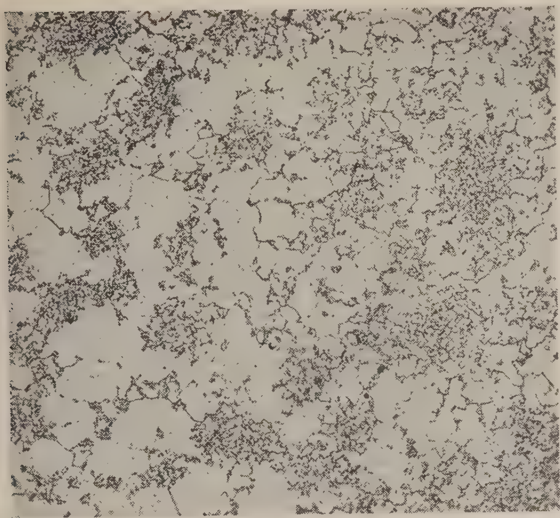

Fig. 4 High-Aluminum-Killed Chromium-Molybdenum Steel, 0.50 Per Cent Cr, Bead-Welded and Treated 5000 Hr at 1025 F
(No graphite. Nital etch; ×250.)

Fig. 5 High-Aluminum-Killed Carbon-Molybdenum Steel, 0.0 Per Cent Cr Control, Bead-Welded and Treated 5000 Hr at 1025 F
(Graphitization tending to segregate at Ac_1 isotherm. Nital etch; ×250.)

Fig. 6 High-Aluminum-Killed Plain-Carbon Steel, Bead-Welded and Treated 7700 Hr at 1025 F
(Graphitization in zone just above Ac_1 isotherm. Nital etch; ×250.)

Fig. 7 High-Aluminum-Killed Plain-Carbon Steel, Bead-Welded and Treated 7700 Hr at 1025 F
(Graphitization in stock unaffected by welding heat. Nital etch; ×250.)

graphite; we must not overlook the simple fact that the steel must be inherently graphitizable. Thus the thermal treatment accompanying welding is a necessary but not a sufficient condition for the production of a brittle zone of segregated graphite.

Carbide Particle Size. There is evidence to indicate that it is the structure of the carbides in the zone heated to the Ac_1 point which induces the graphite segregation. In particular, the observations suggest that a spheroidized or coarse-carbide structure, such as may obtain at the Ac_1 isotherm after welding, is especially susceptible to graphitization. Fine carbides appear to be more resistant to graphitization. This was brought out in a test of a high-aluminum-killed plain-carbon steel which was heated 1350 hr at 1025 F. Specimens water-quenched from 2000 F or 1600 F, or normalized at 1650 F, (all containing fine carbides) showed no graphite, while specimens annealed at 1400 F, or spheroidized 65 hr at 1250 F, had graphitized, forming random graphite nodules. Furthermore, welded carbon-molybdenum steel pipe, normalized at 1650 F, was also resistant to graphitization, presumably because the susceptible structure near the weld was replaced by a relatively fine and resistant structure. It is to be assumed that longer heating would ultimately produce random graphite, a point that is more adequately treated under the kinetics of graphitization.

Normality. The accumulated evidence seems to confirm the hypothesis that a steel which is structurally completely "normal" in the McQuaid-Ehn test is highly resistant, though not necessarily absolutely immune, to graphitization (2). In this light the action of aluminum as an accelerator of graphitization may be its known capacity for promoting spheroidization of car-

bides and for inducing "abnormality" in steel. Thus the influence of aluminum may actually arise from its effect on structure.

Contrariwise, the persistence of well-defined lamellar pearlite shown by Kerr and Eberle for long-continued heating of normal steels may be directly associated with freedom from graphitization (2).

Postweld Heat-Treatment. Postweld heat-treatments at 1200 F or 1300 F have shown a little less graphitization as compared to the as-welded condition. The reason for this small effect is not clear, though for molybdenum steel it has been suggested that the effect of 1300 F is due to the formation of molybdenum carbide (3). A striking effect, however, is produced by a postweld stress relief carried out slightly above the Ac_1 temperature of the steel which re-forms the structure produced by welding. In tests of a high-aluminum-deoxidized plain-carbon steel, stress-relieved at 1370 F, the formation of segregated graphite was completely prevented, although random graphitization was not prevented, and in fact may even have been accelerated.

The same steel, stress-relieved at 1290 F or less, after welding, developed segregated graphite near the Ac_1 isotherm. These tests were carried out at 925 F and 1025 F for time intervals up to 7700 hr.

It may be pointed out here that some instances of resorption of segregated graphite have been observed in test samples.

Creep Strength. One of the points immediately considered upon discovery of the apparently beneficial effects of stress-relieving above the lower critical temperature (Ac_1) was the effect of such a treatment on the creep strength of steel so treated. Creep tests could not be made in the limited time available, but samples of carbon-molybdenum steels were "stress-relieved" at 1425 F, and the microstructures thus produced were compared with those shown by Weaver (4), in his study of creep in carbon-molybdenum steel. This comparison suggested that the creep properties would most probably correspond to Weaver's specimens representing an annealed initial condition. As such, the stress-relieving treatment above the lower critical point would not be excessively harmful to the creep strength.

Influence of Stress and Strain

It is not possible at this time to distinguish very clearly between stress and strain (cold work) as factors in the graphitization problem. High-pressure steam lines are under stress in service and some evidence has been gathered to indicate that stress may accelerate graphitization and in so doing, produce a typical stress pattern. It is not considered a primary cause of graphitization, however, and the information at hand indicates that it is effective only in the case of a readily graphitizable steel.

As an example of the effect of cold work, a tensile bar of high-aluminum-deoxidized plain-carbon steel was pulled to fracture at room temperature and heated 1350 hr at 1025 F. Random graphitization was found in the bar, increasing in severity toward the fracture. A similar test in a low-aluminum-killed plain-carbon steel produced no graphite whatever. Straining by bending at room temperature slightly increased graphitization in a high-aluminum-killed plain-carbon steel, while compressive straining at 1025 F considerably increased the amount of graphitization. No relationship was found between the strain lines (Lüders' lines) produced in a steel specimen by tensile deformation and the location of graphite nodules.

The information so far accumulated points toward stress, and particularly that developed at steam-line operating temperatures, as a probable accelerator of graphitization which should not be overlooked.

Effect of Joint Preparation

Among the original questions raised by the power industry regarding the susceptibility of carbon-molybdenum steel pipe to graphitization, three dealt with the possible effect of joint preparation as a cause of graphitization. Specifically these were: (*a*) Is grease applied to the pipe end during fabrication a possible source of graphite? (*b*) Does gas-cutting of pipe ends preparatory to welding cause graphite to form? and (*c*) Does cold-working, such as produced by too coarse a tool feed in machine-beveling of pipe ends, cause graphite to form?

Tests indicate that none of these three factors is likely to contribute to the graphitization of steam lines. However, the evidence is not absolutely conclusive, particularly in regard to the effect of grease. Hence it appears advisable to adhere to the practice of cleaning all joints free from grease and other foreign matter prior to welding. It is known, of course, that perfectly clean steel graphitizes if it is of the susceptible type.

As for flame-cutting and heavy tool feeds, it is believed, in view of the depth of penetration of weld metal below the original surface of a joint, that neither of these surface preparations would cause the formation of graphite.

Kinetics of Graphitization

On the basis of preliminary work that has been done an attempt has been made to apply the principles of chemical kinetics to the problem of graphitization.

The approach consists in treating the graphitization process as a slow chemical reaction and determining its course with time at constant temperature, as given by the fraction of carbon transformed from the initial condition (carbide) to the final condition (graphite), for different intervals of time. The plot of the fraction of total carbon transformed, against time at temperature, is the reaction curve of the process. By correlating a series of reaction curves, each representing the progress of graphitization at a different temperature, the effect of temperature may be determined; at least for the temperature range over which the mechanism of the process remains the same.

Fig. 8 shows reaction curves obtained from data for the graphitization at 925 F, 1025 F, and 1125 F, of "reconditioned"[6] specimens of the original Springdale pipe, a high-aluminum-deoxidized carbon-molybdenum steel. The fraction of carbon transformed to graphite was determined by metallographic methods and plotted semilogarithmically against time. The solid portion of each curve represents the experimental data. The broken portions have been added to show the probable form of the complete curves.

For the purpose of illustration only it was decided to use these data to indicate the effect of temperature on the rate of graphitization. For this the half-life time at each of the three test temperatures was read directly from the reaction curves as the time required to convert one half the total carbon in the steel to graphite. These half-life times for 925 F, 1025 F, and 1125 F, plotted semilogarithmically against the reciprocal of the absolute temperature, yielded the approximately straight-line relationship shown in Fig. 9.

Reports on the progress of graphitization usually make use of some simple description of amounts, particle size, and distribution, with photomicrographs of typical areas. That is not adequate for quantitative treatment; and for the present work it was necessary to determine the amount of graphite in each sample. This was done by polishing several microsections of each sample and then estimating the average area covered by graphite. This figure in per cent gives the per cent by volume. Upon converting the volume per cent to weight per cent, it was necessary to apply a correction factor which was experimentally determined by a comparison of a limited number of microscopic determinations with direct chemical analyses for free carbon.

Some refinements in technique are still desirable but the data

[6] Air-cooled from 2200 F and normalized at 1650 F.

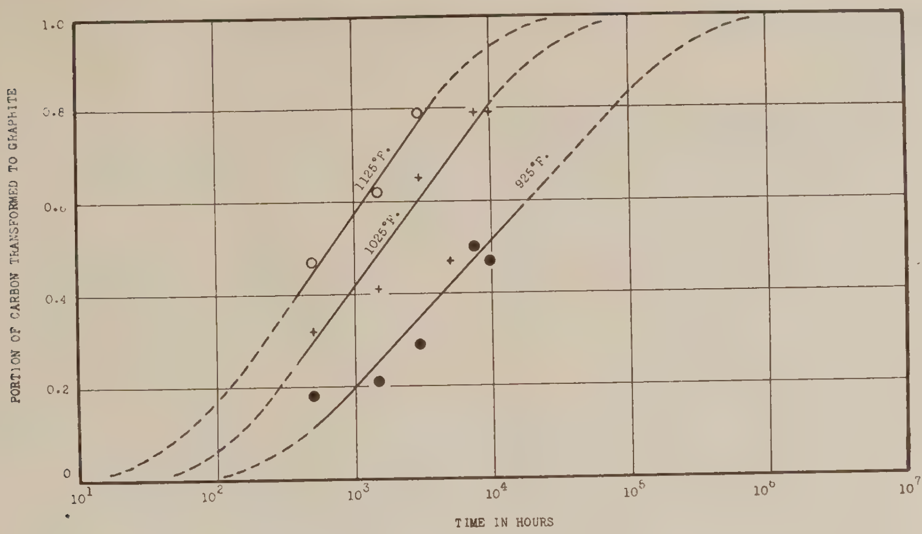

FIG. 8 REACTION CURVES FOR GRAPHITIZATION OF SPRINGDALE PIPE

presented are doubtless adequate for illustrating the principles. At this time no great accuracy can be claimed for the individual determinations. However, the data are good enough at least to suggest that the graphitization process, when it occurs, follows a definite law. Furthermore this law, whatever it may be, appears to be reasonably uniform in behavior over the range of temperature involved in the test program. Should this initial finding be corroborated by additional and more precise determinations, this is a conclusion of major significance since it gives legitimate grounds for predicting, from the findings at higher and accelerating test temperatures, what will happen at lower temperatures, or down to approximately 925 F. There is some reason to believe that at some temperature below 925 F, which would probably be different for different steels, some sort of change in the process sets in, such as one or more of the conditions for graphitization dropping out, and hence that a temperature range of immunity is entered. That temperature, if adequately determined and defined, would then become the maximum safe operating temperature for the steel in question.

While this preliminary study was conducted for the primary purpose of checking the soundness of the use of accelerating test temperatures, in other words, checking the experimental approach to the problem, it suggests, and possibly justifies, certain speculations on the type of data or observations which are needed to give an adequate understanding of the process. That should be the ultimate goal of an experimental program which seeks the control of graphitization. Since this is speculation, the discussion will be brief.

There is some thought that graphitization is a nucleation and growth process. The characteristics of such a process are: (*a*) An incubation period of nucleation over which it appears that nothing happens; (*b*) a period of relatively rapid change with corresponding depletion in the reacting substance; and (*c*) a final slow change to the end point. The graphitization data indicate that the reaction starts relatively fast and proceeds at a decelerating rate. (The curves in Fig. 8 do not show this on account of the logarithmic scale, but it becomes clear when time is plotted on a linear scale.) At least the data are consistent with the idea that nuclei are already present in the steel used for this test, though in other graphitizable steels that might not be so and the difference should show up in the reaction curves. Of

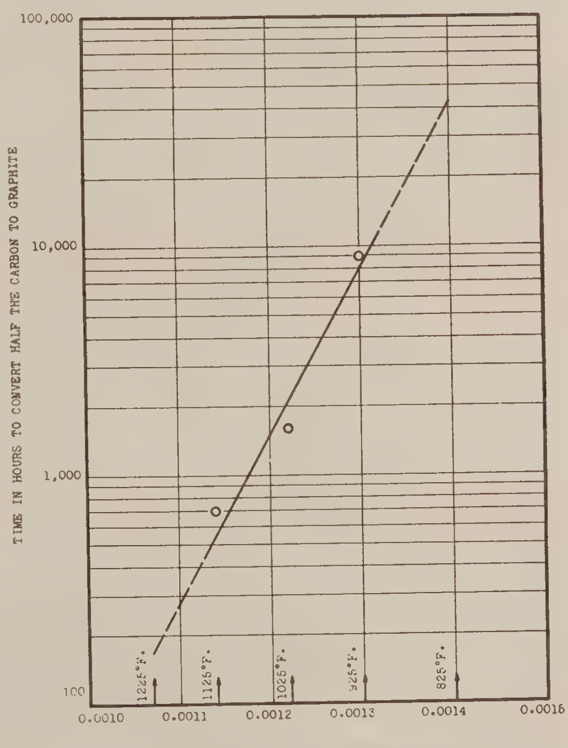

FIG. 9 EFFECT OF TEMPERATURE ON RATE OF GRAPHITIZATION PROCESS

course if the second type of steel were tested for too short a time one might erroneously conclude that it was immune.

The idea of graphite nuclei being present in some steels is not without support since Mr. Bolton of the Lunkenheimer Company has shown by chemical analysis that the Springdale steel before service contains about 0.04 per cent free carbon. Other steels

which are either immune or relatively resistant, ran about 0.01 per cent. These amounts are too small to be seen under the microscope.

An experimental verification of these points is greatly to be desired. While they may appear to be of greater theoretical than practical interest, that would probably be a shortsighted viewpoint. Laboratory tests of 10,000 hr duration are relatively long, yet an accelerated test at 1125 F, which steps up the rate thirteenfold would require over 30,000 hr to produce the effect of normal operating life at 925 F. In other words, we are required by the force of circumstances to pass judgments which are based on relatively short acquaintance with the steels in question. Obviously, the more complete our understanding of graphitization, the better able we will be to appraise steels and structures in short-time tests.

Another line of speculation which may not be without interest relates to the effect of structure, particularly the striking segregation of graphite along the Ac_1 isotherms. From thermodynamics it is known that the slope of the straight line in Fig. 9 is proportional to the energy change involved in the mechanism which controls the rate of the reaction. Calculation gives a heat value (32,900 cal per mol) which is consistent with the assumption that the controlling factor is diffusion, presumably the diffusion of carbon atoms through the ferrite matrix from the carbide source to the graphite nodules. This immediately raises the question of what carbide particles serve as sources.

In most processes of nucleation and growth it is not difficult to identify the reacting material, or source. For the case of graphitization, the malleabilization of white cast iron comes to mind, with nodules of graphite surrounded by uniform, impoverished zones. The present type of graphitization departs from this ideal picture, at least in numerous instances, in that the graphite nodules may be surrounded by carbide-containing areas. Obviously, those areas did not provide carbon for the graphite; the carbon must have come from more distant points. This seems to be just a simple statement of the facts, and if correct it shows that not all of the carbon is equally "available."

There is some evidence on availability. In the Battelle program it was found that a steel that was either quenched and drawn, or normalized, did not graphitize under test conditions which produced graphite in the same steel when it was annealed. The obvious structural difference is a fine carbide in the first two conditions and a coarse carbide in the latter. The concept of availability is sufficiently intriguing to induce one to attempt to apply it to the case of segregated graphite. Upon heating the steel to Ac_1 during welding the carbide has an opportunity to coarsen and this would probably be aided by aluminum deoxidation. If graphite nuclei are already present, graphitization would start almost immediately by drawing carbon from those carbides which were able to provide it. Even so, the process is slow and, particularly at service temperatures, some time would elapse before much visible graphite had formed. If nuclei had to form first, still more time would be required. In either case the graphite which formed at the Ac_1 isotherm preferentially would serve as a "sump" and drain carbon from coarsened carbides in the vicinity.

While this picture may be adequate for some of the cases of graphitization, a review of the findings to date indicates that it by no means covers all cases. Smith and Brambir (5) found, for example, in experiments with an end-quenched bar of graphitizable steel that the quenched end produced graphite on being heated subsequently. That seems to contradict the work at Battelle on a quenched and drawn sample, but more likely it is probably just telling a different story.

Without going into all the details, this seems to be the status of our knowledge of the graphitization process. By this we mean that from the standpoint of understanding the process, the data which have been accumulated do not fall into one simple pattern, They suggest, rather, that there are varying and multiple conditions which must be satisfied for the initiation and progress of graphitization.

Work on the kinetics of graphitization suggests that it should be valuable for distinguishing between the cases of free graphite initially present in the steel and of the spontaneous generation of nuclei. The latter condition would require a longer period for the growth of finite particles, and in the incubation period would be difficult to distinguish from the case of a nongraphitizable steel. (The determination of free graphite might make this possible.) Furthermore, it now appears that absence of graphitization at a given time interval may be due either to (*a*) absence of nuclei in an otherwise graphitizable steel, (*b*) absence of available carbon in an otherwise graphitizable steel, or (*c*) stable carbide. Conditions (*a*) and (*b*) could conceivably become modified with additional time and produce late graphitization. Condition (*c*) is the case desired for good practice, for both fabrication and service.

The ultimate use of the kinetic approach would be to make it possible to make reasonable predictions of behavior in service from accelerated test data gathered in the laboratory.

Summary

Considerable information has been gathered on the effects of several of the variables operating in the graphitization of high-pressure steam lines. Some data have been accumulated on the influence of deoxidation and the composition of the steel, on the effects of welding heat and other preservice heat-treatments, and on the influence of microstructure and cold deformation. While the results are largely qualitative, they throw some light on the mechanism of graphitization and give a basis, the only one at present, for prescribing the steel to be used and for setting safe operating temperatures.

In summarizing, it is thought that the following can be said on the points that appear now to be most important in this problem:

Deoxidation. The deoxidation practice used in making steels which are presumably of the graphitizable type is of great importance. In particular, the use of $1^1/_2$ to 2 lb of aluminum per ton is known to have a pronounced effect. This applies to both carbon and carbon-molybdenum steels of the type used for steam lines. This effect of aluminum has been found by controlled tests to be more pronounced than is the stabilizing effect of 0.5 per cent molybdenum. Steels which have been tested with not over 0.5 lb per ton and straight silicon-killed steels are relatively immune. These steels are characterized by showing a highly normal case structure in the McQuaid-Ehn test while the same steels deoxidized with the larger amount of aluminum showed the abnormal structure. While there is still some small doubt as to the absolute immunity of these normal steels, they should be regarded as highly resistant and capable of operating under more severe conditions without graphitizing than the same steels of the abnormal type. (No. = 0.001 ± per cent; Ab. = 0.020 + per cent Al).

Composition. Since carbon is the element which produces the graphite, it would seem advisable to keep it on the "low side," although no quantitative determinations have been made of the effects of different carbon contents. Chemical analysis reports a small amount of free carbon in these steels as manufactured. The indications to date are that the nongraphitizing steels contain not over 0.01 per cent free carbon and the graphitizing steels about 0.04 per cent. This lead is so important that it should be followed up. This might be done by determining free carbons on new lots of both wrought and cast steels for information purposes.

Manganese and silicon are both known to affect graphitization in general. In the present case these elements have been held at fairly constant levels and to date no specific effects of these elements have been uncovered.

Molybdenum is known to form a (metallic) carbide and has been shown to exert a stabilizing effect on pipe steels. Molybdenum also has a characteristic effect on the structure of steel and may have a secondary effect of this type. Just what this effect might be has not been worked out as yet, certainly not quantitatively.

It has been reported that heating the molybdenum steels at about 1300 F has a pronounced beneficial effect on graphitization. It was suggested that this is due to the conversion of iron carbide to the more stable molybdenum carbide, but apparently it is not known that this molybdenum carbide would remain stable at the lower operating temperatures or that the immunization is permanent.

Chromium also forms a highly stable carbide and is now thought to be the best element to add to carbon or carbon-molybdenum steels to keep them from graphitizing in service. In contrast to molybdenum, the addition of as little as 0.25 per cent chromium has at least checked graphite in high-aluminum-killed steel. While tests of chromium-containing steels have not as yet given definite or final answers, particularly on the amount of chromium which would be needed, the tests do indicate that chromium prevents or vastly retards graphitization. In this connection, however, it should be mentioned that tests of steels containing as much as 1 per cent chromium, or more, have shown "traces" of some new substance. Since one is looking for graphite in these samples, he is inclined to refer to this material as graphite since it occurs as very tiny dark specks. On the other hand, this material has not been identified, a determination which would not be without difficulties.

Welding. While random graphitization can occur in pipe steels without causing undue alarm, the heat effects of welding establish a highly critical zone where the stock has been heated to about the Ac_1 temperature. This is the major cause of segregated graphite.

While the usual stress-relieving temperatures of 1200 to 1300 F retard graphitization, it seems to be necessary to eradicate the structure of the heat-affected zone to have any important effect on the elimination of segregated graphite. Such a stress-relieving would be carried out just above the critical temperature for a long enough time to accomplish this. This treatment should be regarded as a good "corrective measure" for use with a known or suspected graphitizable steel, but by no means as a substitute for a stable steel.

Inasmuch as welding sets up the structural conditions which invite graphite segregation, welding techniques should be looked for to ameliorate this condition. The use of a high heat input and preheat which widen the heat-affected zone should be beneficial and should improve the effectivenss of stress-relieving above Ac_1.

Prior Structure. Prior structure of the part as welded does not appear to have a dominating effect on graphitization. While the point has been kept in mind by investigators, it does not appear to warrant very much attention. Even so it might have a secondary effect on the susceptible structure which is set up by welding. With this factor, as with various other factors of graphitization, an analysis of its precise effects is far less important than finding a steel which is immune to graphitization and not sensitive to deoxidation, welding, structure, etc. In particular, it is not thought that the structure set up by hot-upsetting, hot-forming, etc., is of primary importance in the graphitization problem.

Cold Work. The manufacture of pipe and castings for steam lines does not involve cold work except scarfing and similar operations. On this account the effects of cold work do not enter into the problem in a practical way, but laboratory tests have shown that cold deformation of graphitizable steels accelerates the process. On this account it might be used in a testing program to distinguish between graphitizable and nongraphitizable steels.

Service Temperatures. In view of the present interest in higher steam temperatures it is advisable to point out that the various investigational programs have all related to service temperatures not above the 925 F to 950 F mean-operating-temperature range. Steam temperatures of the order of 1000 F and above would very likely call for a different type of steel.

Suggestions for Power Industry. (*a*) For new steam-line installations it is suggested that a "normal" (low-aluminum-, or silicon-killed) carbon-molybdenum steel with initial graphite content of not more than 0.01 per cent, or a chromium-molybdenum steel containing 0.50 to 1 per cent chromium and 0.50 per cent molybdenum be used. Also, since postweld stress-relieving is general practice, it appears advisable to stress-relieve these steels above their Ac_1 temperature after welding. Future work may show this to be unnecessary.

(*b*) Though joints exhibiting some graphitization in service can generally be "healed" by normalizing at 1650–1700 F, they should be checked periodically.

(*c*) When a joint is severely graphitized, the only safe procedure is to cut it out and reweld, or replace it with a spool of new pipe. In fact, where severe graphitization is encountered in a large number of joints throughout the installation, it is safest to replace all the pipe with a type resistant to graphitization.

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Furthermore, the authors wish to acknowledge the data from other research investigations and operating company reports made available to the Joint Subcommittee from which they have drawn freely.

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Discussion

ARTHUR MCCUTCHAN.[7] This paper is an excellent summary of

[7] Engineer, Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

the findings both at Battelle and elsewhere. The tentative position taken in answering some of the important questions is justified by incompleteness of the findings and the different conclusions possible from the data at hand.

From a comparison of microstructures the authors have concluded that a postweld heat-treatment just above the lower critical temperature will not be excessively harmful to the creep strength. It is hoped that subsequent creep tests will support this conclusion. Examination of weld-probe samples from both high- and low-aluminum-deoxidized carbon-molybdenum welded joints after a few years' service at 900 F shows that a definitely spheroidized structure has resulted. It would seem important therefore to have creep tests and stress-rupture tests made on this spheroidized material for comparison with service creep measurements.

Efforts to reproduce the "strain-line" type of preferential graphitization apparently were not successful. The importance of understanding the cause of this type of graphitization is so great that further efforts should be made to reproduce this phenomenon under laboratory conditions.

While the structure produced by hot-upsetting may not, as stated by the authors, be of primary importance in the graphitization problem, it is the writer's opinion that the segregated type of graphite found adjacent to the welds and in the strain lines at Springdale is in some manner related to hot-upsetting.

I. A. Rohrig.[8] The data embodied in this report represent a vast amount of work and constitute a fundamental contribution that can be utilized in the selection and specification of new high-temperature-steam pipe material.

In association with graphitization, the effect on the physical strength of infected joints that are in service is of the greatest importance. In an endeavor to evaluate the effect of graphitization, two 10-in-diam welded joints in medium-carbon steel A.S.T.M. A-106 Grade B pipe that had been in service at 825 F for 59,000 and 64,000 hr, respectively, were investigated by the writer's company. Each weld joined a corrugated filler at the downstream side and a pipe bend at the upstream side. The graphitization that had occurred at these joints was judged to be "moderate" in amount and is shown in Fig. 10 of this discussion. Tensile tests made on samples taken from the joints indicate that the graphitization had no adverse effect on the tensile strength. Notched-bar tests made on samples cut from the two joints showed, however, a reduction in shock resistance in three of the four graphitized areas tested.

The results from these tests are given in Table 1 of this discussion. Samples from the same joints were laboratory-heated at temperatures that were varied between 850 and 950 F for a period of 6000 hr to develop further graphitization. Typical graphitization found in these samples after laboratory-heating is shown in Fig. 11. Since only small samples were used, notched-bar tests were not made after laboratory-heating, but bend tests made on strips cut from these laboratory-heated samples resulted in a distinct crack at one side of one weld as shown in Fig. 12. These samples were taken from welds in vertical piping, and it is interesting to note that the most severe graphitization at each weld occurred at the lower or downstream side of the weld.

The results obtained for these joints after laboratory-test-heating of samples removed from service show that although only moderate graphitization had developed as a result of 60,000 hr of service, a critical type of graphitization might possibly develop in these joints as a result of somewhat longer service.

The results presented by the authors regarding stress-relieving treatments above the lower critical temperature are of particular interest since heat-treatment within or above the critical-temperature range after welding offers a means of overcoming dangerous graphitization. A graphitization test is under way at the writer's company, using welded samples of three different heats of A.S.T.M. A-206 low-aluminum carbon-molybdenum pipe material in the normalized, and also in the hot-rolled and drawn condition. Some of the samples received a treatment of 2 hr at 1400 F, following welding. The graphitization test is being conducted using laboratory-test-heating and temperatures varying between 950 and 1150 F. The results obtained after 1500 and also after 2500 hr of test are given in Table 2 herewith. In one of the heats under test, namely, heat 10,600, the 1400 F stress-relieving treatment appears to be effective in preventing graphitization, whereas in heats 3099 and 6347 graphite was found in the samples that received the 1400 F treatment. Further study is to be made of the time-temperature relationship of various heat-treatments in that temperature range.

Each of the three heats was judged to be "normal" in the McQuaid-Ehn test. Heat 6347 had the most clearly defined normal structure and was classed "Type 1," while the other two

[8] Research Department, The Detroit Edison Company, Detroit, Mich.

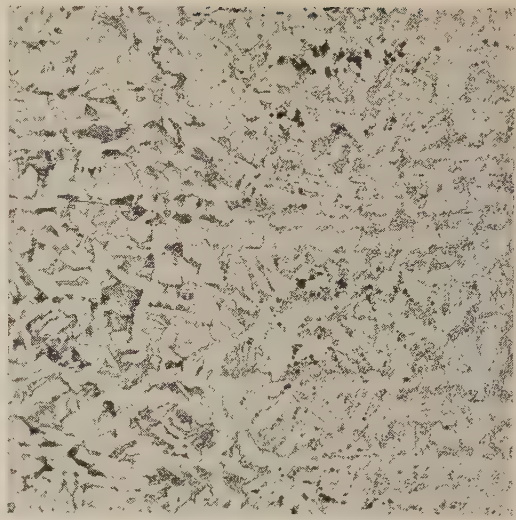

Fig. 10 Moderate Graphitization in Heat-Affected Area of Weld in Medium-Carbon Steel Pipe After 59,000 Hr of Service at 825 F; ×100

TABLE 1 NOTCHED-BAR DATA FOR WELDED JOINTS IN MEDIUM-CARBON STEEL AFTER HIGH-TEMPERATURE SERVICE

	Charpy V-notch toughness, ft-lb			
	Pipe metal unaffected by welding		Heat-affected area containing graphite	
Joint No. 1				
After 59,000 hr of service at 825 F: Corrugated filler	41		34	
	41		22	
	44	42	35	30.3
Pipe bend	72		78.5	
	72.5		86	
	76	73.5	102	88.8
Joint No. 2				
After 64,000 hr of service at 825 F: Corrugated filler	59		46	
	43		35	
	46	49.3	55	45.6
Pipe bend	103.5		81	
	65		77	
	69.5	79.3	57	71.6

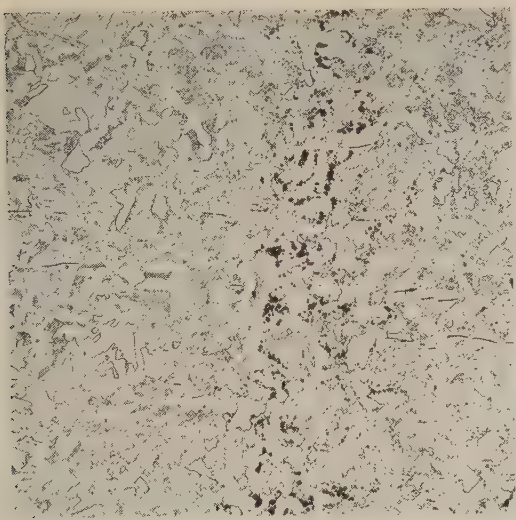

FIG. 11 GRAPHITIZATION AFTER 59,000 HR OF SERVICE FOLLOWED BY 6000 HR OF LABORATORY-TEST-HEATING: ×100

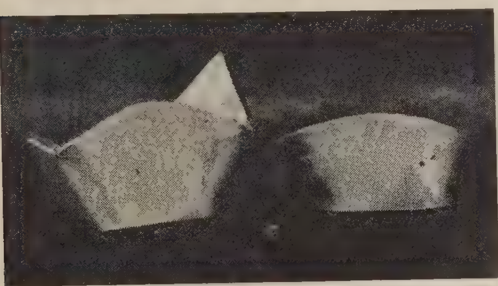

FIG. 12 BEND SPECIMENS OF WELDED SAMPLES THAT WERE LABORATORY-HEATED AFTER REMOVAL FROM SERVICE

TABLE 2 GRAPHITIZATION TEST OF LOW-ALUMINUM CARBON-MOLYBDENUM PIPE MATERIAL

(Test temperature varied between 950 and 1150 F)

	Hot-rolled and drawn			Normalized 1725 F		
	As welded	S-R 1200 F	S-R 1400 F	As welded	S-R 1200 F	S-R 1400 F
Heat 3099						
1500 hr...	None	None	Little	None	None	Little
2500 hr...	Little	Little	Increased	Little	Little	Increased
Heat 6347						
1500 hr...	None	None	None	None	None	None
2500 hr...	None	None	Trace	None	None	Trace
Heat 10,600						
1500 hr...	Little	Little	None	Little	None	None
2500 hr...	Increased	Increased	None	None	None	None

heats were classed as "Type 2." At the end of 2500 hr of test the least amount of graphite was found in the samples of the steel classed as having Type 1 normality.

With respect to the influence of strain in promoting graphitization, as given in the paper, the results obtained from an examination of stress-rupture samples may be of interest. Stress-rupture samples of low-carbon carbon-molybdenum, and chromium-molybdenum steel that had been tested at Purdue University in 1200 F steam, were examined for graphitization by the writer. These samples have been discussed in detail in a paper by J. T. Agnew, G. A. Hawkins, and H. L. Solberg.[9] The samples were obtained through the courtesy of Professor Solberg and were examined for the presence of possible graphitization and for determination of their response to the McQuaid-Ehn test.

Six specimens of each of the three types of steel were examined. Although each of the three steels was highly "abnormal," which indicated that graphitization might have been expected in the carbon and carbon-molybdenum samples, no clearly defined graphite was found in any of the samples. While this result is different from that reported in the paper by Dr. Hoyt, it should be remembered that the samples obtained from Professor Solberg, and here referred to, were tested at 1200 F instead of at 1025 F. The range of stresses under which the samples were tested as well as the hours of test are shown in Table 3 of this discussion. The absence of graphitization in the carbon and carbon-molybdenum samples, even though abnormal, may have been due to lowering of the Ac_1 temperature for the steels as a result of stress at the relatively high temperature of test. Because of its alloy content it is unlikely that graphitization would occur in the chromium-molybdenum samples.

In other work a probable indication of graphitization as influenced by stress was found in a failed low-carbon superheater tube. Creep failure had occurred as a longitudinal crack at the front of the tube due to overheating. It was interesting to note

[9] "Stress-Rupture Characteristics of Various Steels in Steam at 1200 F," by J. T. Agnew, G. A. Hawkins, and H. L. Solberg, Trans. A.S.M.E., vol. 68, 1946, pp. 309–315.

TABLE 3 STRESS-RUPTURE TEST IN 1200 F STEAM

	Type of steel	Stress during test, psi[a]	Duration of test, hr[a]	Result
1	Low carbon[b].........	4740	4620	No graphite
2	Low carbon[b].........	5770	3588	No graphite
3	Low carbon[b].........	6800	857	No graphite
4	Low carbon[b].........	7000	1261	No graphite
5	Low carbon[b].........	12500	10	No graphite
6	Low carbon[b].........	16000	27	No graphite
7	Carbon-0.50 Mo[b].....	6900	7590	No graphite
8	Carbon-0.50 Mo[b].....	7400	1300	No graphite
9	Carbon-0.50 Mo.[b]....	7800	3047	No graphite
10	Carbon-0.50 Mo[b].....	9100	312	No graphite
11	Carbon-0.50 Mo[b].....	10700	696	No graphite
12	Carbon-0.50 Mo[b].....	16000	144	No graphite
13	2-1/4 Cr-1 Mo[b].......	8400	1519	No graphite
14	2-1/4 Cr-1 Mo[b].......	8400	1908	No graphite
15	2-1/4 Cr-1 Mo[b].......	9200	2885	No graphite
16	2-1/4 Cr-1 Mo[b].......	10400	1114	No graphite
17	2-1/4 Cr-1 Mo[b].......	11300	226	No graphite
18	2-1/4 Cr-1 Mo[b].......	14400	46	No graphite

[a] Purdue University data.
[b] "Abnormal" in the McQuaid-Ehn test.

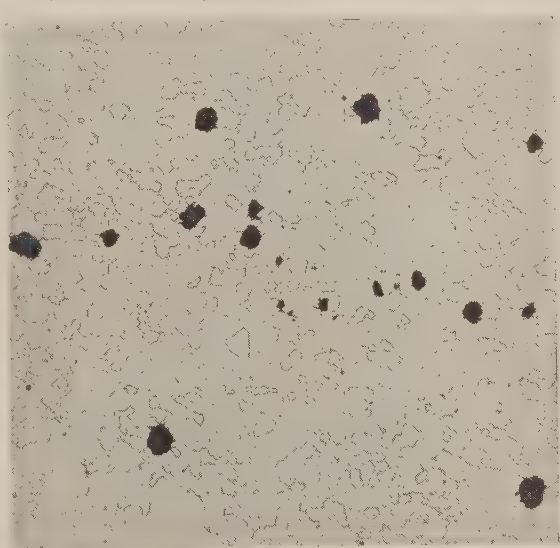

FIG. 13 GRAPHITE IN A LOW-CARBON STEEL, 1³/₄-IN-OD SUPERHEATER TUBE; ×100

that graphitization had occurred parallel to the failure and at a short distance at either side of the failure where it appeared to be localized as a result of probable critical stressing of the tube wall. Graphitization found in the tube is shown in Fig. 13 of this discussion. This is believed to be one of the few instances reported of the occurrence of graphitization in tubing smaller than 4 in. diam.

Authors' Closure

The authors appreciate Mr. McCutchan's thoughtful analysis of the present status of the graphitization situation. The determination of the creep strength of pipe steel in various structural conditions does not come within the scope of the investigation of graphitization, but they concur in Mr. McCutchan's comments. As for possible effects of hot-upsetting, we have examined many samples which compared structures produced by heating the steel to 1650 and 2200 F, respectively, and have observed no significant difference in graphitization. Therefore it seems unlikely that the structure from hot-upsetting is significant though, of course, there may be other effects of that operation which are.

The discussion of Mr. Rohrig is an important contribution and is greatly appreciated. Referring to the stress-relieving treatment just above the critical temperature, that was devised to eliminate the structure of the heat-affected zone and hence to remove what was thought to be the particular reason for the highly segregated type of graphite. The treatment at about 1400 F appears to be successful, though our experiments have shown that the random type of graphitization may actually be accelerated. Of course, it is only a corrective treatment and would not be needed with the right kind of steel. The other data which are contributed by Mr. Rohrig are all pertinent to the broad problem and are a welcome addition to the E.E.I.-A.E.I.C. program.

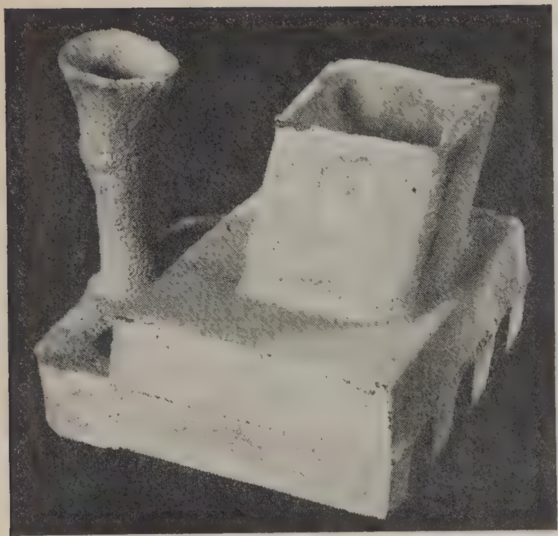

Fig. 1 4-Keel Test-Block Casting Showing Gate and Riser

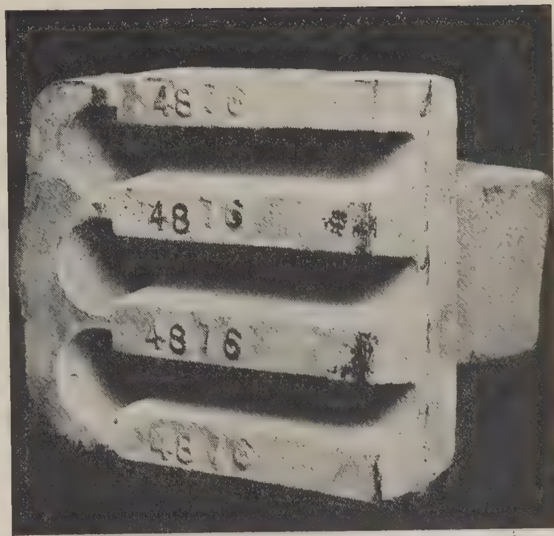

Fig. 2 4-Keel Test-Block Casting, Drag Side, Showing Coupons Measuring Approximately 1 × 1 × 6 In.

Studies on Susceptibility of Casting Steels to Graphitization

By J. J. Kanter,[1] Chicago, Ill.

This is a progress report for project No. 29 of the Joint A.S.T.M.-A.S.M.E. Committee on Effect of Temperature on the Properties of Metals. One of the activities undertaken by project No. 29 has been the study of certain aspects of the graphitization problem applying to the carbon-molybdenum casting steels used in welded piping structures. The steels so involved are of the types covered in A.S.T.M. Specifications A217 covering Pressure Alloy Castings Suitable for Fusion Welding such as valves, flanges, fittings, and other pressure-containing parts for high-temperature services. This investigation was undertaken in co-operation with the Manufacturers Standardization Society (M.S.S.) of the Valve and Fitting Industry, through a committee representing the constituent manufacturers engaged in the founding and manufacture of steel valves, flanges, and fittings.[2]

A program of tests was adopted by a group of manufacturers[2] from the valve and fitting industry for comparing the graphitization susceptibilities and general high-temperature stability of weldable grades of alloy cast steel using test castings in the form of a 4-keel block. This block, used by all of the investigators, is cast in green sand with the keels in the drag position, each fed from an end and through a risered mass above the keels as illustrated in Figs. 1 and 2. Fig. 1 shows the orientation of the casting as it is made, while Fig. 2 is a view of the four keels and the manner in which they are fed is more apparent.

The keels were removed from the blocks by sawing after heat-treatment, consisting of normalizing and drawing. The sawed surfaces of these keels were then machined flat. Upon the machined surface was deposited a single bead weld as illustrated in Fig. 3. The procedure in laying on the bead was essentially that established at Battelle Memorial Institute for the studies sponsored jointly by E.E.I.-A.E.I.C. The steel test coupon is placed in water at 60 to 65 F, at a level $^1/_4$-in. below the face machined for depositing, using end slugs for starting and finishing the bead. Using a $^5/_{32}$-in-diam type 7010 (C-Mo 0.50) electrode and a travel of approximately 6 ipm, the bead is deposited with direct-current negative polarity, at 25 to 30 volts and 125 to 135 amp.

After deposition of beads, the test specimens were given a stress-relieving heat-treatment at 1200 F. A cross section of a bead so deposited is illustrated in Fig. 4, showing the structure gradient between heat-affected zone and parent metal thus provided for study of susceptibility to segregated graphitization upon aging.

Drawing upon the experience at Battelle Memorial Institute in the E.E.I.-A.E.I.C. studies, aging treatments on the weld-bead cast-steel specimens have been made at 1025 F to attain an accelerated indication of graphitization susceptibility. Cur-

[1] Materials Research Engineer, Crane Company. Mem. A.S.M.E.

[2] The various manufacturers contributing steel test castings and investigational services include the following: Chapman Valve Manufacturing Company, Crane Company, Edward Valve and Manufacturing Company, Inc., Lunkenheimer Company, Powell Company, Reading-Pratt & Cady Division, Walworth Company.

Contributed by the A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperature on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., November 26–29, 1945, of The American Society of Mechanical Engineers.

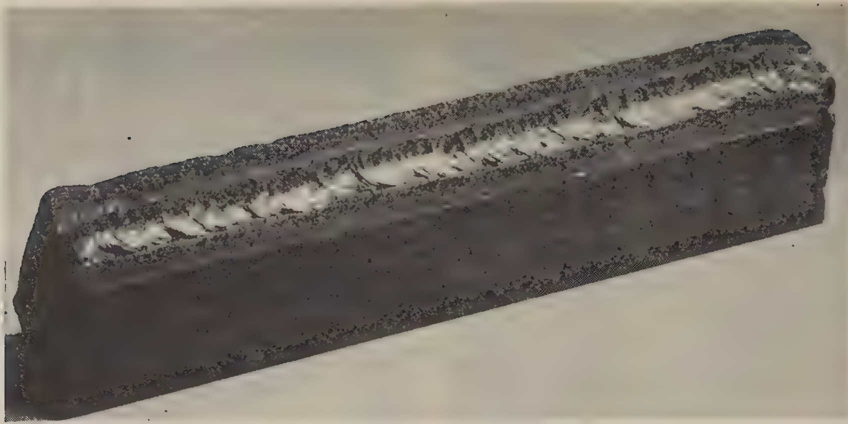

FIG. 3 WELD-BEAD SPECIMEN FOR GRAPHITIZATION TEST

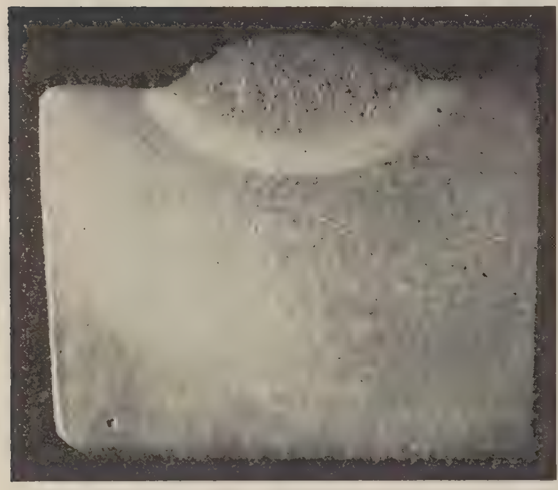

FIG. 4 ETCHED SECTION FROM SPECIMEN SHOWN IN FIG. 3 AT 3 DIAM

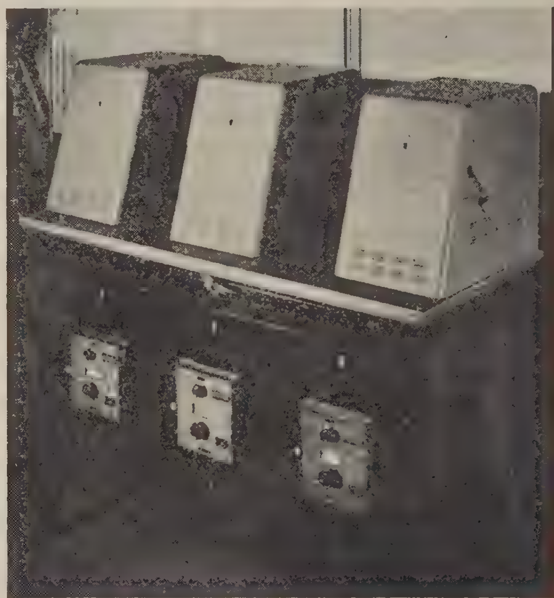

FIG. 5 ONE OF FOUR INSTALLATIONS OF FURNACES IN USE FOR AGING WELD-BEAD SPECIMENS SHOWN IN FIG. 3

rently, a number of studies in these series of tests are available for aging periods up to about 6000 hr and the specimens are continuing to age in furnaces at four of the co-operating laboratories. One of the furnace installations being so used exclusively for graphitization agings at 1025 F is shown in Fig. 5.

The original plan of the M.S.S. group was to survey the difference in graphitization susceptibility and stability of high-temperature properties in the carbon-molybdenum cast steels as made by various electric melting processes, namely, acid arc, basic arc, induction. Test heats by these processes were separately made using two types of deoxidation practice, including silicon addition, and silicon addition plus 2 lb aluminum per ton.

Almost unanimously, the co-operators reported difficulty and inability to obtain test-block castings free of "pinholes" using the straight silicon addition and it did not appear practicable from the standpoint of steel-pressure-vessel founding to consider such a practice. It was therefore decided to add another series of test materials using silicon addition plus $^1/_2$ lb aluminum per ton, which produced reasonably satisfactory test-block castings. Through this additional series of test castings it was expected that a variety of residual metallic aluminum contents would be obtained among the various heats, which would afford a basis for correlation with graphitization susceptibility.

Each co-operator furnished another set of test blocks by one or more of the electric melting processes containing 0.5 per cent of chromium in addition to the 0.5 per cent of molybdenum. These chromium-bearing castings were all made using the silicon plus 2 lb per ton aluminum-killing practice.

RESULTS OF STUDIES

The results of the studies from the cast-steel test bars upon observation after agings up to 6000 hr seem to bear out the already well-known contention that aluminum additions to carbon-molybdenum steels make for graphitization susceptibility as will be apparent from the comparison of photomicrographs, Fig. 6. A companion McQuaid-Ehn carburized structure is shown on this chart for each of the aged structures. It becomes further apparent that these tend to substantiate the Kerr-Eberle con-

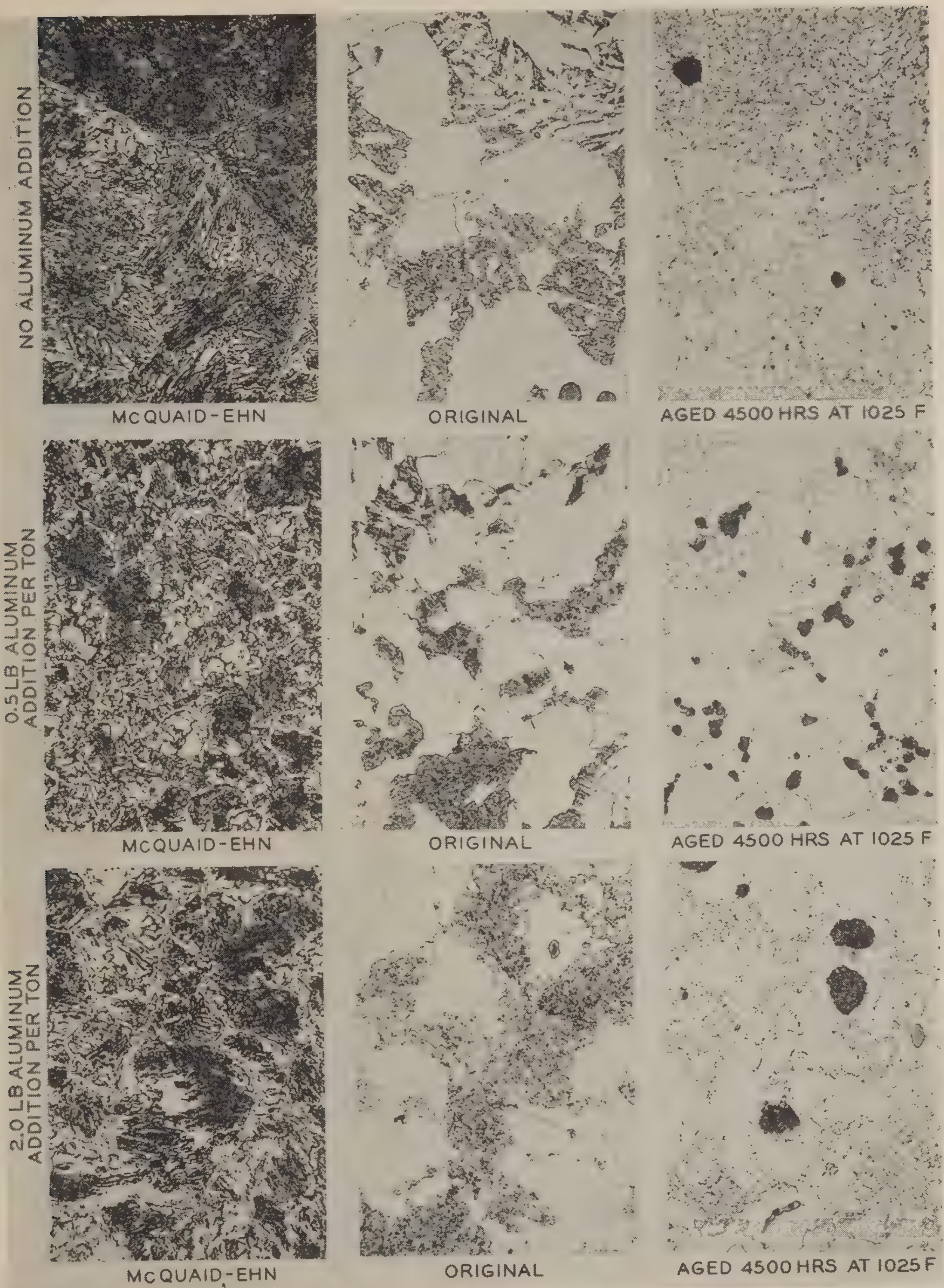

Fig. 6 Comparisons of Typical McQuaid-Ehn Carburized Structures for Carbon-Molybdenum Cast Steels With Structures at Heat-Affected Zone at 500 Diam

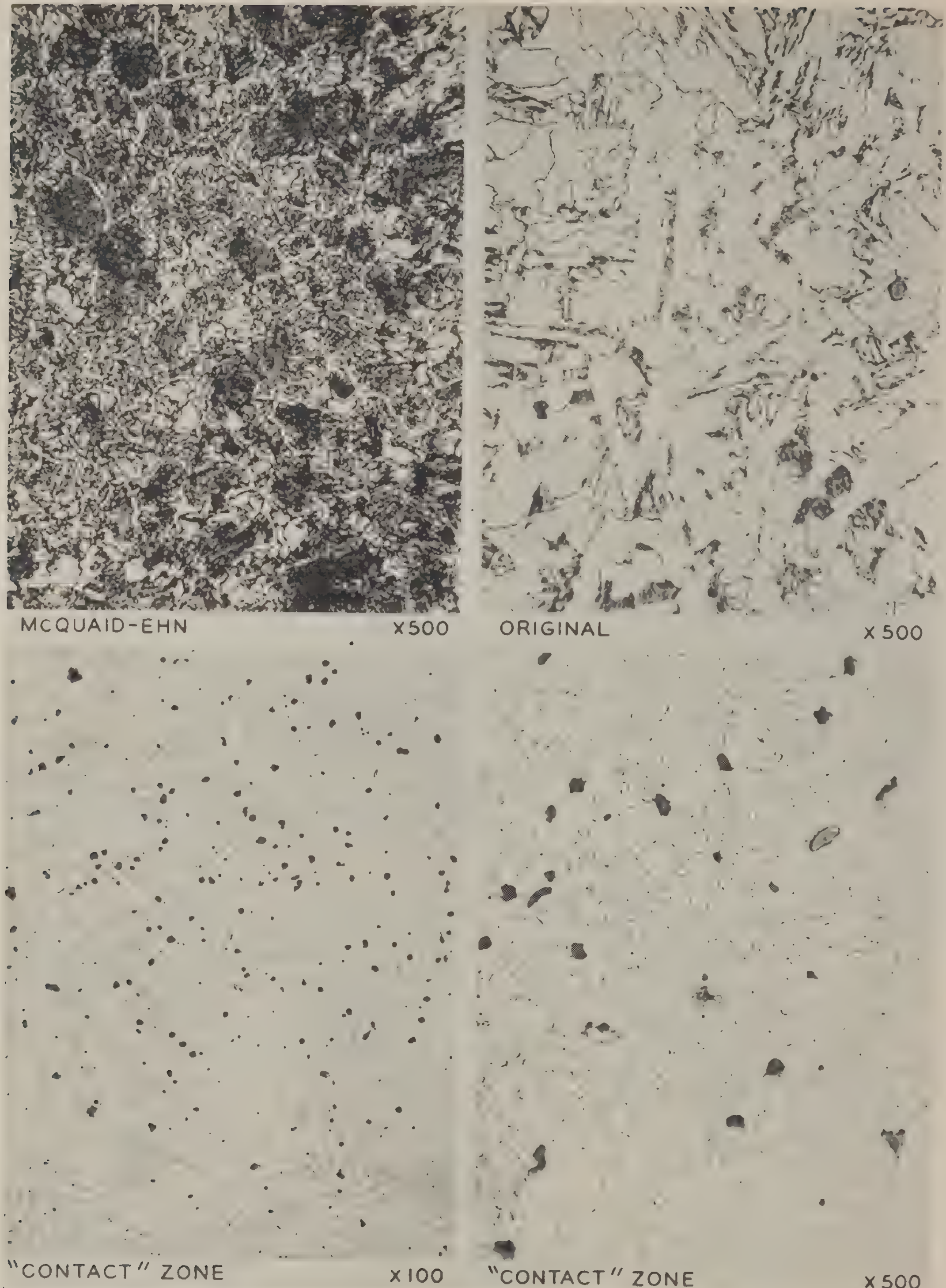

Fig. 7 Graphite Developed at Heat-Affected Zone of Carbon-Molybdenum Cast Steel With 2 Lb of Aluminum Addition per Ton, Normalized After Welding, Aged 4500 Hr at 1025 F

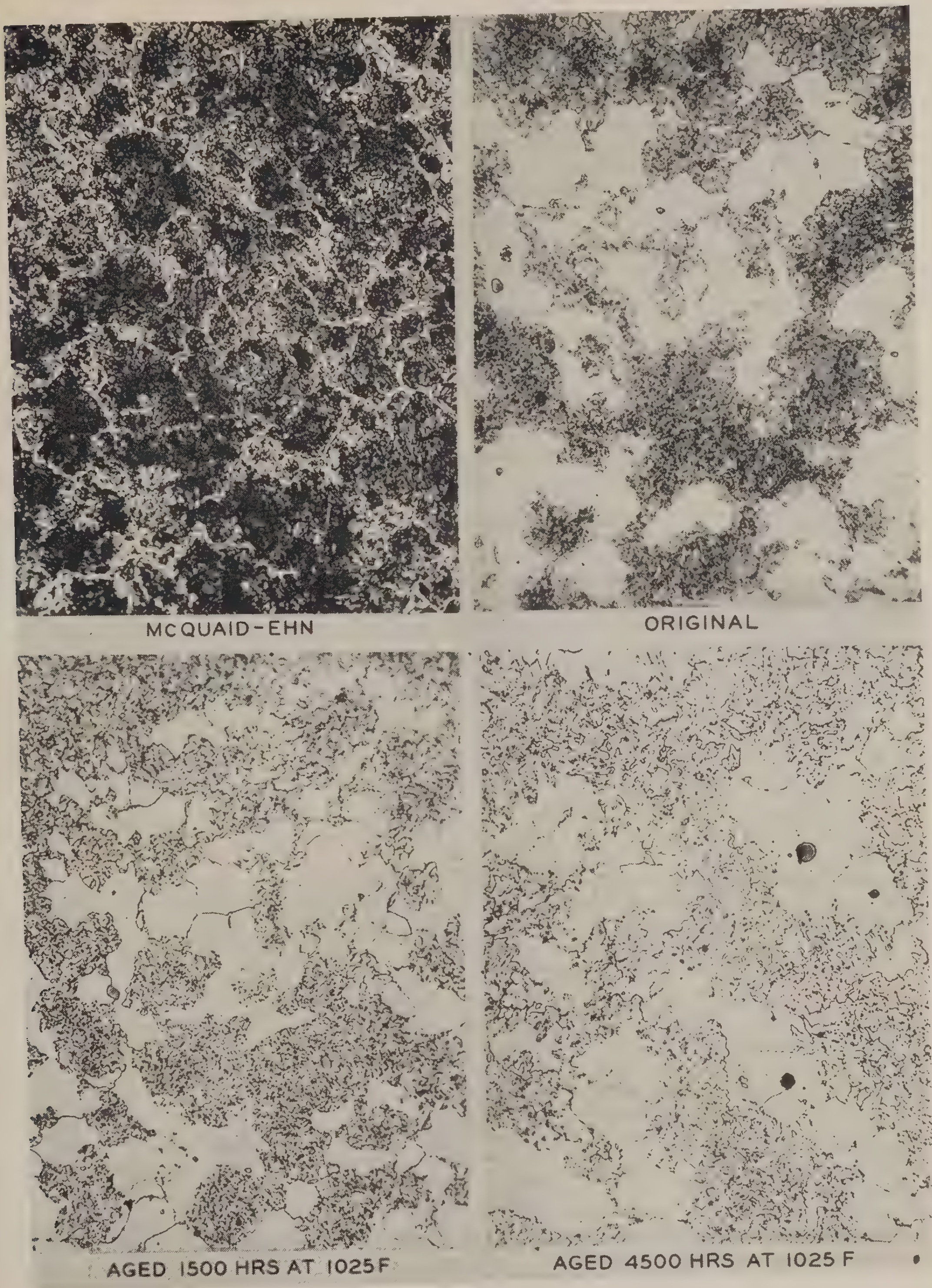

FIG. 8 TYPICAL STRUCTURES AT 500 DIAM OF CHROMIUM-BEARING CARBON-MOLYBDENUM CAST STEEL WITH 2 LB OF ALUMINUM ADDITION PER TON, AT HEAT-AFFECTED ZONE BEFORE AND AFTER AGING AT 1025 F

tention that "abnormal" structure by this criterion is an index of graphitization susceptibility in aluminum-treated carbon-molybdenum steels. This rule, however, does not necessarily apply to the steels having chromium additions. Study of microstructures indicates that the particular kind of electric-furnace process, all other factors being equal, is a minor influence as regards graphitization susceptibility.

One generalization which seems to apply with regard to the tests under discussion as well as much other observation, is that the inception of spheroidization agglomeration or coalescence of the carbide precedes its breakdown to graphite. A number of examples have been observed where in a single microfield, one spheroidized-carbide area will show graphite nodules, whereas an adjacent lamellar-carbide area will show no signs of graphite. Where chromium or other alloying ingredients prove effective in preventing graphitization under a certain condition of thermal history, the inhibition seems to be associated with resistance to spheroidization of the carbide. The susceptibility of aluminum-treated steel to graphitization seems to be associated with an accelerated tendency toward spheroidization.

In previous reports by the Joint A.S.T.M.-A.S.M.E. Committee, it has been apparent that strong aluminum treatment of steels led to easy spheroidization and consequent lack of creep resistance.

Proposals have been made to the effect that graphitization at the margin of the heat-affected zone or "contact zone" might be forestalled, or at least reduced to an impotent nature by a grain-refining normalizing heat-treatment subsequent to the welding operation. That such a treatment may not be effective in the case of a susceptible cast steel is evidenced by the micrographs, Fig. 7, comparing one of the weld-bead specimens normalized after deposition. It is seen that "contact zone" graphite has not been inhibited in this specimen. The transformation resulting from welding in the heat-affected zone seems to have some persistent effect, although it does appear that the zone of segregated graphite is somewhat broader, diffuses, and perhaps requires longer aging to detect. Studies upon more specimens normalized after welding are needed to clarify the effect.

Chromium in Cast Steels

In so far as the cast-steel studies have progressed (3000 to 5000 hr of aging), no definite evidence of graphitization has become apparent in any of the samples containing chromium in the range of 0.43 to 0.70 per cent as witnessed by Fig. 8, where relatively spheroidization-resistant carbides are well defined. It should be noted that the lowest chromium content is 0.43 and that the time periods involved are as yet relatively short and inconclusive.

It may eventually be determined that resistance to spheroidization is a necessary and perhaps sufficient condition to prevent graphitization over an expected service period.

Specimens of a number of the cast steels upon which the graphitization studies are being made are aging for the eventual purpose of making creep-strength studies on structures representing various stages of carbide deterioration. It has become quite apparent that graphitization has occurred in some of the coupons being aged for creep tests after advanced spheroidization of the carbide had developed. In the case of carbon-molybdenum steel with 2 lb per ton of aluminum addition, well-developed graphite is apparent after about 4000 hr of aging at 1025 F. The coupons containing 0.50 per cent chromium have shown no definite evidence of graphite after 5000 hr of aging. The committee has decided to continue the aging of all coupons for creep tests up to at least 10,000 hr before considering the starting of creep-rate determinations upon any of them.

Discussion

I. A. Rohrig.[3] This report shows that under suitable conditions graphitization may readily occur in cast as well as in wrought carbon-molybdenum steels. The observation of an apparent "persistent effect" resulting in segregated graphite in the heat-affected area of a weld even after normalizing is of particular significance because normalizing has appeared to be the most effective postwelding treatment for preventing segregated graphite. With respect to the normalizing of welds, the results of laboratory graphitization tests conducted on normalized welded samples by The Detroit Edison Company may be of interest.

A sample of high-aluminum carbon-molybdenum pipe mate-

[3] Research Department, The Detroit Edison Company, Detroit, Mich.

Fig. 9 Segregated Graphite in the Heat-Affected Area of "As-Welded" Carbon-Molybdenum Sample After 10,300 Hr of Laboratory Test Heating

Fig. 10 Random Graphite in Former Heat-Affected Area of a Normalized Weld in Carbon-Molybdenum Sample That Was Laboratory-Heated for 10,300 Hr After Normalizing

rial was welded to low-aluminum material and then normalized by heating for 1 hr at 1725 F, followed by air-cooling. To promote graphitization, "as-welded" as well as "normalized" samples were heated at temperatures varied between 1000 and 1060 F for 3300 hr, and then at temperatures varied between 900 and 1150 F for an additional 7000 hr. After a total time of 10,300 hr of test heating, the as-welded samples showed a distinct segregation of nodular graphite in the heat-affected area whereas the normalized samples showed only random graphitization. Segregated graphite in the heat-affected area of as-welded high-aluminum material is shown in Fig. 9 of this discussion. Random graphitization at the former heat-affected area of the sample that was normalized after welding is shown in Fig. 10. Similar results were found in the low-aluminum material with the exception that considerably less graphite was present.

These results indicated that the normalizing treatment of 1 hr at 1725 F had been fully effective in removing the nucleation effect resulting from the heat of welding. Kerr and Eberle[4] have also reported on the apparent beneficial effect of normalizing after welding as a means of preventing the formation of segregated graphite. The results obtained from test prompt the suggestion therefore that the normalizing treatment used by the author, and which is not stated in the paper, may not have been fully effective in reconstituting and homogenizing the structure of the heat-affected area in his sample.

The author's suggestion that resistance to spheroidization may be a necessary and perhaps sufficient condition to prevent graphitization is probably based upon the hypothesis that spheroidization is the first stage of graphitization. Referring again to the work of Kerr and Eberle,[4] it is noted that they have reported on a number of cases in which steels that had spheroidized were found to be resistant to graphitization. Although spheroidization and graphitization may be parallel phenomena in steel at high temperature, it does not necessarily follow that spheroidization is the first stage of graphitization. In this association it might be pointed out that although the photomicrographs presented in Fig. 6 of the paper, are for the apparent purpose of showing the graphitization susceptibility of the aluminum-treated sample in contrast with the graphitization resistance of the sample having no aluminum addition, the two samples appear to be about equally spheroidized.

[4] Graphitization of Low-Carbon and Low Carbon-Molybdenum Steels" by H. J. Kerr and F. Eberle, A.S.M.E. pamphlet "Graphitization of Steel Piping," included in Trans. A.S.M.E., vol. 67, 1945.

Author's Closure

Mr. Rohrig's evidence that a sample of high-aluminum carbon-molybdenum pipe material, welded and normalized at 1725 F showed only random graphitization, is in interesting contrast to that on the carbon-molybdenum cast steels studied in the Project No. 29 investigation. Of seven different heats of carbon-molybdenum steel with 2 lb of aluminum addition per ton and normalized at 1650 F for one hour after welding by the various contributors of the specimens, all specimens showed a development of graphite after 5,000 hours of aging at 1025 F, of which the "contact" zone photomicrographs of Fig. 7 are typical. Since the presentation of the report in question, these aging tests have progressed far enough to show that upon 10,000 hours, the average degree of contact zone graphitization for the seven heats of steel normalized after welding is about the average degree which was obtained for a group of six similar heats of steel stress-relieved only after welding upon 4000 hours of aging. This degree of deterioration represents about one third of the carbon transformed to graphite. While normalizing has served to retard the development of graphite at the contact zone, it by no means promises to be a satisfactory measure of control for the carbon-molybdenum cast steels.

Mr. Rohrig questions the observation that spheroidization of carbide is a prelude to graphitization and supports his objection by pointing out that certain samples reported upon showed spheroidization but no graphitization. However, since the report was presented, examinations after 10,000 hours of aging at 1025 F have revealed further spheroidization and graphitization in a number of samples where it had not been found after 5000 hours, including those containing one half of one per cent of chromium, thus indicating that the development of graphite in sequence to spheroidization may be an eventual effect to be always considered relative to aging time, temperature stress, and the composition of the steel.

Comparative Graphitization of Some Low-Carbon Steels With and Without Molybdenum and Chromium

By G. V. Smith,[1] S. H. Brambir,[1] and W. G. Benz[1]

Study of several experimental heats of steel containing 0.5 per cent molybdenum and up to 1.20 per cent chromium showed that in steels containing 0.5 per cent or more of chromium, no graphitization occurred in 3000 hr at 1025 F; but this finding requires confirmation by study of additional heats, particularly of commercial manufacture. Additional studies relating to the effect of deoxidation practice, particularly in plain carbon steels, and of the effect of posttreatment of welds on graphitization are described.

Introduction and Summary

In continuation of investigations (1, 2, 3)[2] begun shortly after the now well-publicized pipe failure at the Springdale Generating Station of the West Penn Power Company (4), we have studied the occurrence of graphitization in the following samples:

1 Unwelded samples of 10 heats of 0.1–0.2 per cent carbon steel made and deoxidized differently.

2 A series of experimental heats of chromium-molybdenum steels to ascertain the effectiveness of chromium in preventing graphitization.

3 Molybdenum steel treated to simulate the structures occurring in the heat-affected region of a weld.

4 Molybdenum steel (0.5 per cent), which had been heated at 1300 F after welding, as a means of preventing localized graphitization in the heat-affected region.

The results are as follows:

1 In normalized samples from ten heats of 0.1–0.2 per cent carbon steel of different manufacturing and deoxidation practice exposed for 2000–3000 hr at 1025 F, graphite formed in only four steels. These four had been deoxidized with 2 lb or more of aluminum per ton of steel and were fine-grained and moderately abnormal in the carburizing test. Steels with 1 lb or less of aluminum per ton did not graphitize, even though in some cases slightly or moderately abnormal.

2 The results indicate that in a 0.5 per cent molybdenum steel at least 0.5 per cent chromium is necessary to prevent graphitization, but do not prove that this would always suffice. Experimental heats containing 0.5 per cent or more of chromium did not graphitize under the conditions of our experiments, no matter how they were deoxidized or treated before or after welding. This inference should be checked by additional tests on commercial heats, however, since, although a 0.25 per cent chromium—0.5 per cent molybdenum steel, deoxidized with more than 1 lb of aluminum, graphitized locally under certain conditions, a similar steel without chromium did not. Graphite did not form in the 0.25 per cent chromium—0.5 per cent molybdenum steel when the welded sample had been heated to 1300 F for 4 hr prior to exposure at 1025 F.

3 Simulation of the temperature-time cycles which occur during welding indicated that the maximum amount of graphite appears in that zone of the weld-heat-affected region which reached a temperature in the vicinity of 1400 F. This temperature is only an approximation and is interrelated with the time at temperature and cooling rate. Further confirmation was obtained of the efficacy of a 1300 F treatment, prior to simulated service, in preventing graphitization.

4 In a sample of pipe from the Schuylkill Generating Station normalized from 1650 F, graphitization was entirely suppressed by a treatment of 4 hr at 1300 F after welding and prior to simulated service. On the other hand, similar tests on samples which had not been normalized showed some graphite; exposure for 8 and 12 hr, instead of 4, at 1300 F still further reduced the amount of graphite to the point where it could hardly warrant concern.

I Graphitization of Ten Carbon Steels (0.1–0.2 Per Cent Carbon), Made and Deoxidized Differently, at 1025 F

Because of indications that manufacturing and deoxidation practices affect the tendency to formation of graphite in steels, we exposed samples of ten commercial carbon steels which happened to be available, in a furnace at 1025 F for a period of (*a*) 1000 hr, (*b*) 2000 hr, and (*c*) in the case of Nos. 1, 2, and 3, 3000 hr. The samples were examined metallographically for the presence of graphite, with results given in Table 1, along with the composition and mode of deoxidation of each steel. Each steel had initially been normalized from 1650 F; the samples were 3 in. long and $^3/_4$ to 1 in. thick. We also carburized[3] a sample of each steel and determined the resulting grain size and structure "normality" because of the recent claim that degree of "abnormality" is an indicator of tendency to graphitize.

Graphite was detected in only four of these ten steels, namely, in Nos. 7, 8, 9, and 10, as illustrated in Fig. 1. These four comprise all which had been deoxidized with 2 lb or more of aluminum, had residual aluminum in excess of 0.025 per cent, and were fine-grained; on the other hand, the six which did not graphitize had been deoxidized with not more than 1 lb of aluminum, had less than 0.007 per cent residual aluminum, and were coarse-grained. The four which graphitized were moderately abnormal, but so were some of those (Nos. 4 and 5) which did not; thus these limited data indicate that although carbon steels which graphitize are abnormal, the converse is not necessarily true. Consequently, the normality cannot be regarded as a reliable indicator of tendency to graphitize. More-

[1] Research Laboratory, United States Steel Corporation, Kearny, N. J.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Joint A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperatures on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

[3] By holding 8 hr at 1700 F in a commercial carburizing compound (Houghton's Pearlite No. 50) and furnace-cooling at a rate which averaged 4 deg F per minute in the range 1700 to 1000 F.

TABLE 1 CHEMICAL ANALYSIS, DEOXIDATION PRACTICE, CARBURIZED GRAIN SIZE AND STRUCTURE NORMALITY, AND GRAPHITIZATION SUSCEPTIBILITY OF TEN HEATS OF CARBON STEEL

Grade	No.	Main deoxidizers added,[a] in lb per ton	Chemical analysis[b] C	Mn	P	Si	Al	After carburizing test A.S.T.M. grain size	Structure normality	Graphitization at 1025 F[c] 1000 hr	2000 hr	3000 hr
Open-hearth, 0.15 carbon	1	FeSi 1Al	0.13	0.45	0.011	0.19	0.007	2–4	Slightly abnormal	No	No	No
Open-hearth, 0.15 carbon	2	FeSi 1/2Al 4FeCTi	0.14	0.47	0.008	0.21	0.004	1–4	Slightly abnormal	No	No	No
Open-hearth, 0.15 carbon	3	FeSi 6FeCTi	0.14	0.46	0.009	0.19	0.005	3–4	Slightly abnormal	No	No	No
Capped open-hearth	4	1/4Al	0.07	0.45	0.006	0.01	...	1–4	Moderately abnormal	No	No	...
Capped bessemer	5		0.08	0.38	0.079	0.01	...	2–3	Moderately abnormal	No	No	...
Grade B, Si-killed	6	FeSi	0.27	0.52	0.016	0.22	0.006	2–4	Normal	No	No	...
Grade B, Si-Al-killed	7	FeSi 2Al	0.24	0.86	0.013	0.23	0.033	7–8	Moderately abnormal	Slight amount	Slight amount	...
Grade XB, bessemer	8	FeSi 4Al	0.14	0.54	0.074	0.19	0.025	7–8	Moderately abnormal	Yes	Yes	...
Grade XB, bessemer	9	FeSi 3Al	0.13	0.48	0.087	0.21	0.047	7–8	Moderately abnormal	Yes	Yes	...
Grade XB, bessemer, Al-killed	10	4Al	0.18	0.51	0.077	0.03	0.066	8	Moderately abnormal	Yes	Yes	...

[a] Other than FeMn which was added in all steels.
[b] Sulphur was less than 0.04 in all steels.
[c] No—none detected.

FIG. 1 TYPE AND AMOUNT OF GRAPHITE IN NORMALIZED CARBON STEELS
(*a*, No. 7; *b*, No. 8; *c*, No. 9; *d*, No. 10 of Table 1, after 1000 hr at 1025 F. Picral etch; ×500.)

over, according to these observations, the amount of residual aluminum, even though the accuracy of the analytical procedure is uncertain, appears to be a better indicator of tendency to graphitize than is the abnormality of the carburizing test.

The amount of graphite was greatest in steel 9, and progressively less in Nos. 8, 10, and 7, as determined by counting the number of nodules in the following manner: In an area at the center of the $^{3}/_{4}$-in. × 1-in. polished but unetched surface, the number of nodules in 50 fields 1 mm apart was determined after both 1000 and 2000 hr at 1025 F. To test whether a sufficient number of observations had been made to obtain a true picture, a second set of 50 observations was made, midway between the original ones. The results are given in Table 2.

TABLE 2 NUMBER OF GRAPHITE NODULES IN STEEL SAMPLES

Steel no.	Hours at 1025 F	Number of nodules in First 50 fields	Second 50 fields	Total 100 fields
7	1000	2	2	4
	2000	4	9	13
8	1000	77	88	165
	2000	123	114	237
	1000	191	202	393
	2000	272	257	529
10	1000	16	22	38
	2000	31	28	59

Taking into account both this number and the size of the nodules in Fig. 1, it is evident that the amount of graphite is greatest in steel 9 and progressively less in steels 8, 10, and 7. It was hoped that the counts would also provide an indication of whether the number of graphite nuclei is limited or increases with time. The data appear, on first thought, to show an increase; but when it is considered that nodules already present continue to grow, and that the chance of intersecting a nodule (by the plane of polish) increases with its size, it is realized that the evidence is insufficient to permit any decision on the matter.[4] It can be said, however, that if new nodules form, their rate of appearance between 1000 and 2000 hr at 1025 F is not great.

[4] If the nodules were uniform in size (which they probably are not), the number counted should increase in proportion to the increase in radius, and since this number did not even double (except for steel 7), it would be necessary to make precise measurements of nodule size and size distribution.

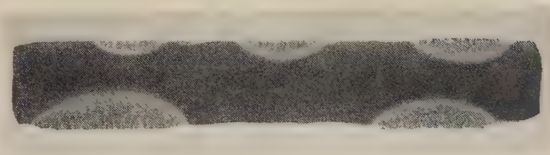

Fig. 2 Longitudinal Cross Section of Weld Sample Employed in Second Section of Paper

(Weld deposited for $^{1}/_{2}$, 1, 2, 4, or 8 sec. Nital-picral etch; ×1.)

The amount of graphite in the several steels could not be correlated with chemical composition nor with initial microstructure.

II Effectiveness of Chromium in Preventing Graphitization of 0.5 Per Cent Molybdenum Steel

Among the suggestions for preventing graphitization in 0.5 per cent molybdenum steel is that of adding chromium, which is one of a group of elements that, when present in sufficient proportion, combine with carbon in steel to form so-called alloy carbides. Molybdenum, also one of this group, is reputed (5) to possess stronger affinity for carbon and presumably forms a more stable carbide than does chromium; consequently, if carbide stability is a factor, there is no a priori reason for thinking that chromium addition will be more effective than equivalent increase in molybdenum.

Materials and Procedure. In an effort to ascertain whether chromium is effective in preventing graphitization and, if so, the minimum amount necessary, we investigated a number of heats with 0.5 per cent molybdenum and up to 1.20 per cent chromium, deoxidized with $^{1}/_{4}$ or $1^{1}/_{4}$ lb of aluminum per ton or with ferrosilicon-zirconium, as listed in Table 3. These were 20-lb induction-furnace heats, forged to $^{1}/_{2}$-in. × 1-in. bars, which were cut into 3-in. lengths. Two specimens of each were heated at 1450, 1650, or 2200 F for $^{1}/_{2}$ hr and air-cooled. After rough-grinding to remove scale, five weld deposits were made on each sample, as illustrated in Fig. 2, using molybdenum-steel weld rod ($^{3}/_{16}$-in. Murex C-Mo 50); the rod was held for $^{1}/_{2}$, 1, 2, 4, or 8 sec at different locations, the sample being cooled to room temperature before each deposit. In this manner, it was hoped that the effect of steepness of the temperature gradient through the heat-affected region during welding could be determined, since an earlier study (3) showed susceptibility to graphitization to be greater the steeper this gradient. One of each pair of specimens was then heated 4 hr at 1300 F to determine the effectiveness of this posttreatment in preventing localized graphitization (1, 3). All samples were then heated for 3000 hr at 1025 F in a muffle furnace, after which they were sectioned longitudinally through the weld deposits, and examined microscopically for graphite.

We also carburized a sample of each steel,

TABLE 3 CHEMICAL ANALYSIS, DEOXIDATION PRACTICE, AND CARBURIZED GRAIN SIZE AND STRUCTURE NORMALITY OF CHROMIUM-MOLYBDENUM SERIES OF STEELS

Ident.	C	Mn	Si	Cr	Mo	Zr	Al	Al_2O_3	Deoxidation, lb Al per ton	After carburizing test at 1700 F: A.S.T.M. grain size	Structure normality
A	0.14	0.45	0.17	..	0.52	..	0.007	0.008	$^{1}/_{4}$	4–6	Moderately abnormal
B	0.15	0.46	0.16	..	0.52	..	0.032	0.019	$1^{1}/_{4}$	8–9	Quite abnormal
C	0.16	0.52	0.18	0.27	0.54	..	0.010	0.015	$^{1}/_{4}$	2	Slightly abnormal
D	0.15	0.54	0.24	0.27	0.53	..	0.055	0.025	$1^{1}/_{4}$	8	Moderately abnormal
E	0.14	0.61	0.25	0.60	0.54	..	0.012	0.022	$^{1}/_{4}$	7–8	Slightly abnormal
F	0.13	0.61	0.23	0.60	0.53	..	0.073	0.030	$1^{1}/_{4}$	7–8	Slightly abnormal
G	0.14	0.58	0.22	1.17	0.52	..	0.013	0.014	$^{1}/_{4}$	7–8	Slightly abnormal
H	0.14	0.57	0.23	1.17	0.54	..	0.081	0.018	$1^{1}/_{4}$	7–8	Slightly abnormal
I	0.15	0.66	0.26	0.03	0.48	0.05	..	..	Fe-Si-Zr	6–7	Normal
J	0.16	0.55	0.20	0.12	0.54	0.05	..	..	Fe-Si-Zr	5–6	Slightly abnormal
K	0.17	0.57	0.21	0.58	0.54	0.04	..	..	Fe-Si-Zr	6–7	Slightly abnormal
L	0.15	0.41	0.16	1.19	0.53	0.03	..	..	Fe-Si-Zr	4–6	Slightly abnormal

Note: Sulphur and phosphorus were less than 0.035 or 0.01 per cent, respectively, in steels I, J, K, and L and were not determined in the remainder.

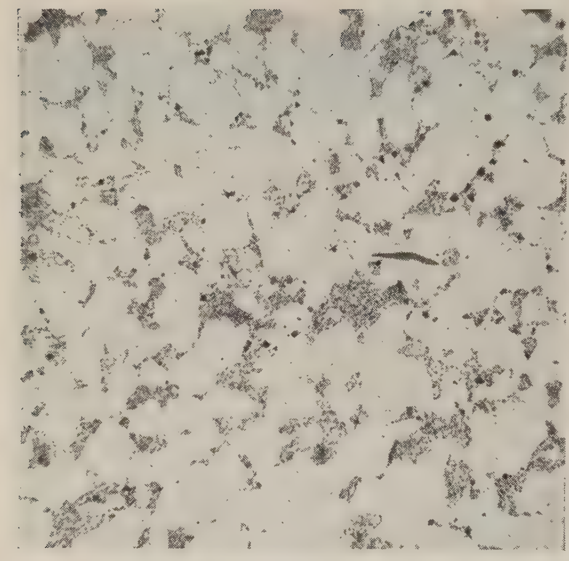

Fig. 3 Graphite in Heat-Affected Zone of Weld in Steel D, $^1/_2$ Mo-$^1/_4$ Cr, After 3000 Hr at 1025 F
(Picral etch: ×500.)

and determined the resulting grain size and normality, with results given in Table 3.

Results and Discussion. Only two samples definitely graphitized; these were both of steel D, 0.25 per cent chromium-0.5 molybdenum, deoxidized with $1^1/_4$ lb of aluminum per ton. In the samples of this steel normalized from 1450 or from 1650 F before welding, graphite formed in the heat-affected region of all welds, the amount being not appreciably different among the welds, contrary to our earlier finding (3). The graphite nodules were quite small, Fig. 3, and only a very few were found in unaffected base metal. No graphite was observed in the companion samples heated 4 hr at 1300 F after welding and prior to exposure at 1025 F, in further confirmation of the efficacy of this proposed preventive treatment (1, 3), nor was any found in samples previously normalized from 2200 F, confirming a tendency, reported earlier (3), toward increased resistance to graphitization of structures resulting from normalizing at high temperature.

None of the others showed any graphite in any condition of prior or posttreatment, with the possible exception of B, normalized at 1650 F before welding and not heated at 1300 F after welding. In this there were several widely scattered nodules which appeared to be graphite, but because they were so small and so few, we cannot be certain. That this steel did not graphitize more extensively, in one or more original conditions, was unexpected, since it was thought that the molybdenum steel would be more prone than the chromium-molybdenum to form graphite.

This possible inconsistency in behavior emphasizes again the danger of basing claims on observations on a small number of heats; the results do indicate, however, that more than 0.25 per cent chromium must be present in order to inhibit localized graphitization.

All these steels were to some extent "abnormal" in the carburizing test except I, the plain 0.5 per cent molybdenum steel deoxidized with ferro-silicon-zirconium. Because of lack of graphitized steels in this series it is not possible to correlate "abnormality" with tendency to graphitization. However, the normality ratings are of some interest in themselves, showing, for example, that it is difficult to produce a strictly normal steel; that 0.5 per cent molybdenum steel is "normal" when deoxidized with ferro-silicon-zirconium confirms an earlier finding (7).

III Simulation of Structures of Heat-Affected Region Which Graphitize

It has been shown that the zone of localized graphitization in welded molybdenum steels is that which during welding reached a temperature slightly above the lower critical, A_1 (1340 F). To learn more about this phenomenon, samples of a molybdenum steel, susceptible to graphitization, were heated for short times in the range 1350–1450 F, and gradiently cooled, thin samples being used to insure rapid temperature change.

From a sample of Springdale Station[5] pipe, air-cooled as a $^3/_4$-in. section after $^1/_2$ hr at 1650 F, a number of samples approximately $^1/_{16} \times {}^1/_4 \times 1$ in. were prepared. Using a coke-covered, deoxidized lead bath held at constant temperature within the range 1350–1450 F, and tongs heated to the bath temperature, one end of a sample was grasped in the tongs, and the whole immersed in the lead bath for $^1/_2$ min; its free end was then immersed to a depth of $^1/_4$ in., in water at room temperature and held until reasonably cool. A duplicate set of samples was treated at each of the temperatures 1350, 1375, 1400, 1425, and 1450 F. Each set was then sealed, in vacuo, in a silica bomb to prevent subsequent oxidation and decarburization, and one of these was heated at 1300 F for $4^1/_2$ hr; both were held for 1000 hr at 1025 F, and each sample was then examined metallographically upon a longitudinal section for graphite.

The samples were held for only $^1/_2$ min in the lead bath, because we thought this might simulate welding conditions. In such a short time at only slightly above the lower critical temperature, there is time for only partial transformation of ferrite and carbide to austenite, and the small regions, necessarily rich in carbon, of this latter constituent presumably set the stage for localized graphitization (1). The specimens were gradiently quenched to provide a progression of cooling velocities, one of which should correspond to the fairly fast rate of the heat-affected region of a weld.

After 1000 hr at 1025 F, graphite was observed in each of the five samples which had not received the 1300 F treatment, but not in any of the companion five which had been so treated. The photomicrographs in Fig. 4, taken approximately $^1/_4$ in. from the quenched end, show a comparison of the five samples. The 1350 F sample showed no graphite in the quenched zone, Fig. 4(*a*), but there were a few small nodules in the tong-held end. At 1375 F there was some graphite near the quenched end, Fig. 4(*b*), but practically none in the opposite end. The quenched zone of the 1400 F specimen, Fig. 4(*c*), showed the maximum amount of any sample, and the tong-held end showed an intermediate amount. At 1425 F the amount in the quenched zone had decreased sharply, Fig. 4(*d*), and there was an appreciable amount in the tong-held end. At 1450 F there were very few nodules at either end, Fig. 4(*e*).

Thus these experiments indicate that graphite forms most readily in that portion of the heat-affected region which, during welding, was at a temperature in the neighborhood of 1400 F. This is only an approximation since our holding time of $^1/_2$ min may not exactly simulate welding conditions; the temperature in any event is not critical. The experiments also indicate that cooling rate is significant. That no graphite occurred in any sample postheated at 1300 F provides further confirmation (3) of the utility of this treatment.[6]

IV Posttreatment at 1300 F as a Preventive of Localized Graphitization

Sometime ago we suggested (1) that localized graphitization

[5] Of West Penn Power Company, where the problem first arose (4).
[6] See also the second and fourth sections of this paper.

FIG. 4 REPRESENTATIVE REGION NEAR QUENCHED END OF THIN SAMPLES OF 1/2 PER CENT MOLYBDENUM STEEL
(Heated to *a*, 1350 F; *b*, 1375 F; *c*, 1400 F; *d*, 1425 F; and *e*, 1450 F, for 1/2 min and gradiently quenched, then heated at 1025 F for 1000 hr. Picral etch; ×1000.)

might be avoided by heating to 1300 F after welding and prior to service, and recently obtained experimental corroboration (3) of the efficacy of this treatment on samples of Springdale Station pipe.[5]

Shortly after our results were published, H. Weisberg of Public Service Electric and Gas Company, reported privately that he had been unable to prevent graphite formation by using this suggested treatment although its amount had been reduced; he then kindly supplied us with a sample of his steel, which was used in the following experiments.

The samples were reported to be from a 0.5 per cent molybdenum-steel pipe manufactured by National Tube Company, Heat No. 6532, in 1937 to A.S.T.M. specification A158-36, Grade P-1, was deoxidized with 1.8 lb of aluminum per ton, and was furnished for the Philadelphia Electric Company's Schuylkill Station as annealed between 1350 and 1400 F. It had not been

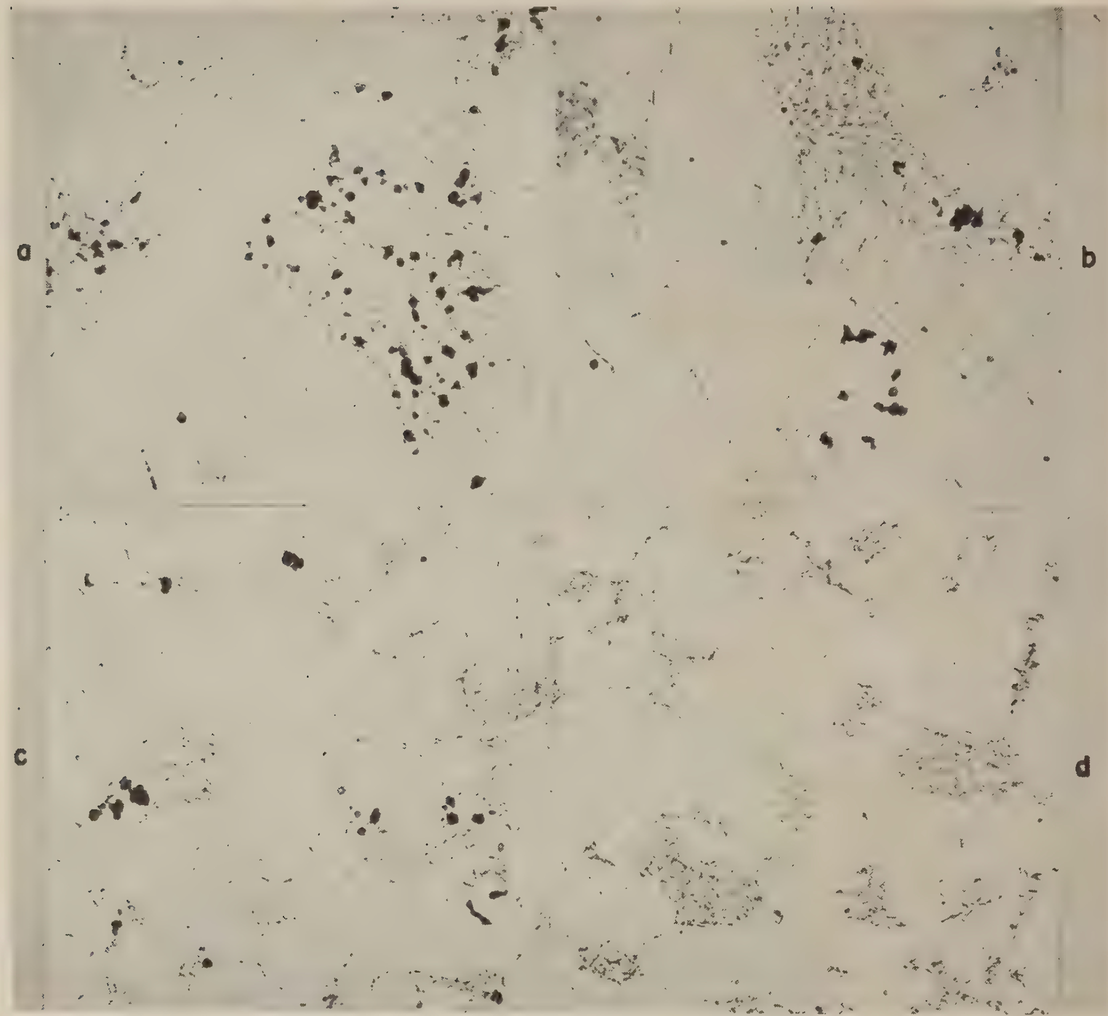

FIG. 5 GRAPHITIZATION IN HEAT-AFFECTED REGION OF 1/2 PER CENT MOLYBDENUM STEEL FROM SCHUYLKILL STATION PIPE AFTER 1000 HR AT 1025 F

(Picral etch: ×500.)

(*a*, As-received, not posttreated; *b*, as received, posttreated 4 hr at 1300 F; *c*, normalized 1650 F, not posttreated; *d*, normalized 1650 F, posttreated 4 hr at 1300 F.)

in service but had, in the course of Mr. Weisberg's experiments, been subjected twice to a 1200 F "stress-relief" treatment.

Since in our earlier experiments (3) with the Springdale Station pipe, we had normalized the samples prior to test, and since Mr. Weisberg had conducted his tests in the as-received condition, we decided to test in both conditions. Accordingly, two specimens approximately $1 \times 2^1/_4 \times 3^3/_8$ in. were prepared, one as received, the other normalized $^1/_2$ hr at 1650 F. Upon each of these a "narrow" and a "wide" weld bead, illustrated in Fig. 6, were laid down according to the following conditions:

Preheat	None	
Electrode	Murex C-Mo 50	
Weld type	Bead weld deposited in flat position	
	Narrow	Wide
Current, amp	200	285
Arc voltage, volts	28	30
Speed, ipm	9	3
Heat input, Btu per in	37	171

The purpose in having both a narrow and a wide weld bead was to obtain further information on the effect of steepness of temperature gradient in the heat-affected region. After welding, each sample was cut in half and one half heated 4 hr at 1300 F and air-cooled; all four were then heated 1000 hr at 1025 F, sectioned, and examined microscopically.

Figs. 5(*a*), (*b*), (*c*), and (*d*), representing the structure of the low-temperature zone of the heat-affected region of the narrow weld bead of each sample, illustrate the results. In the as-received sample, not heated at 1300 F after welding, much graphite had formed, Fig. 5(*a*), but the companion section heated at 1300 F after welding showed considerably less, though there was some, Fig. 5(*b*), corroborating Mr. Weisberg's finding. The sample normalized before welding also showed considerably less graphite, Fig. 5(*c*), and the companion section heated at 1300 F after welding showed none at all, confirming our earlier observations on the normalized Springdale Station pipe. In those which had graphitized the amount was greater in the heat-affected region of the narrow than of the wide weld bead. Graphite was not observed in the unaffected base metal of any sample.

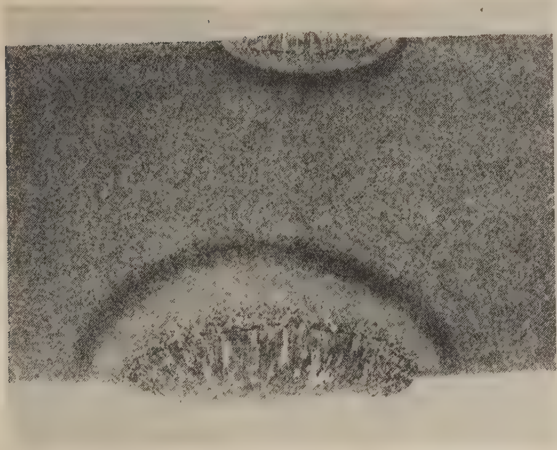

Fig. 6 Transverse Cross Section of Weld Sample Employed in Section IV
(Nital-picral etch: ×2.)

We do not know why graphite formed in the as-received but not in the normalized sample heated at 1300 F after welding. As apparent in Fig. 5, the as-received material was coarse-grained, indicative of a comparatively high temperature during its last supracritical treatment; this, however, probably would contribute to greater rather than to less resistance.[7] One suspicious factor is the 1350–1400 F annealed condition in which presumably the steel was furnished, this being the range (see preceding section) in which the stage is set for graphitization in the heat-affected region of a weld. With this beginning, the two stress-relief treatments at 1200 F which the material had undergone, may have so stabilized the graphite nuclei that they were not dissolved by subsequent heating for 4 hr at 1300 F,[8] i.e., treatment at 1200 F may have initiated the formation of graphite, but was too short to result in visible nodules. Some credence is given to this hypothesis by the results of the following experiment:

Upon another as-received sample a weld bead was laid under conditions identical with those just described for the narrow bead (since this produced more graphite). The sample was then cut in half, and one half heated for 8 hr, the other for 12 hr at 1300 F; both were then exposed for 3000 hr at 1025 F and examined microscopically. Both samples contained graphite in the heat-affected region, but in such small amount that it was difficult to find; Fig. 7 illustrates the largest amount. The amount appeared to be about the same in both samples,[9] and none was observed in the unaffected base metal of either. Thus longer heating at 1300 F resulted in considerably less graphite, presumably because the graphite nuclei had been dissolved or had reverted to carbide.

[7] See reference (3) and also the second section of this paper.

[8] In an earlier paper (2), we showed that graphite in a similar steel could be caused to revert to carbide by heating at 1300 F.

[9] On afterthought, it is realized that the choice of 12 hours was poor; 16 or 24 hours would have been better.

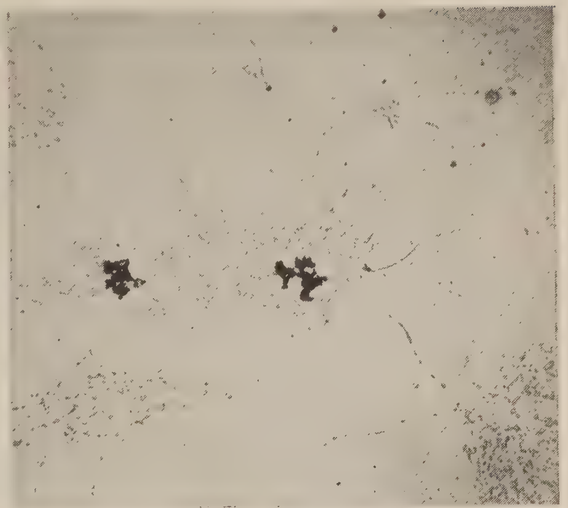

Fig. 7 Most Severe Graphitization in Heat-Affected Zone of Schuylkill Station Pipe Welded in "As-Received" Condition and Then Posttreated 8 Hr at 1300 F and Held at 1025 F for 3000 Hr
(Picral etch: ×500.)

Acknowledgments

Dr. R. F. Miller of Carnegie-Illinois Steel Corporation and Mr. E. C. Wright of National Tube Company furnished the steels studied except for item 4, which was supplied by Mr. H. Weisberg of the Public Service Electric and Gas Company, and for a sample of the Springdale pipe supplied by Mr. R. W. Emerson of Pittsburgh Piping and Equipment Company.

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Further Observation of Graphitization in Aluminum-Killed Carbon-Molybdenum-Steel Steam Piping

By R. W. EMERSON[1] and MATHEW MORROW[2]

Among graphitization of steel pipe studies being carried on, the present one deals with what the authors have termed "slip-plane" graphitization, which describes a condition of graphite segregation along slip planes where local "yielding" or localized plastic deformation had previously occurred. Such graphitization has been found to vary widely from the outside to inside of a pipe and from quadrant to quadrant around the pipe periphery. It has been determined that "nodular" graphite segregation whether of the "isotherm" or slip-plane type is as effective as "chain" graphite in reducing the ductility of low-carbon or low-carbon-molybdenum steel.

Introduction

The graphitization of steel containing less than 1 per cent carbon and particularly less than 0.2 per cent carbon, at subcritical temperatures although reported in the literature (1)[3] was, prior to 1943, of little more than academic interest.

The complete circumferential failure of a carbon-molybdenum-steel steam pipe in January, 1943, which was attributed to subcritical graphitization, and the subsequent discovery of the same phenomenon in pipe lines in many other high-temperature steam power plants during the past 3 years has, however, resulted in the establishment of numerous intensive research programs which have been sponsored by both private industry and joint investigating committees. The results of these various investigations have been viewed by both the consumer and the producer with increasingly keen interest.

A report of the failure mentioned was given, together with three other papers dealing with graphitization, before this Society in December, 1943 (2, 3, 4, 5). The many investigations which have been carried out during the past 3 years have shown that graphite tends to segregate along the extremity of the heat-affected zones of welded joints, provided the steel is one in which the carbide is potentially unstable.

In February, 1945, however, routine sampling of several joints at the Springdale Power Station, at which station the original failure occurred, revealed a condition of graphite segregation which was not only of a serious nature, but one which was totally different from anything previously encountered. Whereas the terms "isotherm" and "eyebrows" were used to describe graphite segregation encountered adjacent to welded joints, due both to the location and macroscopic appearance of the graphite, the authors have chosen in this case the term "slip-plane" graphitization, to describe a condition of graphite segregation which appears to have unquestionably occurred along slip planes where local "yielding" or localized plastic deformation had previously occurred.

Although the discovery of slip-plane graphitization adds further to the general field of knowledge about graphitization, it also points to the over-all complexity of this general problem with which we are dealing.

Location of Slip-Plane Graphitization

In sharp contrast to "isothermal" graphitization adjacent to welded joints, slip-plane graphitization varies markedly from ID to OD and from quadrant to quadrant on the pipe periphery. For this reason the location of the graphitized area with respect to the boiler lead in which the difficulty was found, as well as its location with respect to the original failure, is given in Fig. 1. Twelve inches of pipe, including the upset pipe end of No. 4 boiler lead, was removed at the time of the failure in January, 1943. This was replaced by a spool of pipe from one of the three heats of pipe originally used in this installation in 1937. This pipe, however, had never been in service and was used without

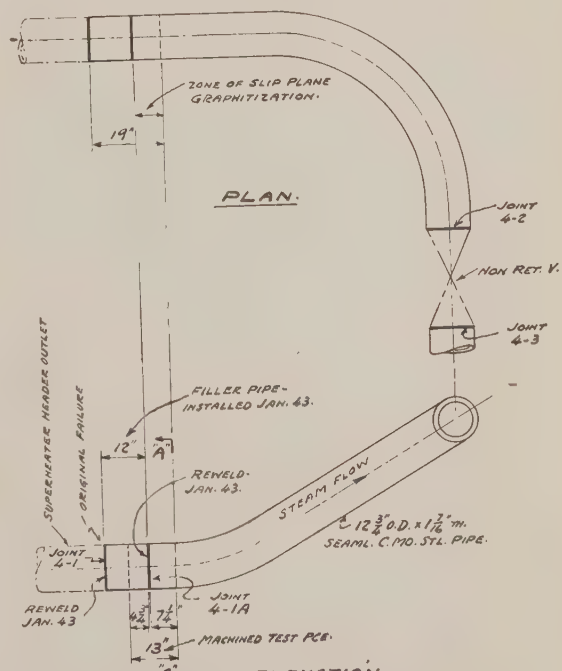

Fig. 1

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[1] Metallurgist, Pittsburgh Piping and Equipment Company, Pittsburgh, Pa.

[2] Assistant Metallurgist, Pittsburgh Piping and Equipment Company.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Joint A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperature on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

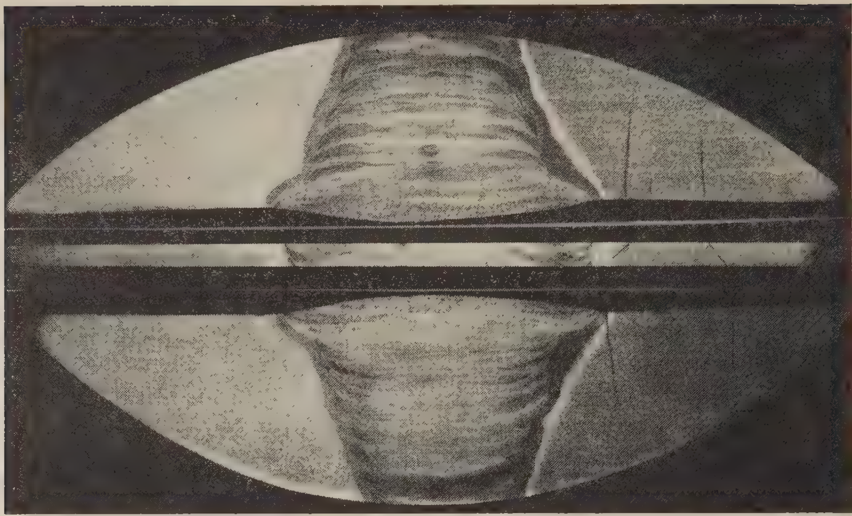


Fig. 2a Panoramic View of Graphitization as Found in "Weld Prober" Specimen From Joint No. 4-1A, 12-In. Sch. 160 Pipe-to-Pipe Weld; ×1
(Graphitization runs from OD to ID and on 45-deg angle with respect to longitudinal direction of pipe. Plastic deformation along planes of maximum shear stress is believed to be responsible for nucleation of graphite.)

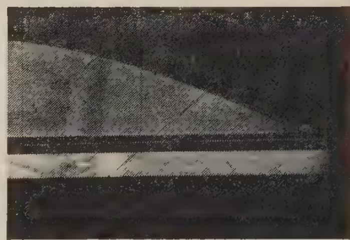


Fig. 2b Circumferential Cracks at OD Which Run at a 45-Deg Angle to Direction of Pipe Rolling

upsetting. Joints 4-1 and 4-1A as shown in Fig. 1, were welded, using a 400 F preheat and a 1200 F stress-relieving heat-treatment.

In January, 1944 (1 year later), $1^1/_2$-in-diam trepanned plugs were removed from both the inlet and outlet sides of joints 4-1A and the inlet (pipe) side of joint 4-2. Microscopic examination revealed incipient graphitization of the 12-in. spool as well as a mild reinfection (graphitization) of both the outlet side of joint 4-1A and the inlet side of joint 4-2. Segregated graphite of a serious nature, however, was not present in the plugs examined.

In February, 1945 (2 years later), "weld-prober" specimens were removed from the same two joints at approximately 45 deg from the location of the trepanned plugs.

During the preparation of the specimen from joint 4-1A it was observed that two cracks were present in the old pipe which had all the characteristics of previously encountered graphite segregation but which were in no way associated with the extremity of the heat-affected zone. Further examination revealed that these cracks, although extending from the OD toward the ID to a depth of $^5/_8$ in., did not run transverse to the direction of rolling on the pipe circumference but rather at an angle of 45 deg to the direction of rolling. The pattern of the cracks, as shown in Fig. 2, is typical of that which is produced in material which fails by shear.

Inasmuch as this condition existed across the entire width of the weld-prober specimen and not knowing to what extent it continued into the pipe or whether or not additional planes of graphite were present in the pipe remaining in service, it was deemed advisable to replace the entire bend between the superheater header and nonreturn valve as shown in Fig. 1. This pipe bend was accordingly removed on March 2, 1945, and replaced by a 1 per cent chromium 0.5 per cent molybdenum steel having an analysis comparable to A.S.T.M. Specification A 213 Grade T-12.

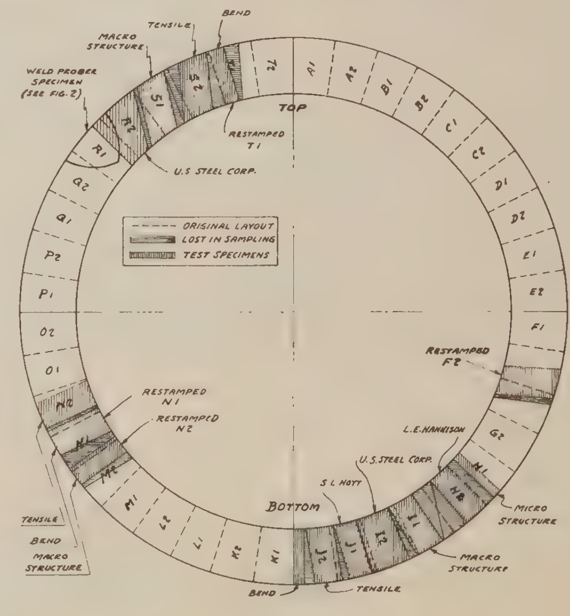


Fig. 3

Results of Macroscopic Investigation

In an effort to determine the extent of the grossly segregated slip-plane graphitization which was found in the weld-prober specimen from joint 4-1A, a 13-in. section from the bend was removed, comprising the joint mentioned, 4.75 in. of the 12-in-long spool which was installed at the time of failure, and 7.25 in. of the old pipe which had been in service since 1937, and which extended to a distance of 19 in. from the original failure.

Having previously found that heavily segregated graphite could be made visible to the unaided eye by either rough-grinding or machining, the surface of the 13-in. length of pipe was "turned" in a lathe to obtain a clean machined surface on both the OD and ID (2, 12).

After machining, the pipe section was stenciled in accordance with Fig. 3, and subsequently quarter-sectioned longitudinally. Before removing any test pieces, however, both the inside and outside pipe surface of each quarter-section was photographed. Figs. 4 to 7, inclusive, show the severity of slip-plane graphitization as revealed by surface-machining only. It is to be emphasized that no special machining, grinding, polishing, or etching techniques are required to locate heavily segregated graphite such as shown in Figs. 4 to 7, inclusive.

Several distinct lines of heavily segregated graphite, running at both a plus and minus 45-deg angle, can be seen in the upper right-hand corner in Fig. 4. (Section F-1 to J-2, inclusive, see Fig. 3.) A multitudinous number of fine lines of segregated graphite are also present in the upper center of Fig. 4. A good example of "isotherm" graphitization (graphite segregation at the extremity of a weld-heat-affected zone) is shown by the semicircular pattern in the upper left of this figure. This condition was produced by the fillet-welding of a 1-in-sq bar to the pipe, apparently for mechanical reasons during the original erection of this piping. This in effect is the result of a "weld-bead" test similar to that devised and used by Battelle Memorial Institute, and others in 1943, as a simple test for determining the susceptibility of steel to graphitization. The bead test shown in this illustration, however, was given an aging treatment of 935 to 950 F for approximately 65,000 hr (actual operating conditions), as contrasted with the artificial accelerated aging tests of 2000 to 8000 hr or longer at 1000 to 1050 F. Also present in Fig. 4 are several lines of graphite adjacent and running into weld joint 4-1A.

Lines of heavily segregated graphite running at both plus and minus 45-deg angles as well as a multitudinous number of the finer lines of graphite were found in section K1 to 02 inclusive (see Fig. 3). This is shown in Fig. 5. In addition to the foregoing, many other small speckled patches of localized graphite segregation of unknown origin are present in this figure. It is to be pointed out that lines of heavily segregated graphite are not present in the location of the trepanned plug (upper left, Fig. 5) which was removed in January, 1944. This plug was removed from the side of the pipe at the location 02-P1 shown in Fig. 3. Metallographic evidence, however, indicated that not only was heavily segregated slip-plane graphitization present at the time this plug was removed in 1944, but that this condition actually existed at the time the 12-in. spool was welded into the line in January, 1943. When sampling pipe for evidences of slip-plane graphitization therefore it cannot be overemphasized that the mere fact that such a condition is not detected in a test specimen does not mean that such a condition is nonexistent in the general vicinity of such a test specimen.

The location of the weld-prober specimen, the macrographs of which are shown in Fig. 2, is shown in Fig. 6 (section P1 to T2, inclusive), and it may be seen that it was coincidental that the location was so chosen that the heavily segregated slip-plane graphitization was intersected by the specimen removed. Had the specimen been removed anywhere between section A1 and E2 this condition would have gone undetected, and this pipe might possibly still be in service. The longest length of slip-plane graphitization, shown in Fig. 6, was found to be 4.5 in.

The inside pipe surface was machined to determine if the condition described could be located on the inside pipe surface or if the graphite segregation found on the outside of the pipe had actually penetrated the entire pipe wall. The condition found on the inside pipe surface, although confined to sections F1 to M2, is believed to be even a more striking example of a graphitization pattern, the orientation of which is unquestionably the direct result of stress.

Having observed the macroscopic appearance of both the inside and outside pipe surface, Figs. 8, 9, and 10 illustrate the graphite pattern found through the pipe wall at sections I1 and S1. A roughly ground and unetched view of section I1 is shown in Fig. 8 at normal magnification.

The same section as shown in Fig. 8 is shown in Fig. 9, after polishing and deep-etching in hot 50 per cent hydrochloric acid. This illustration which takes in a larger area than Fig. 8, is shown at two magnifications.

Since many of the lines of graphite segregation appear to emanate from the weld, it first appears that these lines formed as a result of and after joint 4-1A was completed (January, 1943). On the contrary, however, close examination of this figure reveals that these lines have been partially eradicated along the heat-affected zone of the weld. Microscopic observations further substantiate that re-solution of the graphite took place in the heat-affected zone as a result of the localized high temperature during the rewelding of this joint. This fact was subsequently substantiated by G. V. Smith and S. H. Brambir, U. S. Steel Corporation Research Laboratory (14). Furthermore, had the lines of graphite shown in Fig. 9 been initiated at the time of or subsequent to the welding of joint 4-1A, the condition of stress which initiated the graphite pattern in the old pipe would have been transmitted through the weld into the 12-in. spool and it would therefore be expected that similar lines of graphite segregation would be present in the 12-in. spool which was installed at the time joint 4-1A was made, and which was of similar chemistry and deoxidation as the original pipe.

Still another reason for believing that this condition was initiated prior to and present in 1943 is that the average graphite-nodule size of the grossly segregated slip-plane graphitization closely approaches the size of random graphite located in the many specimens of the originally installed material previously examined microscopically. Although growth of isotherm graphite nodules have been followed in several welded joints exhibiting incipient graphitization in 1943, there have been no cases where the rate of nodule growth over a 2-year period would approach that found in this case (assuming this condition to have developed after the original failure).

The depth and direction of the graphite segregation through the pipe wall at section S1 is shown in Fig. 10.

Physical Properties

Several bend and tensile tests were made on pipe sections, the locations of which are shown in Fig. 3. The results obtained are shown in Figs. 11 and 12 and in Table 1. These tests were made principally to confirm the authors' belief that gross segregation of "nodular" graphite whether of the isotherm or slip-plane type is as effective as "chain" graphite in reducing the ductility of low-carbon or low-carbon-molybdenum steel.

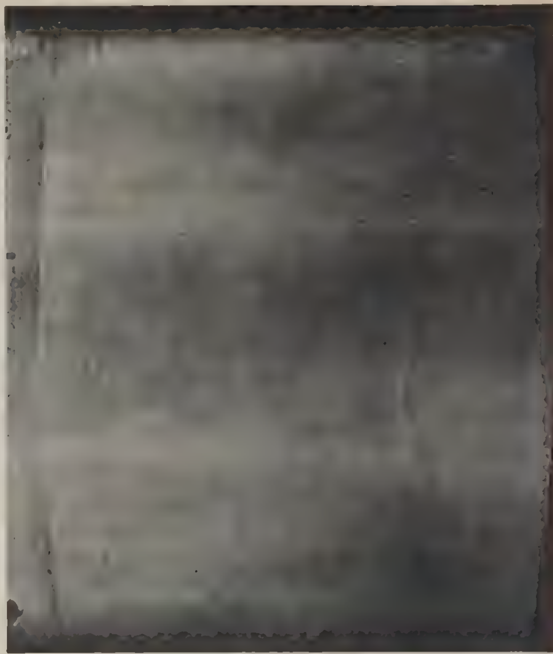
The bend test shown in Fig. 11 was removed from section T1 (see Fig. 3). From the location of the fissures it may be noted that this specimen was cut from the upper portion of the quadrant shown in Fig. 6. Although the graphite segregation in this case has no direct relation to the heat-affected zone of a weld, it is to

F1

J2

Fig. 4 Graphite Pattern on Machined Outside Pipe Surface of Section F1 to J2; ×0.8

T2



P1

Fig. 6 Graphite Pattern on Machined Outside Pipe Surface of Section P1 to T2; ×0.8

O2

K1

Fig. 5 Graphite Pattern on Machined Outside Pipe Surface of Section K1 to O2; ×0.8

F1

N1

Fig. 7 Graphite Pattern on Machined Inside Surface of Section F1 to N1; ×0.8

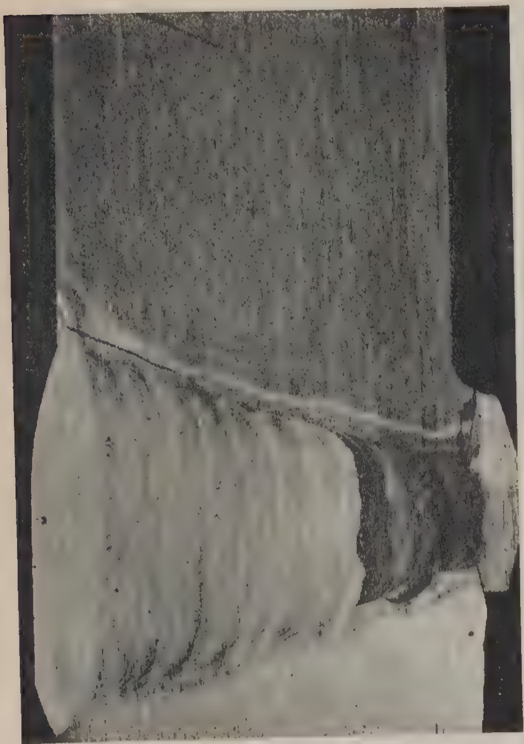


Fig. 10 Macrosection S1, Showing Slip-Plane Graphitization; ×1½
(Etchant, hot 50 per cent hydrochloric acid.)

Fig. 8 Unetched Macrograph of Section I1 Showing Lines of Segregated Nodular Graphite Running From ID Toward OD of Pipe; ×1

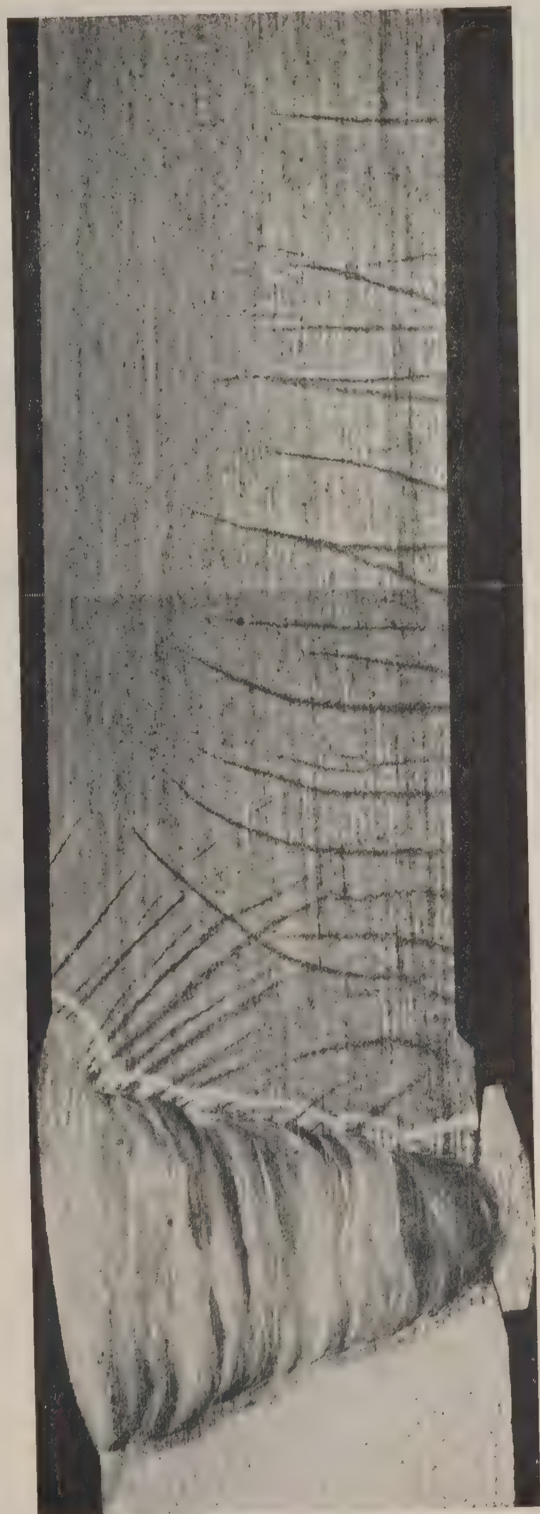


Fig. 9 Twelve-Inch Spool, Rewelded Joint No. 4-1A and Original Pipe 12 In. to 16 In. From Original Failure; ×1.8
(Location I1, see Fig. 3. Note large graphite stringers emanating from ID and intersecting those emanating from OD adjacent to weld. Etchant, hot 50 per cent hydrochloric acid.)

Fig. 11 Bend Test on Section T1, Showing Fissuring Along Graphitized Slip Planes; ×1

TABLE 1
Tensile Tests Removed From Graphitized Pipe

Section	Tensile strength, psi
J2	59130
N2	58860
S2	46050

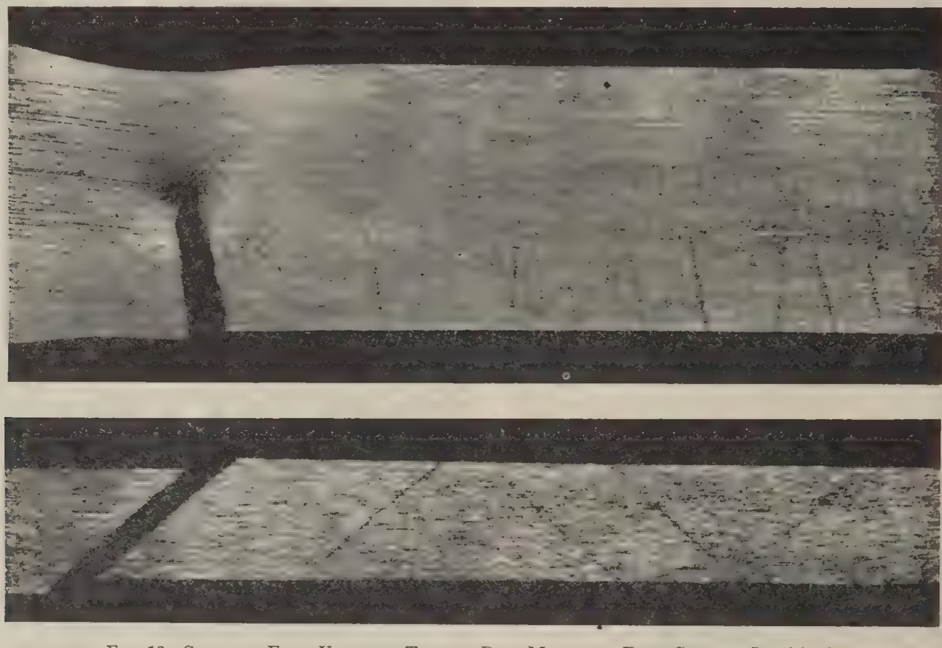


Fig. 12 Side and Edge Views of Tensile Bar, Machined From Section J2; $\times 1^1/_2$
(Note that failure occurred along one of the planes of graphite segregation.)

be carefully noted that its ultimate effect on the ductility of the pipe is similar to that which has been found to occur at the "contact" zone of a welded joint.

Both the side and edge view of a partially pulled tensile bar removed from section J2 is shown in Fig. 12. Planes of graphite may be observed to be "running" at both plus and minus 45 deg on the edge of the machined specimen. The edge view shown was adjacent to the inside pipe surface before removal of the specimen from the pipe. Numerous planes of graphite may be seen to extend from the ID toward the OD in the side view of this specimen. Failure occurred abruptly along one of the graphite planes at a stress of 59,130 psi, followed immediately by a necking-down of the ductile portion of the specimen.

Microscopic Examination

Although macroscopically, slip-plane graphitization as shown in Figs. 4 to 12, inclusive, appears as a continuous crack, microscopically it appears as discontinuous large gray nodules. The relative size and spacing of these nodules determine the resultant ductility of the material.

One of the planes of segregated graphite which was intersected by weld 4-1A in January, 1943, is shown at a magnification of 100 in Fig. 13. It may be seen that the nodules which were outside of the heat-affected zone of the weld remained unchanged, whereas those just within the heat-affected zone were partially put into solution. Those nodules which were in the immediate vicinity of the line of fusion may be seen to be completely dis-

Fig. 13 Intersection of Slip-plane Graphitization With Line of Fusion of Welded Joint 4-1A; ×100

solved by the heat from welding. The sudden solution of the graphite in the austenite, followed by the rapid rejection of the carbon (graphite) as a carbide phase, resulted in the formation of a localized band of steel of hypereutectoid composition. The carbide present in the darkened area in Fig. 13, which is probably in the range of 1 to 2 per cent carbon, can be seen in Fig. 14 to have partially regraphitized in the 2-year period following its inception.

Fig. 14 Partial View of Fig. 13, at Higher Magnification, Showing Structural Detail of Area in Which Graphite Was Redissolved and Reprecipitated as a Carbide Phase During Welding; ×600

The fact that the graphite was put into solution by the heat from the rewelding of joint 4-1A in 1943, is believed to be adequate proof that the graphite was present some time prior to the welding of this joint.

Further proof of this fact is offered in Figs. 15 to 17, inclusive. Slip-plane graphitization at a remote distance from the weld is shown at a magnification of 100 in Fig. 15. The size and detail of these segregated graphite nodules are shown in Fig. 16 at a magnification of 600.

In following the rate of growth of graphite adjacent to welded joints under both actual operating conditions and in accelerated aging tests made in laboratory furnaces, the authors have never encountered a rate of growth of graphite over a 2-year period in low-carbon-molybdenum steel which would even approximate that shown in Figs. 15 and 16.

This is believed to be substantiated by the published work of Kerr and Eberle (3, 6); Smith, Miller, and Brambir (4, 11); Weisberg (5, 12); Hoyt and Williams (7); Van Duzer, McCutchan,

Fig 15 Gross Segregation of Nodular Slip-Plane Graphitization; ×100

Fig. 16 Size and Detail of Graphite Nodules Shown in Fig. 15; ×600

and Rohrig (8, 9); as well as certain unpublished work on this subject by other investigators. In contrast to Figs. 15 and 16, however, the rate of growth of graphite over a 2-year period in the contact zone adjacent to joint 4-2 is shown at a magnification of 600 in Fig. 17. This joint was also rewelded in January, 1943, and it is known that graphite did not exist adjacent to this joint at the time it was rewelded.

Since such a vast difference in nodule size exists between Figs. 16 and 17, it is concluded by the authors that the slip-plane graphitization was initiated prior to 1943, and that the many graphite planes which appear to emanate from the weld are coincidental.

Discussion and Probable Cause of Slip-Plane Graphitization

Owing to the over-all complexity of the graphitization problem as it is known to exist, an attempt has been made to correlate the existing facts with other published information. In so doing

Fig. 17 Growth of Graphite in Regraphitized Joint No. 4-2 After 2 Years of Service; ×600

certain postulations could not be avoided in this discussion.

The iron - iron carbide system, although extremely useful and of real importance to the ferrous metallurgist, has been recognized for many years as a metastable system; the iron-carbon system being the truly stable one. Hoyt and Williams recently stated, "In spite of the apparent stability of iron carbide in the common carbon steels, in which it is the active strengthening and hardening constituent, it has a definite, although usually slight tendency, to change to the more stable form of graphite" (7).

A logical theory regarding the precipitation of carbon (graphite) in steam piping subjected to fluctuating temperatures was presented before this Society in 1944 by F. B. Foley (13). His theory was based upon the slope of the solubility curve of the alpha phase of the iron - iron carbide diagram together with the composition of the alpha phase (ferrite) with respect to aluminum or silicon or both.

It has been shown by Fink and Smith that in precipitation-hardening alloys of the duralumin type, precipitation of the $CuAl_2$ upon aging will always occur preferentially along slip planes resulting from any plastic deformation, before general precipitation occurs (15).

In this respect, it is to be pointed out that the solubility curve for the aluminum-rich solid solution in the aluminum-copper system is qualitatively the same as the solubility curve of the alpha phase in the iron - iron carbide system.

Luders' lines, or "stretcher strains" have been stated by Winlock and Leiter "to occur as a result of the uneven and nonuniform flow of metal at the yield point, caused by the sudden transition of different grains or groups of grain from the elastic to the plastic state." They further state, "Luders' lines occur in the range of deformation corresponding to an elongation of 1.5 to 10 per cent" (16).

On the basis of the foregoing it is believed that slip-plane graphitization is initiated by localized yielding or plastic deformation along slip planes with the formation of Luders' lines or stretcher strains. In the case under discussion major slippage occurred along planes of maximum shear stress. In Fig. 1 it is to be noted that the slip-plane graphitization is located along the pipe length just outside of the 30-deg bend. It is postulated that although this bend was made hot, a sufficient load was applied in making the bend to produce a critical strain (1.5 to 10 per cent elongation) in the cold pipe just outside of the hot-bent region. This bend was subsequently furnace-stress-relieved at 1150 to 1200 F before installation. This temperature was apparently too low to remove the effects of previous strain, in fact it is suggested that the stress-relieving temperature may have initiated actual submicroscopic graphitization along the planes which received localized plastic deformation by virtue of the increased solubility of carbon in alpha iron at the stress-relieving temperature, followed by graphite precipitation upon cooling in accordance with the theory advanced by F. B. Foley.

Just why graphite, or for that matter any precipitating phase should precipitate preferentially along planes which have been plastically deformed is not too clear, although it is suggested by Smith and Brambir that it is due to a greater ease of nucleation of the precipitating phase in the plastically deformed regions (14).

After nucleation of graphite once occurs, its continual growth along either slip planes or isotherms appears quite logical. It is believed, however, that temperature fluctuations in a steam line will result in graphite precipitation only at those locations where a graphite nucleus or a strong graphite nucleating effect is present. It is believed that temperature fluctuations will otherwise result only in the solution and reprecipitation of iron carbide. During this time, however, diffusion of the carbide from the carbide-rich areas toward the ferrite areas, which are low in carbon owing to the slow but continual precipitation of graphite at the graphite nuclei, is taking place.

As the graphite nodules grow in size the surrounding ferrite becomes depleted of carbon. The ferrite around the graphite nodules can therefore only precipitate graphite upon each cooling cycle as fast as the carbon in the alpha iron (ferrite) can diffuse from the carbide-rich areas to the ferrite surrounding the graphite.

Based upon the foregoing assumptions, the rate of graphite precipitation for a given set of temperature conditions would appear to depend upon the number of graphite nuclei present and the rate of diffusion of carbon in alpha iron. This is in general agreement with the postulation made by Smith and Miller who stated, "Conceivably, then, graphite in a limited region could bring about, in time, the substantial depletion of carbide within an entire specimen."

Contrary to the belief or implied belief by many that graphite forms directly from the carbide (Fe_3C) phase due to the higher carbon content of this phase as compared to the alpha phase (ferrite), it is believed by the authors that the mechanism by which graphite precipitation occurs is through the gradual solu-

Fig. 18 Luders' Lines on Back of 8-In. Pipe After Cold-Bending

tion of cementite in ferrite followed by the precipitation of graphite from the ferrite.

As additional evidence that the slip-plane graphite shown in Figs. 4 to 7, inclusive, was nucleated along planes where localized plastic deformation had occurred, Fig. 18 is presented. This illustration was taken from the work of William Hovgaard (17), and was brought to the attention of the authors by Messrs. McCutchan and Crocker, of The Detroit Edison Company. Fig. 18 shows a cold-bent 8-in. pipe which has a stress pattern identical to the slip-plane graphitization pattern shown in Fig. 7. The pattern on the pipe in Fig. 18, however, is formed by Luders' lines which resulted from localized plastic deformation during cold-bending, whereas the pattern shown in Fig. 7 is slip-plane graphitization which was nucleated and grew along Luders' lines which also resulted from a critical amount of plastic deformation during the original bending of this pipe.

Assuming therefore that a material is susceptible to general graphitization, and the service temperature is adequate to obtain graphitization, it is suggested that slip-plane graphitization may be encountered in the following locations of critical strain:

1 Entire bend if the bend was made cold without subsequent full heat-treatment.

2 Cold ends of hot bends if not subjected to subsequent full heat-treatment.

The latter postulation, however, has not as yet been verified, although two bends similar to that shown in Fig. 1 have since been removed for examination.

New Theory Suggested for Nucleation of Graphite at Contact Zone of Welded Joints

It is believed that reasonably good evidence has been presented to show that localized yielding or plastic flow has a strong graphite nucleating effect. Once graphite nuclei have been formed it has been theorized that the graphite will grow through the solution of cementite (Fe_3C) in ferrite, followed by graphite precipitation from the ferrite.

The mechanism of the growth of isotherm graphite formed at the extremity of the heat-affected zone of welds, is believed identical to that of slip-plane graphitization. The condition or agent which causes graphite nucleation at the contact zone adjacent to welded joints, however, has not been too clear. It is now suggested that isotherm graphite is also nucleated by stress, although of a somewhat different nature.

Whereas, slip-plane graphitization is believed to be nucleated by plastic flow from an externally applied load, isotherm graphitization is believed to be nucleated by self-compensated stresses known as "tessellated stresses."

The work of Dr. F. Laszlo on tessellated stress was brought to the attention of the authors by C. H. Davy[4] (18, 19, 20, 21, 22).

Although Dr. Laszlo approaches the subject of tessellated stress mathematically, the following extract which qualitatively defines the subject, is taken from his work: "Anisotropy of the single crystals of most materials and the difference between the bulk physical properties of the components of compound solids readily cause internal self-compensated stress systems to develop around such centers as crystals or components of the compound structure, respectively. These self-compensated stress systems are called tessellated stresses."

One of the first conclusions drawn, after the examination of the pipe failure at the Springdale Power Station, was that the failure occurred at the Widmanstätten grain boundaries at a distance from the weld where the temperature from the welding was

[4] Chief Research Engineer, Babcock & Wilcox, Ltd.

just sufficient to cause grain-boundary transformation.[5] It was also this location where grain-boundary graphite of the "chain" type was located and which was the basic reason for failure.

Just outside of the heat-affected zone the metal structure consists of iron carbide or complex iron-molybdenum carbide, embedded in a matrix of alpha iron (ferrite) whereas, just within the heat-affected zone a portion of the alpha iron goes through a phase change to gamma iron, simultaneously absorbing the carbide phase or as much of it as time will permit under the short heating cycle during welding. Rapid cooling of the heat-affected zone may result in the formation of minute amounts of martensite. There is present in the transition zone (zone of isothermal-graphite formation), therefore, alpha iron which has a body-centered cubic lattice, iron carbide or complex iron-molybdenum carbide which has a complex lattice structure, austenite which has a face-centered cubic lattice, and martensite if formed, which goes through a transition structure of body-centered tetragonal and finally to a body-centered cubic lattice.

Along the transition zone adjacent to a weld, therefore, numerous crystal structures, all of which have their own physical constants and which are surrounded by each other, exist at one time or another during the welding cycle. It is common knowledge that a considerable volume change occurs when alpha iron transforms to gamma iron (austenite) and that upon rapid cooling such as occurs in welding the metal does not go back to its exact original dimensions.

It seems quite reasonable to believe therefore that a system of self-compensated stress is developed around crystals or grains at the transition zone between the heat-affected and unaffected parent metal adjacent to a weld, owing to the different physical constants and directional properties of the several different crystal structures which exist at this location during the welding operation.

It is therefore postulated that graphite is nucleated in the contact zone adjacent to a welded joint by self-compensated stresses known as tessellated stresses.

Summary

1 The term slip-plane graphitization has been chosen to describe a condition of graphite segregation which appears unquestionably to have occurred along slip planes where local yielding or localized plastic deformation had previously occurred.

2 Slip-plane graphitization, unlike isotherm graphitization which has been found to be reasonably uniform through the pipe wall and around the pipe circumference, varies markedly from OD to ID and from quadrant to quadrant around the pipe periphery.

3 Slip plane like isotherm graphitization (if severe) is readily visible to the unaided eye after machining or rough-grinding.

4 Gross segregation of "nodular" graphite, whether of the isotherm or slip-plane type, is as effective as chain graphite in reducing the ductility of low-carbon or low-carbon-molybdenum steel.

5 Substantial evidence has been presented to show that plastic deformation is an excellent nucleating agent for graphite of the slip-plane type.

6 The mechanism of graphite growth of either the isotherm or slip-plane type is believed to occur through the solution of the carbide phase in ferrite, followed by the precipitation of graphite from the ferrite.

7 Having presented evidence to show that slip-plane graphitization is nucleated by stress, the postulation is made that isotherm graphite is also nucleated by stress. In this case, however, a self-compensated stress system is believed to be set up, resulting from the difference in directional properties and physical constants of the several different crystal structures which exist in the transition zone adjacent to a welded joint at the time of welding. Self-compensated stresses of this nature are known as tessellated stresses.

[5] This is known as the lower critical temperature (Ac_1) which is approximately 1350 F for carbon-molybdenum steel.

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Graphitization in Some Cast Steels

By A. J. SMITH,[1] J. B. URBAN,[1] AND J. W. BOLTON[1]

Field service and controlled laboratory experiments have demonstrated that it is not possible to insure against graphitization of carbon-molybdenum steel through any of the controlled procedures of melting and deoxidation practice or heat-treatment. It is probable that freedom from graphitization is to be sought through the use of alloy additions which confer greater stability to the carbide. A hypothesis has been put forth concerning the mode of graphitization based upon the observed graphitization behavior of both plain-carbon and carbon-moly steels. Use has been made of this hypothesis to explain the resistance to graphitization of the nickel-chrome-moly steels. The experimental evidence would appear to indicate that the hypothesis is substantially correct.

GRAPHITIZATION characteristics of steels for steam service at elevated temperatures have been a subject of study by many workers since the original failure of a welded carbon-molybdenum steel pipe at the Springdale Station of the West Penn Power Company in January, 1943. Subsequent to that failure graphitization has been found in various classes of wrought and cast products, particularly in weld-affected zones, in the examination of piping systems made by the different power companies.

Happily, the Springdale failure was not disastrous, and to our knowledge none of the other piping systems examined has shown equally extensive graphitization. Nevertheless, it is highly desirable to ascertain means whereby deterioration due to graphitization can be avoided. At the same time it is necessary to preserve in graphitization-resistant material high mechanical properties such as creep strength, soundness, good weldability, and overall economy.

For increasing efficiency in power-plant operation increase in operating temperature is among the promising approaches. Increase to the present 900-950 F has been a development of the past decade. Even now plans are under way for 1000 F operating temperature, and the end of the upswing does not seem in sight. With longer experience acquired as to serviceability of materials at these temperatures, undesirable characteristics have been revealed which hitherto have been unsuspected. From this experience some materials have been shown to be unsuitable for the service. Carbon steel which serves admirably at lower temperatures shows marked loss in mechanical properties at 800 F and is not recommended. Carbon-molybdenum steel which has excellent mechanical properties at 900 to 950 F now is revealed to possess unfortunate graphitization characteristics, limiting its useful application to perhaps 850 F as top operating temperature where welding is a consideration.

In developing and prescribing steels for these higher temperatures, close attention must be paid toward avoiding failures that by chance might occur. All the factors that could contribute to deterioration and failure must be thoroughly investigated. This point is emphasized by occurence of the Springdale failure.

[1] The Lunkenheimer Company, Cincinnati, Ohio.

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The Springdale failure was unexpected. Prior to the failure but one instance of graphitization of low-carbon steel had been reported in the literature (1).[2] Use of molybdenum in steels in high-temperature service led graphitization to be an unsuspected probability since molybdenum is normally considered a relatively strong carbide former and as such likely to promote carbide persistence. There have been arguments for many years on the logic of a double iron-carbon constitutional diagram. Weight of the evidence indicates, however, that iron and graphite are the equilibrium forms, iron carbide or cementite being at best metastable. Since a vast amount of work was required for assuring suitable strength at temperature, possibility of graphitization and other types of structural degradation were neglected.

The Graphitization Problem

The problem of graphitization is complex. It would be quite impossible to study the problem in all of its various phases simultaneously, and it becomes necessary to isolate the various factors and study them individually before any attempt may be made at a comprehensive answer. The problem may be broken down into several subheads, among which are the following:

1 Effects of steel-making practice.
2 Effects of alloying and other elements.
3 Effects of heat-treatment.
4 Effects of fabricating practice.
5 Effects of service operating conditions.
6 Mode of graphitization.
7 Repair of graphitized structures now in service.

Most of the product of the authors' company is in the form of castings. Hence its work in general has been restricted to castings. Certain phases of the foregoing outline have not been touched upon in this work and will be brought up only incidentally. In view of the extensive investigation being carried on by the joint E.E.I. – A.E.I.C. Subcommittee on Graphitization of Piping, no work has been done on possible methods of repair of graphitized structures.

Effect of steel-casting-making practice has been studied here in respect to deoxidation practice.

Fabricating practice, i.e., rolling, upsetting, welding, etc., has not been described in the present report although some work is described on graphitization of structures simulating weld structures.

In the work of the authors' company, some 1300 specimens have been prepared and examined under various conditions of alloy, heat-treatment, and aging-treatment. In this report only those samples will be commented on which were productive of positive evidence in these studies. The others represent duplicate tests for verification of original findings, tests on samples from other heats for the same purpose, cycling tests, etc. The samples chosen for illustration thus should be considered as representing generic classes of structures. The evidence presented for them has been fully corroborated.

Graphitization in Steel Castings

After the Springdale failure many piping systems were examined by the various power companies chiefly by removal of bolt samples from the welded joints. An early discovery of graphiti-

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

zation in a valve casting led to replacement of the valve in the line so that complete and more leisurely examination could be made. The valve was used as a boiler stop valve at nominal operating temperature of 900 F and was in service a period of 44,137 hr, 1909 of which were 950 F or above, $4^1/_2$ at 1000 F.

The casting had been made in 1936 by the older melting practice of "catching the heat coming down" with partial aluminum deoxidation in the ladle, the remainder in the stream on pouring the mold. Total aluminum added was 1.86 lb per ton.

Analyses were made at several locations. Analyses of inlet and outlet ends were as given in Table 1.

TABLE 1 ANALYSES OF INLET AND OUTLET ENDS OF PIPE

	Inlet, per cent	Outlet, per cent
Total carbon	0.31	0.31
Graphitic carbon	0.12	0.04
Silicon	0.40	0.41
Manganese	0.77	0.76
Phosphorus	0.020	0.021
Nickel	0.11	0.05
Copper	0.05	0.04
Molybdenum	0.42	0.44
Chromium	None	0.01
Aluminum (as Al_2O_3) per cent Al	0.012	0.010
Free aluminum per cent Al	0.017	0.018
Total aluminum per cent Al	0.029	0.028

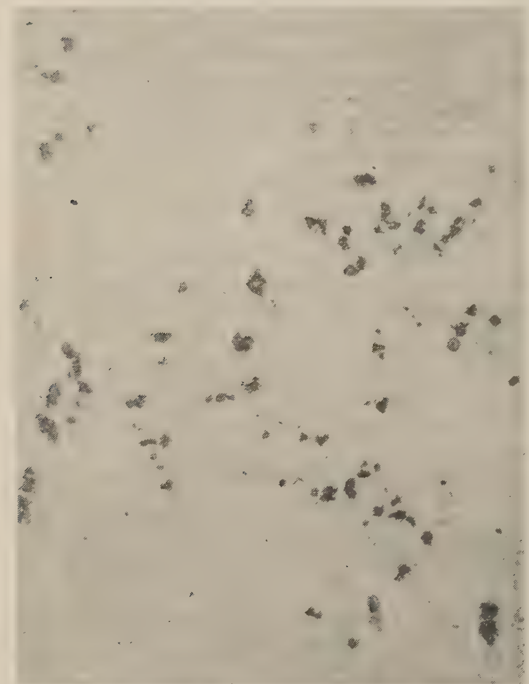

FIG. 1 GRAPHITE IN WELD-AFFECTED ZONE OF CASTING GRAPHITIZED IN SERVICE; ×500

It will be seen that there is no significant difference except in graphite content between the two ends of the valve. Marked visible graphitization had taken place at the inlet end, none could be discovered microscopically in examining several segments of the outlet end. The original cast structure throughout was fine pearlite-ferrite. Graphite in the weld-affected zone was nodular and associated with the carbide areas as shown in Fig. 1.

The wrought-pipe structure was coarse Widmanstäten-ferrite, graphite being similar in amount, type, and distribution to the casting. Just as graphite was not found in the outlet end of the casting neither was it found in the pipe welded thereto. Welding conditions, in so far as they can be judged at this time and presumably as closely as they can be held commercially, were identical.

Service conditions are suspect, yet graphitization was fully reversed in a stop valve from another boiler just adjacent with both high-pressure lines leading to the same manifold. In this other case graphitization was found in the outlet but not in the inlet of both pipe and casting. Truly remarkable circumstances but illustrative of the point that graphitization is in some cases a "hair-trigger" proposition.

Mechanical properties of the graphitized inlet end were investigated, with the finding that there was little impairment of tensile strength and appreciable impairment of elongation and impact. Actual properties were as follows:

Yield strength, psi	57000
Tensile strength, psi	72000
Reduction of area, per cent	54
Elongation, per cent	11.2
Charpy (check tests) ft-lb	7, 9, 6

The tensile bar was made with axis normal to the weld interface, yet the break occurred in the weld metal, not in the graphitized zone. Vee notches in the Charpy bars were placed at the line of graphitization. (Impact value for the base metal of the casting was 18 ft-lb.)

In bend tests made with the bend at the graphitized zone, samples withstood a bend of 90 deg without cracking.

These values show that graphitization has by no means approached the point where it can be considered dangerous, even though it is undesirable. A metal with 6-9 Charpy is hardly "brittle."

Graphitization of Welded Structures

Inasmuch as the Springdale failure was associated with a weld-affected structure, it became incumbent on investigators to include in their agenda of research, studies of this phase of the graphitization problem. Previous extensive studies on welding by the authors' company and others indicated that for an investigation of this type there are too many variables involved for proper assignment of causes of acceleration of graphitization. While normal control of welding insures a commercially desirable result in most cases, absence of exact thermal and composition control leads more to speculation than to scientific fact in considering graphitization rates.

It was evident from our examination and that of others of the Springdale failure that graphitization was most acute in metal which had been raised to the neighborhood of the lower critical temperature. In any work this, then, is a zone of interest. To simulate such structure under closely controlled conditions of temperature, yet free from such influences as composition, gas, slag, etc., present in welds, standard bars ($^7/_8$ in. round × 6 in. long) were heated at one end in a bath at 1600 F, the other end at 600 F, and after holding at temperature 1 hr the bars were quenched. Hardness surveys were then made to determine the critical zones, whereupon the bars were placed in aging furnaces at 975 F, 1025 F, and 1100 F. Graphitization was found to be most rapid at 1100 F, very slow at 975 F, hence 975 F was discontinued as an aging temperature.

In steels prone to graphitization, graphite appeared first in the critical zone, subsequently appearing in the zone quenched from a higher temperature, and later in the zone quenched from a lower temperature. From this it would appear that welding, *per se*, is not responsible for the severe graphitization in the weld-affected zone, but that this condition is induced by heating the material to perhaps just above the critical followed by rapid cooling. To verify this point several samples were heated to various temperatures

TABLE 2 COMPOSITIONS AND MECHANICAL TEST PROPERTIES OF STEEL SAMPLES

Heat	A	B-S	B-A	C-S	C-A	D	E-S	E-A	F
Carbon, per cent	0.30	0.28	0.28	0.21	0.20	0.27	0.26	0.23	0.25
Silicon, per cent	0.42	0.38	0.42	0.42	0.53	0.40	0.27	0.36	0.43
Manganese, per cent	0.65	0.70	0.70	0.75	0.74	0.64	0.61	0.59	0.70
Phosphorus, per cent	0.037	0.034	0.031	0.036	0.034	0.027	0.024	0.020	0.025
Sulphur, per cent	0.044	0.034	0.033	0.033	0.034	0.026	0.026	0.029	0.020
Molybdenum, per cent	...	...	...	0.57	0.56	0.51	0.90	0.40	0.36
Nickel, per cent	...	...	...	...	...	...	0.81	0.93	0.99
Chromium, per cent	...	...	...	...	...	...	0.39	0.53	0.63
Aluminum (total), per cent	0.070	0.008	0.053	0.007	0.058	0.085	0.0030	0.024	0.063
Al_2O_3, per cent	0.023	...	0.019	...	0.018	0.026	...	0.015	0.026
Al in Al_2O_3, per cent	0.012	...	0.010	...	0.010	0.014	...	0.0079	0.014
Yield point, psi	48900	38100	44800	53200	53100	56400	75000	57000	71400
Tensile strength, psi	75800	69000	78100	79400	79000	81200	100700	88300	94400
Elongation, per cent	26.9	32.6	27.2	27.8	29.5	84.4	18.8	24.1	21.0
Reduction in area, per cent	36.7	60.1	49.2	61.0	57.8	46.0	44.9	45.1	41.6

Fig. 2 Graphitization in Carbon-Moly Steel After 500 Hr at 1100 F in Sample, Quenched From 1390 F; ×1000

within this critical range. Graphitization was found to occur most rapidly in samples heated to 1390 F and water-quenched, appearing extensively as early as 500 hr at 1100 F. Time of heating these samples at 1390 F was 3 hr. Graphitization as it appears is shown in Fig. 2.

Welding and its effects on graphitization in piping systems are not to be set aside briefly. However, simple one-bead welds, or restriction to a single set of welding conditions are likely to lead to unsound conclusions. Precise time-temperature conditions (at given locations) and other variables are neither known nor controllable in usual welding techniques. This means that welding studies, to be safely useful, must include many variations in welding practices, hence must be voluminous in respect to experimental manipulations. Extensive investigation is under way to study its full bearing on the problem, the results of which it is hoped will be presented at an early date.

If heating to a zone in the neighborhood of 1390 F (for carbon-molybdenum steels) is critical for promotion of graphitization, as is indicated by the foregoing experiments, and the time at this temperature is not too significant, as the experiments also suggest, then any method of differential heating (welding or other) will produce a zone critical for graphitization. This would call for viewing with caution any method designed to offset the effects of welded structures such as have been proposed, for example, austenitic welding, postnormalizing, etc. Such procedures might serve merely to move the zone to a different location without certain amelioration.

Effects of Deoxidation Practice and Heat-Treatment on Graphitization Tendencies

Certain studies were carried out to ascertain some of the effects of silicon and silicon-aluminum deoxidation and of various structures and treatments on the graphitization tendency of carbon (A.S.T.M. A-216-WC-B), carbon-molybdenum (A.S.T.M. A-217-WC-1), and nickel-chrome-moly (A.S.T.M. A 217-WC-4) steels.

Steels were made up according to regular commercial practice, i.e., $2^1/_2$-ton charges in acid electric furnace. Practice includes ore addition and boil, followed by recarburization and furnace deoxidation by ferrosilicon, ferromanganese, and silicomanganese. Before pouring, tests of such metal lie quietly due to the "silicon deoxidation." ("Silicon-deoxidized" samples were taken at this period.)

When the steel is poured into a large ladle, aluminum is plunged in, 2.4 lb per ton being used. Samples taken therefrom are referred to as "silicon-aluminum-deoxidized."

All samples received a normalize after 5 hr at 1650 F, and were drawn for 5 hr at 1200 F, before subsequent experimentation.

Compositions and room-temperature mechanical test properties of the steels referred to are given in Table 2.

Carbon Steel— WC-B (Heat A). Samples, silicon-aluminum-deoxidized, were heated 3 hr at 1550 F, and quenched into molten tin at 1300 F, at which temperature they were held for 30 sec, 2 min, 20 min, 2 hr, and 5 hr, followed by water-quenching. A complete range of structures was developed, from martensite to lamellar pearlite. Substantial amounts of proeutectoid ferrite were evident in all samples. A comparison sample quenched directly into water from 1550 F, yielded a martensitic structure with small amounts of proeutectoid ferrite appearing only in the boundaries of the original dendrites.

Aging 2000 hr at 1100 F produced full spheroidization, of about the same particle size, and graphitization, to about the same degree, in all samples.

Carbon Steel—Heat B. This metal was divided into silicon-deoxidized and silicon-aluminum-deoxidized portions. A series of isothermal quenching treatments was performed on samples of both coarse and fine austenitic grain size. Coarsening temperature was 1900 F, followed by furnace-cooling to 1550 F, and quenching. Refining was carried out by heating to 1550 F, followed by direct quenching. Quenching was carried out in a controlled tin bath.

Temperatures employed were those used in earlier work on car-

bon-molybdenum steel (4) and which gave the structures for carbon moly shown in Table 3.

TABLE 3 CARBON-MOLYBDENUM STEEL STRUCTURES, HEAT B

	Structure (carbon moly)	Transformation temperature, deg F
I	Coarse pearlitic	1200
II	Coarse mixed pearlitic-acicular	1000
III	Coarse acicular	800
IV	Coarse martensitic	500
V	Fine pearlitic	1200
VI	Fine mixed pearlitic-acicular	1000
VII	Fine acicular	800
VIII	Fine martensitic	500

The acicular structure of the carbon-moly steels is not a member of the plain-carbon-steel series, hence was not developed in this experiment. Also it was not possible to develop a martensitic structure at 500 F in this carbon steel. However, since we are more concerned with comparative temperature levels in processing these materials than in the actual structures, it was thought desirable to retain the quenching-bath temperatures used earlier and on which fairly complete data were available.

TABLE 4 CARBON-STEEL STRUCTURES

	Structure (carbon steel)	Transformation temperature, deg F
I	Coarse pearlitic	1200
II	Coarse mixed pearlitic-bainitic	1000
III	Coarse upper bainitic	800
IV	Coarse lower bainitic	500
V	Fine pearlitic	1200
VI	Fine mixed pearlitic-bainitic	1000
VII	Fine upper bainitic	800
VIII	Fine lower bainitic	500

In the carbon steels Table 4 gives the structures approached. The structures shown in Figs. 3 to 6 respectively, are of the coarse series, and it may be seen that they do not correspond in all respects to the type description. This is because of the difficulty of securing a homogeneous austenite in cast materials. Because of these inhomogeneities there was marked separation of proeutectoid ferrite similar to that developed in conventional heat-treatment of castings. This alters the effective carbon concentration in the carbide areas and affects the time of reaction at a given temperature.

All of the silicon-aluminum-deoxidized samples graphitized within 1000 hr at 1100 F. Graphitization was most general in the type VIII sample (fine lower bainite). Size of graphite particles was about the same in all samples.

Of the silicon-deoxidized samples the following graphitized at 1100 F within the time indicated:

I	Coarse pearlitic	2000 hr
II	Fine pearlitic	6000 hr
VII	Fine upper bainitic	4000 hr

The following were resistant for the duration of the test, 6000 hr:

II Coarse mixed pearlitic bainitic
III Coarse upper bainitic
IV Coarse lower bainitic
VI Fine mixed pearlitic-bainitic
VIII Fine lower bainitic

The foregoing suggests that fine and uniform distribution of carbide may lessen initial graphitization tendency. Since such distribution indicates less steep carbon-concentration gradients, such influence might be expected, at least in initiation of the reaction. However, once action is started, a more general nucleation might well cause more general graphitization.

In case of this carbon steel, silicon-aluminum deoxidation favors graphitization, but the absence of aluminum does not always prevent it.

The two deoxidation practices influence hardenability, silicon deoxidation tending toward coarser grain and deeper hardening. Since bainitic structure in silicon deoxidized graphitized, the effects of differences in hardenability characteristics (as influenc-

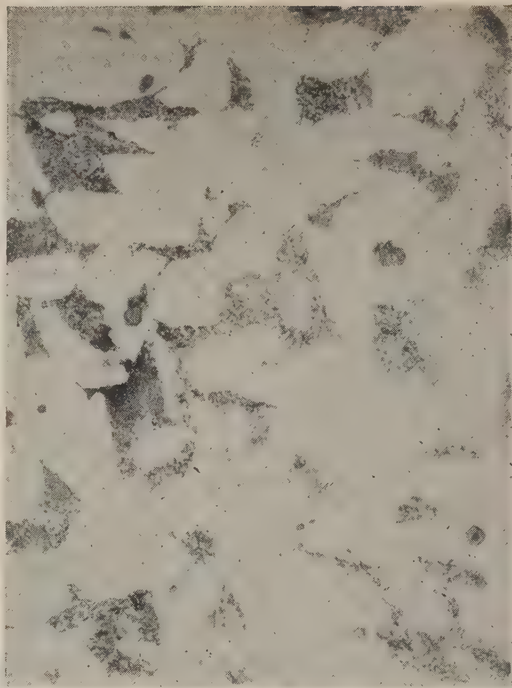

Fig. 3 Coarse Pearlitic Carbon Steel: ×500

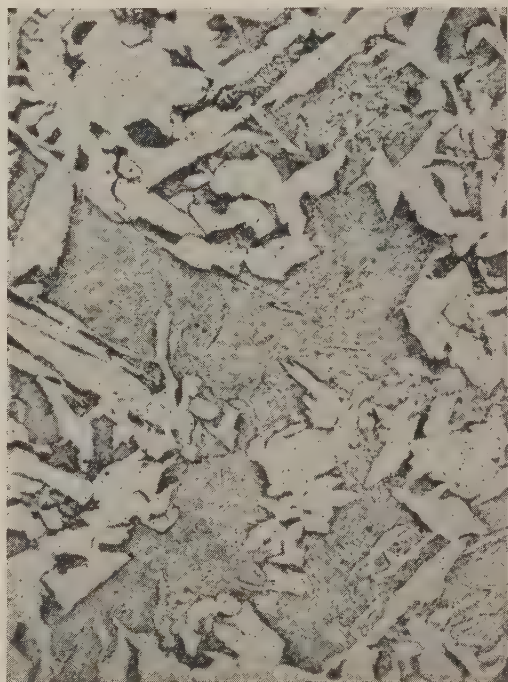

Fig. 4 Coarse Mixed Pearlitic Bainitic Carbon Steel; ×500

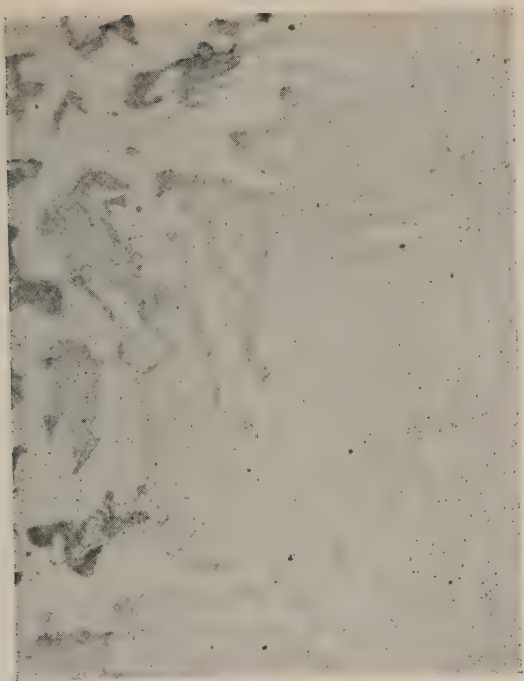

FIG. 5 COARSE UPPER BAINITIC CARBON STEEL; ×500

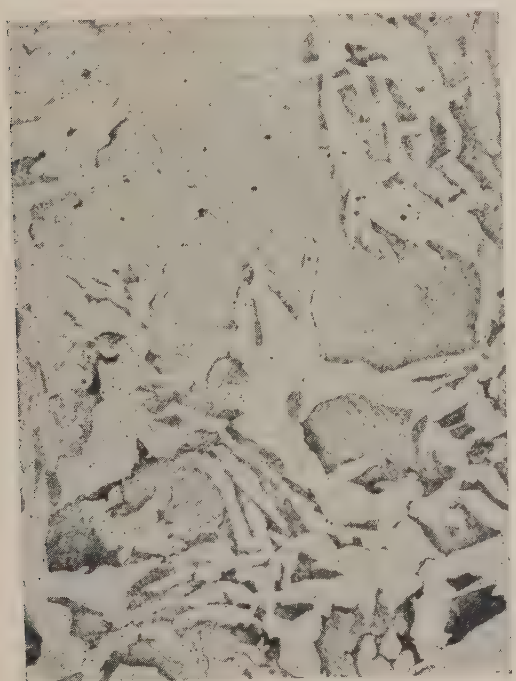

FIG. 6 COARSE LOWER BAINITIC CARBON STEEL; ×500

ing graphitization) may be somewhat discounted. However, a number of these points require more extensive study before firm generalities can be safely made. Subject to this qualification, the experimental work on carbon steel (WC-B) suggests the following:

(a) Silicon kill retards graphitization tendency, but does not safely prevent it.

(b) Carbon-concentration gradients appear to promote the earlier formation of graphite.

(c) Greater carbon homogeneity may promote more general graphitization, once the reaction is initiated.

(d) Coarser austenitic grain and deeper hardening characteristics tend toward greater resistance, but may not assure it. (This might well be independent of agencies promoting such structure.)

Conventional normalize and anneal, under production conditions permitted graphitization in both silicon and silicon-aluminum-deoxidized conditions. Silicon-aluminum deoxidized graphitized in less than 4000 hr; silicon deoxidized in 6000 hr.

Principal structures aged at 1100 F were aged in duplicate at 1025 F. Of the silicon-aluminum deoxidized all had graphitized at 1000 hr, with the exception of the mixed pearlitic-bainitic, again showing the resistance of this structure to graphitization. Otherwise little difference was to be noted from 1100 F aging.

Carbon Moly (WC-1) Steel (Heat C). This material was divided into silicon-deoxidized and silicon-aluminum-deoxidized portions. Samples were put into the following structural conditions, and graphitization time as it appeared in the silicon-aluminum deoxidized samples which developed graphite within 6000 hr is as given in Table 5.

TABLE 5 CARBON-MOLYBDENUM STRUCTURES; HEAT C

Structure (carbon moly)	Graphitization
Coarse pearlitic	2000 hr
Coarse-pearlitic-acicular	None
Coarse acicular	2000 hr
Coarse martensitic	None
Fine pearlitic	6000 hr
Fine pearlitic-acicular	None
Fine acicular	None
Fine martensitic	None

In these structures there was considerable separation of proeutectoid ferrite leaving the carbides as islands in a ferritic ground mass. This appears to be the common pattern developed in carbon-moly steels. In looking for graphitization such structures are suspect. On the other hand, the structure description describes the state of aggregation of the carbide to much better degree than was found for the carbon steels.

No graphitization was found in the corresponding silicon-deoxidized samples, within 6000 hr.

The grain size for the silicon-deoxidized portion was 2-3 with normal McQuaid-Ehn as shown in Fig. 7.

In heat D, silicon-aluminum, killed, similar samples all graphitized within 6000 hr or less. Grain size and normality were similar for both silicon-aluminum-killed heats, i.e., 6-7 and markedly abnormal McQuaid-Ehn, Fig. 8.

Production heat-treatment of normalize from 1650 F yielded graphitization within 2000 hr in the silicon-aluminum-deoxidized, none within 6000 hr in the silicon-deoxidized. However, graphitization was found in 6000 hr in a silicon-deoxidized sample held 5 hr at 1650 F, and furnace-cooled as shown in Fig. 9. The sample illustrated is the only one that has been developed experimentally in our work showing an approach to chain graphite. All others have been nodular. The correspondingly treated silicon-aluminum-deoxidized sample showed nodular graphite.

The results on carbon moly (WC-1) indicate that silicon deoxidation resulting in coarse-grain normal McQuaid-Ehn structures retards graphitization, but does not always prevent it, as was established also in case of carbon (WC-B) steel. It is apparent that as compared with carbon steel the molybdenum content of WC-1 increases carbide persistence.

In another experiment on a silicon-aluminum-deoxidized sample, heat C was held 5 hr at 1650 F and furnace-cooled, then

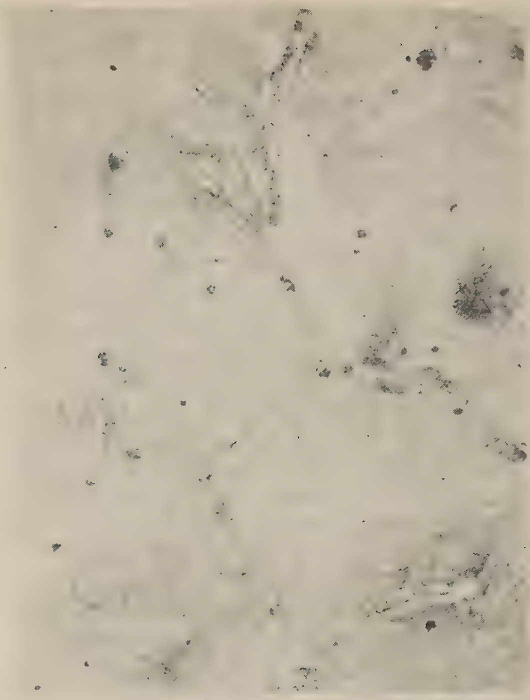

Fig. 7 McQuaid-Ehn Case Heat C Silicon Deoxidized; ×500

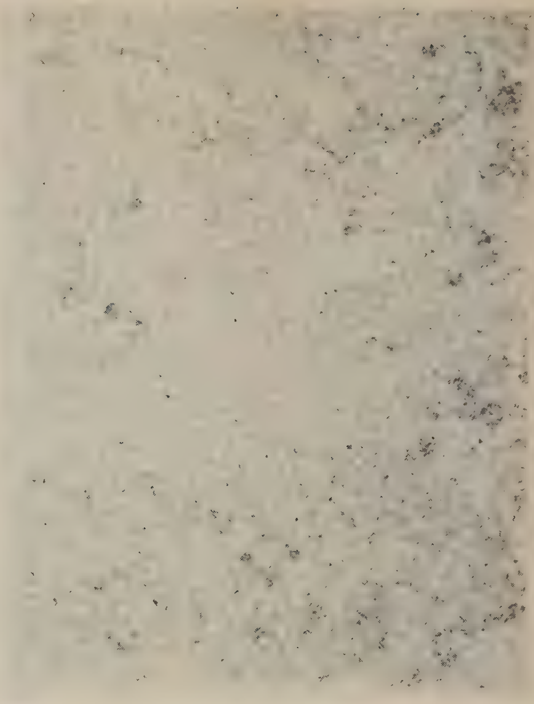

Fig. 8 McQuaid-Ehn Case Heat C Silicon-Aluminum Deoxidized; ×500

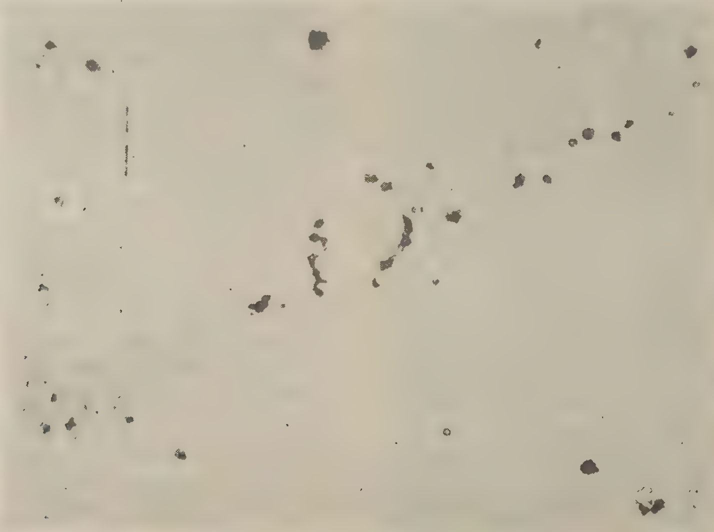

Fig. 9 Graphitization in Silicon-Deoxidized, Heat C; ×500

drawn for 10 hr at 1300 F; graphitization being produced within 4000 hr at exposure to both 1025 and 1100 F.

This treatment was employed to determine the effectiveness of high-temperature annealing in order to stabilize the carbides in somewhat different form as suggested by Smith and Miller (5).

Consideration of these and other studies, and of the published literature and field data, indicates that carbide persistence is influenced by many variables; some whose general influence seems clear, others which must be better known and evaluated before conclusions can be drawn.

Since strength at temperature is fully as important as structural stability, creep tests were run at 950 F on samples from heat D, heated for prolonged periods at the suggested stabilizing temperature, i.e., 110 hr at 1320 F, after normalizing from heating at

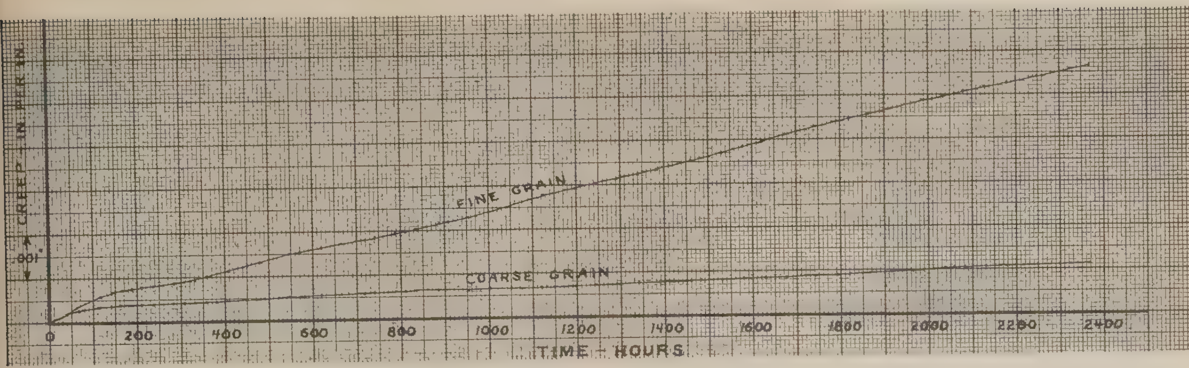

FIG. 10 CREEP CURVES FOR CAST CMO STEEL

1550 F in one case and from heating at 2200 F, followed by furnace-cooling to 1550 F in another to yield both fine- and coarse-grained structures in a highly spheroidized condition. The resulting creep curves are shown in Fig. 10. Tests were carried out for a period of 2400 hr under loads of 12,000 psi. In both cases graphitization was found after test. The test temperature of 950 F was the lowest at which graphitization has been produced in this laboratory. At least in this steel, treatment at 1320 F has not insured against graphitization. Structures of the samples before and after test are shown in Figs. 11 to 14, inclusive.[3]

Nickel-Chrome Moly (WC-4) Steel. Samples of silicon-deoxidized and silicon-aluminum-deoxidized metal, heat E, were prepared. Each series was placed in the eight structural conditions listed for carbon moly (WC-1) given previously. After exposures to a maximum of 6000 hr at 1100 F, no graphization appeared in any sample.

In another series of experiments on a silicon-aluminum-deoxidized heat, heat F, with samples placed in the same structural condition, no graphite was found after 7976 hr at 1100 F. Duplicate experiments at 1025 F also failed to reveal any evidence of graphite development.

As in the carbon-moly steels, creep tests on nickel-chrome-moly steel were carried out with identical prior heat-treatment including commercial normalize and draw conditions. No graphitization was found in the bars after test. Structures before and after testing were as shown in Figs. 15 to 18, inclusive.

It will be noted that a number of the structural and deoxidation conditions in which WC-4 was placed correspond to those in which

[3] "Coarsened" means coarsened austenitic grain size. This treatment results in greater carbon homogeneity. Thus the resulting transformation product aggregations (Figs. 11 to 14, inclusive) give appearance of finer "grain" in the coarsened sample photomicrographs. There is no anomaly. The reader is cautioned not to confuse "pearlitic" grain-size appearance with austenitic grain which is not shown.

FIG. 11 COARSENED WC-1 BEFORE CREEP; ×500

FIG. 12 REFINED WC-1 BEFORE CREEP; ×500

Fig. 13 Coarsened WC-1 After Creep; ×500

Fig. 14 Refined WC-1 After Creep; ×500

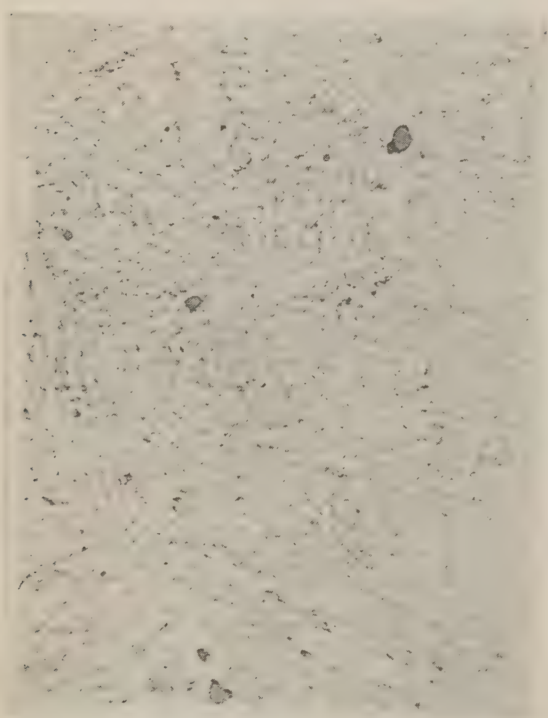

Fig. 15 Coarsened WC-4 Before Creep; ×500

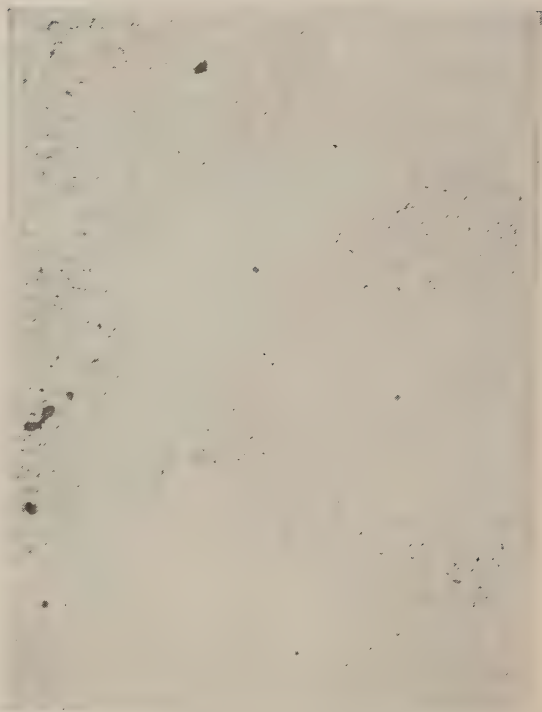

Fig. 16 Refined WC-4 Before Creep; ×500

Fig. 17 Coarsened WC-4 After Creep; ×500

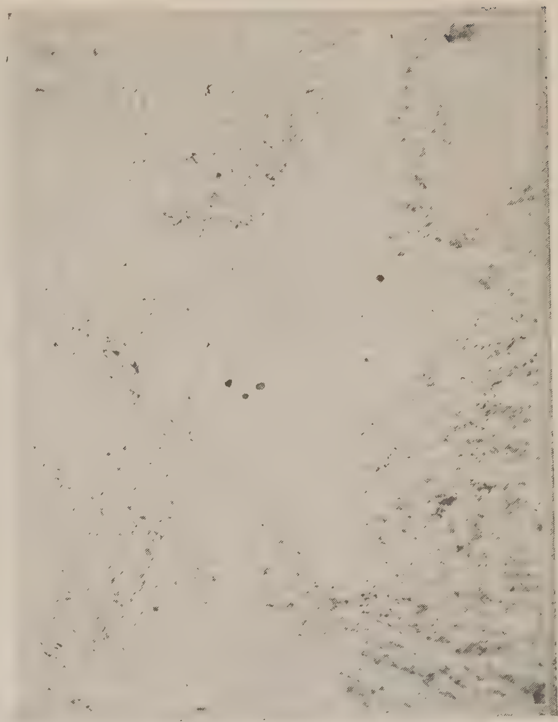

Fig. 18 Refined WC-4 After Creep; ×500

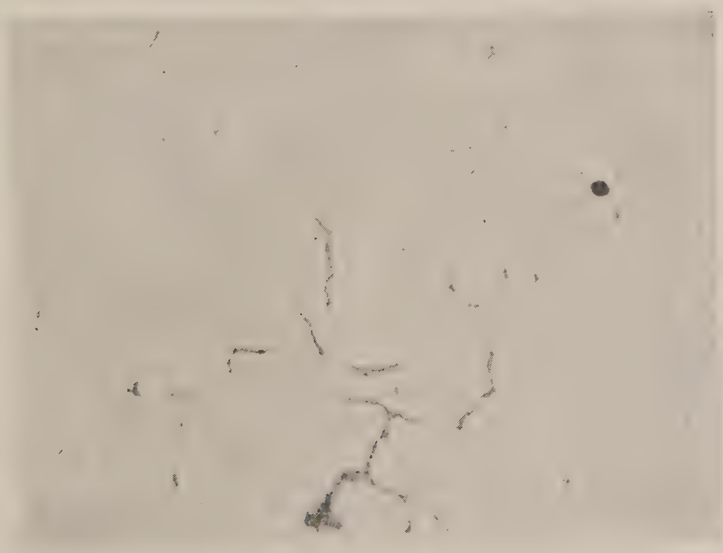

Fig. 19 Pipe Removed From Service and Given Cycling Treatment; ×500

rapid graphitization was found in carbon-moly (WC-1) steel. Work on WC-4 was confirmed by study of several heats. (Differential quench samples also showed no graphitization.)

Mode of Graphitization

Emerson (6) has distinguished between two types of graphite formation, "nodular" and "chain." Our study suggests the desirability of including a third type, "flake," a formation shown in Fig. 19. Graphite formations and groupings in reality are in three dimensions, not two, as they appear in a photomicrograph. This must be borne in mind in any attempt to evaluate apparent orientations in respect to their evolution and effects.

Emerson (6) has illustrated the chain formation in such a way as to leave little doubt as to its distribution and possible conse-

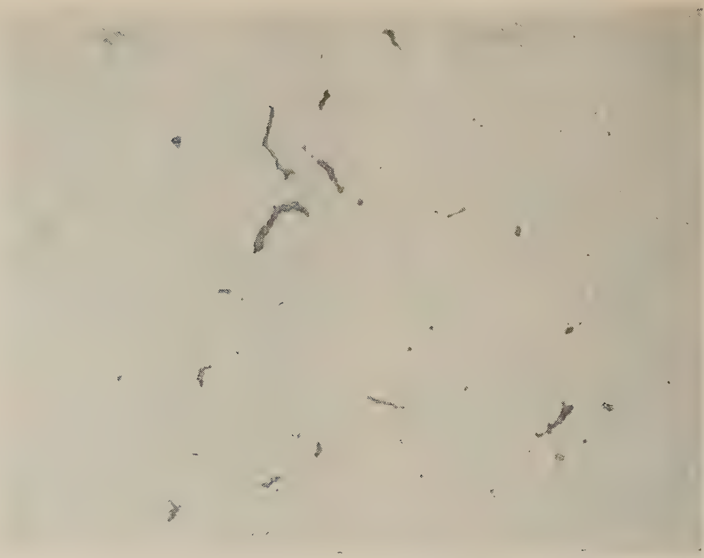

Fig. 20 Casting as Removed From Service; ×500

quence. A flake may be but a progressive development of a chain although our studies suggest otherwise.

For the purpose of identifying graphite, polarized light has been very useful. Graphite crystallizes in the hexagonal system and shows extreme pleochroism and birefringence in polished section (7). Using a first-order red compensator plate, the change from red to green with stage rotation is striking and distinctive. This and other characteristics of graphite have been observed before in this laboratory in the course of its studies on cast iron. Hence the tool fits very nicely into investigations of graphitization in steel.

The color change at extinction for a particular crystal depends, among other factors, on its orientation with respect to the plane of polarization of the light. Differently oriented anisotropic crystals will show extinction at different angles of stage rotation. Nodular graphite shows a mottling of red and green with a first-order red compensator. This indicates that it is formed of many small crystallites of various orientations. However, in many of the graphite formations described as flake, uniform extinction of the whole flake was shown at the same angle. This was evident in the flakes shown in Fig. 19. This suggests that the formation is of uniform crystal orientation and is of the character of a single crystal, bearing close similarity to the flakes found in cast iron. The massive portion attached to one of the flakes of Fig. 19 shows the mottled effect and is definitely polycrystalline. This sample was from a segment of pipe removed from service and given a cycling treatment for 800 hr between 950 F and 1100 F. Fig. 20 is from a sample taken from a casting as removed from service. It also shows the single crystal or flake behavior.

Were these flakes built up from a chain it is improbable that there would be crystallographic uniformity along the whole chain. The Widmanstätten relationship which exists between matching planes of precipitate and matrix can be met in several different ways with respect to the plane of polish, so there is no requirement that all graphite particles separating from a single ferrite grain should have the same orientation with respect to the plane of polish even though the growth mechanisms are identical. Thus a plate of graphite built up from a series of particles originally in chain form would be expected to be broken up crystallographically into a number of segments which would be clearly revealed by examination in polarized light.

There is one point of similarity between flake and chain formation which seems significant in attempting to explain the mode of graphitization. As shown in Figs. 19 and 20, in Emerson's illustrations of chain formation (6), and also in certain of the illustrations of Weisberg (8) and Kerr and Eberle (9), graphite in these instances is located in the ferrite grain boundaries. Nodular graphite usually is associated with the carbide areas. While chain and flake graphite may be attached to carbide areas, perhaps at other locations than the plane of polish, the formations extend out into regions of primary ferrite.

The general mechanism of graphite formation involves two steps, i.e, formation of a nucleus or initial graphite particle, and growth of the particle by diffusion of carbon to it.

The first step necessitates a local concentration of carbon together with some nuclear influence. The nuclear influence may be developed in many ways, submicroscopic impurities or other discontinuities probably playing a major role. A grosser sort is illustrated in Fig. 14 where a graphite flake is seen radiating from a manganese-sulphide inclusion. The second step requires diffusion of carbon toward the first formed particles from carbides or from other graphite particles less capable of sustained growth. As the region about the growing particle becomes depleted in carbon, migration through the ferrite occurs. Ferrite dissolves a certain amount of carbon (varying with temperature) and local depletion in one area results in carbon diffusion from another, thus maintaining a statistical balance. The diffusion distance is extreme in some cases as in the chain or flake graphite lying at some distance from the carbide "reservoirs."

Thus there are two approaches in the problem of preventing deterioration: Promotion of relative carbide stability; and the lessening of the possibility of diffusion of carbon through ferrite.[4]

Other factors being equal, tendency toward graphitization in the iron-carbon series increases with increasing carbon content. Hypereutectic cast irons (with primary cementite) graphitize in the freezing range with the formation of kish or free graphite in the partly liquid metal. The gray irons graphitize rapidly in the

[4] "Stability," as used in this paper implies resistance to breakdown over a normally expected service life. It is not possible to say that any carbide is stable in the true equilibrium sense. However, the term is descriptive when used in its practical sense.

interval between the eutectic and eutectoid temperatures. This graphitization may continue even below the eutectoid under slow-cooling conditions or in the presence of sufficient silicon.

In the case of malleable iron, the composition is so chosen that there is no graphitic carbon as cast. Subsequent treatment is such that the cementite is graphitized both above and below the eutectoid-transformation range. The mechanism of carbon diffusion through the ferrite toward graphite particles is perhaps most vividly shown by the formation of "bull's eyes," graphite particles surrounded with ferrite in a general pearlitic matrix.

Graphitic steel is a product of commerce. Austin's work (10) shows how readily high-carbon steel can be graphitized in quite a short time.

In normal carbide areas the concentration of carbon is high, well above that of the average for the whole sample. As more and more proeutectoid ferrite is present in a given analysis, it is readily seen that the carbon concentration in the carbide areas becomes much greater, ultimately approaching the eutectoid composition. With such concentration gradients a graphitizable steel will usually graphitize most readily regardless of state of aggregation of carbide. Thus steels with large areas of free ferrite are more suspect for serious graphitization than steels of a more homogeneous character even though the more homogeneous steels may ultimately graphitize.

The problem of stabilizing the carbide and preventing diffusion in ferrite is complex but its solution is necessary for insuring against service failures due to graphitization. All of the carbide formers have a solubility in ferrite and under proper circumstances may dissolve even when carbon is present, e.g., in the presence of a stronger carbide former. In a straight molybdenum steel the molybdenum undoubtedly is associated in large measure with the carbide. Hence with little distortion of the ferrite lattice, the interstitially dissolved carbon is able to diffuse at will. The binding forces of the molybdenum are evidently satisfied fully as well in an iron-molybdenum carbide as in a straight molybdenum carbide with the result that carbon can be given up by the molybdenum to be subsequently precipitated as graphite by the ferrite without too greatly altering the state of free energy of the system.

Although good creep properties have been shown for carbon-free iron-molybdenum alloys (11) in the normal steels, molybdenum is depended upon for conferring strength at elevated temperatures in the formation and distribution of a stable and resistant carbide. Yet it is known that certain amounts are dissolved in the ferrite. In a higher-carbon steel, 0.70 carbon, Bowman, Parke, and Herzig (12) have shown the amount in the ferrite to be of fairly low order, possibly less than 20 per cent of the total. Partition data on steels such as described in this report, are not available. It may be reasoned, however, that the partition coefficient would be changed with composition and with lower carbons more molybdenum would dissolve in the ferrite. Carbon-molybdenum steel castings for ET service contain some 30 to 50 per cent more carbon than carbon-moly piping, and hypothetically, the partitionment of molybdenum in ferrite in the piping would be greater than in the castings. Yet the piping materials graphitized perhaps more readily than the castings, at least such appears to be the case from the service record. This suggests that increasing molybdenum dissolved in ferrite would not bring about assurance of freedom from graphitization.

Creep properties at elevated temperatures are markedly altered by carbide pattern and distribution. Hence it would appear that the superior creep properties of molybdenum-bearing steels are dependent in large part upon the molybdenum or iron-molybdenum carbide and are not to be sought in steels alloyed with stronger carbide formers in which the molybdenum would be forced to much greater extent into the ferrite, leaving the new alloying element, such as vanadium or titanium, to form the principal carbides. Such steels have not been shown to possess superior elevated-temperature properties in the absence of molybdenum and improve molybdenum steels to but minor extent. Thus such carbides cannot be looked on as a source of strength at such temperatures.

To protect against graphitization by use of alloys it appears that greater success might be achieved by use of weaker carbide formers than molybdenum or by noncarbide formers, which would dissolve in the ferrite and force more molybdenum into the carbide. By distortion of the iron lattice with dissolved elements, diffusion is interfered with and at the same time the ferrite is strengthened. The molybdenum being forced further into the carbide should confer additional elevated temperature strength to the carbide. Copper, cobalt, and nickel do not form carbides and might be found to perform this particular function. Chromium would show similar behavior since, although a carbide former, it is weaker in this respect than molybdenum. It is probable that combinations of these would be of more effect than when used singly, in accordance with general knowledge of the behavior of alloying elements.

The foregoing reasoning is hypothetical. Partition data are not as yet available by which the hypothesis can be fully evaluated. However, behavior of steels of the type of WC-4, nickel-chrome-moly lends substantiation to its correctness. This steel could not be graphitized under any set of conditions in the laboratory and has not graphitized in service.

Furthermore, it has creep properties superior to carbon-moly steel as is indicated by the various creep studies. Creep tests with various comparable structures confirm this. (They are not a part of this report since they in themselves have little bearing on the graphitization problem.) The nickel may play an important part in this stabilization or it may be looked upon more as a safety measure since, not being a carbide former, it opposes ferrite solution of the other elements but would not in itself be partitioned. The finding by Smith, Miller, and Tarr (13) that a molybdenum steel containing 0.17 chromium was readily graphitizable and by Hoyt (14) that a 0.50 chromium steel could be graphitized leads to some question whether steels containing this chromium alone (as has been proposed) would provide complete assurance against graphitization. It remains for subsequent work to demonstrate conclusively whether or not such additions can be relied on for this purpose, over a wide range of manufacturing, fabrication, and operating conditions.

Discussion of Results of Studies

While the Springdale failure was disconcerting, possibilities of graphitization probably should not have been unexpected. In the high-carbon steels the problem had been studied rather extensively, especially by Austin and co-workers. A failure in straight low-carbon steel also had been observed under quite similar circumstances.

In detection of early stages of graphitization the most useful tool is the microscope. To observe graphite in a polished section a special technique is involved for its retention in the section, the graphite being readily removed in the the normal polishing operation. Undoubtedly, were graphite present it would have been preserved with inclusion polishing. However, inclusions almost invariably are studied before extended heat-treatment rather than after. This probably accounts for graphite not being detected even in creep samples until its presence was suspected.

The major question that arises is whether or not any carbide is stable under all conditions of temperature and pressure. It is entirely possible that none is, ultimately; yet certain carbides may prove sufficiently persistent to endure for the expected service life of the parts without undue deterioration.

It is not known what conditions are required for carbide stability. Molybdenum is normally considered a strong carbide former yet failure took place in a molybdenum-bearing steel. Chromium is less strong as a carbide former than molybdenum yet a nickel-chrome-moly steel proved quite resistant to graphitization experimentally and evidence from field service supports the laboratory findings. Results of others suggest that sufficient percentages of chromium likewise might be proved effective. The hypothesis has been advanced in the present paper that ferrite plays an active part at least in chain or flake graphitization. It must play a part in nodular graphitization also. If this hypothesis is correct, then an explanation of the stability of the nickel-chrome-moly steel may be found in the fact that the nickel is dissolved in the ferrite, and, with a stronger carbide former, molybdenum, being present than the chromium, the chromium likewise is largely dissolved in the ferrite, resulting in the conferring of stability to the ferrite against carbon rejection in the form of graphite.

It is believed that if a steel initially is seriously unstable from its composition, no amount of "doctoring" will insure sufficient stability for prolonged service. Thus production of a straight carbon-moly steel of coarse grain size and normal McQuaid-Ehn structure is not sufficient to prevent graphitization over normal service periods although the graphitization rate may be very considerably slower or its initiation require more unfavorable fabricating and operating conditions. A desirable initial structure likewise may retard but will not insure against graphitization. This has been shown not only for different distribution of constituents but also for their constitution, as, contrary to the suggestion of some workers, steels annealed at temperatures which would promote formation of a presumably more stable molybdenum carbide subsequently graphitized at lower temperatures.

It appears also that little is to be safely hoped for by modification of weld practice short of normalizing the whole assembly before installation since any type of localized heating will result in some zone being heated into the temperature range critical for graphitization.

Thus far, critical field graphitization has been found adjacent to welds. In many instances, however, noncritical graphitization has been found elsewhere in piping and castings. In no case that has come to the authors' attention has graphitization proceeded as far as at Springdale. Thus it is perhaps fortuitous that critical graphitization has not been found in regions not associated with welds. The recent finding at Springdale of graphite precipitated on slip planes well removed from a weld leads to suspicion that finding of such critical graphitization may be made in the not too remote future.

The use of alloys for stabilization appears to offer the most promising hope against such a condition. As indicated, it may be as fully necessary to stabilize the ferrite as to stabilize the carbide. Both laboratory and field experience appear to indicate that a steel conforming to A.S.T.M. A-217-WC-4 is an approach in this direction.

Summary

From many samples studied both from field service and controlled laboratory experiment it has not been shown possible to insure against graphitization of carbon-molybdenum steel through any of the controlled procedures of melting and deoxidation practice or heat-treatment. The evidence would appear to indicate that freedom from graphitization is to be sought through the use of alloy additions which confer greater stability to the carbide. Cases cited in the literature of graphitization of low-alloy chrome-molybdenum steel at conventional service temperatures lead to little assurance of freedom from graphitization over the normal service life by the use of such low chromium additions to the moly steels alone. Critical percentage ranges, if existent, should be revealed by co-operative work being undertaken by various agencies. Extensive experimental data lead to such assurance for the nickel-chrome-moly steels within the present specification range of A.S.T.M. A-217-WC-4. Experiments with carbon-moly steel have been duplicated for the most part with nickel-chrome-moly steel with special reference to those conditions which are critical for carbon-moly steel, silicon-aluminum kill resulting in a highly abnormal steel structure. In no case has graphitization been found in the nickel-chrome-moly steel.

A hypothesis has been put forth concerning the mode of graphitization based on the observed graphitization behavior of both plain-carbon and carbon-moly steels. Use has been made of this hypothesis to explain the resistance to graphitization of the nickel-chrome-moly steels. The experimental evidence would appear to indicate that the hypothesis is substantially correct.

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Discussion

F. Eberle.[5] One of the most important facts developed in this paper is the experimental evidence of the fundamental instability and, incidentally, also of the relatively great stability of straight silicon-killed McQuaid-Ehn coarse-grained and normal carbon-molybdenum steels. At 975 F, which may be considered as an upper temperature limit for the use of this steel in steam-

[5] Research Metallurgical Engineer, The Babcock & Wilcox Company, Barberton, Ohio.

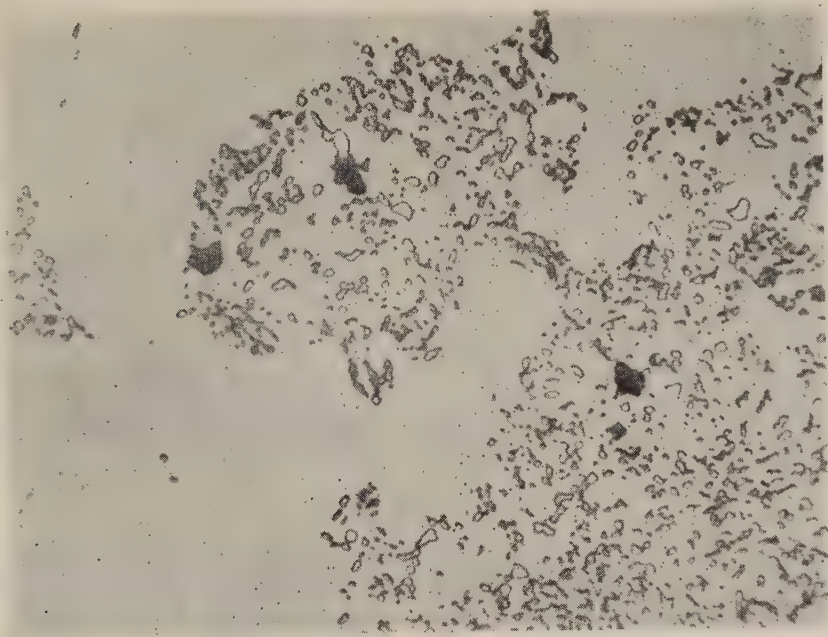


FIG. 21 LOW-TEMPERATURE END OF WELD-AFFECTED ZONE IN CREEP-TEST SPECIMEN CASE 3; ×2000

plant operation, the authors found the rate of graphitization to be very slow and therefore they conducted most of their experiments at 1100 F. Even at this extreme and impractical temperature only one out of the described ten test samples of straight silicon-killed carbon-molybdenum steel produced graphite within 6000 hr. The sample which did graphitize had been in the 1650 F-furnace-cooled condition in which the steel is most susceptible to graphitization.

As a further illustration of the fundamental instability and the relatively great stability of McQuaid-Ehn normal carbon-molybdenum steel, we present the following results of tests and examinations conducted in the laboratory of the writer's company:

Case 1. A creep-rupture specimen containing a 1200 F, stress-relieved, standard, multiple-pass arc weld in the center of the gage length was found free of graphite after 11,400 hr at 950 F under 20,000 psi which is $2^1/_2$ times the allowable working stress. The steel was silicon-aluminum-deoxidized (0.75 lb. of aluminum per ton) and was McQuaid-Ehn normal- and medium-grained (grain size No. 6). Prior to the welding the steel had been normalized at 1650 F.

Case 2. A creep-test specimen made of 1650 F normalized unwelded silicon-aluminum-killed carbon-molybdenum steel (0.5 lb. of aluminum per ton) also was found free of graphite after 8380 hr at 950 F under 10,000 psi which is $1^1/_4$ times the permissible working stress. This steel, too, was McQuaid-Ehn normal- and medium-grained (grain size No. 4-5 with some No. 6).

Case 3. A creep-test specimen made of the same material as previously described (case 2), but containing a standard multiple-pass arc weld, stress-relieved at 1200 F, in the center of the gage length, was found to contain a very few tiny graphite nodules in the low-temperature end of the weld-affected metal after 8380 hr at 950 F under 10,000 psi which again is $1^1/_4$ times the allowable working stress. These graphite nodules were so small that they could be recognized only at very high magnification and then identification was not absolutely assured (see Fig. 21 of this discussion).

Case 4. Another creep-test specimen, again made of the same material as used for the last described two samples, but containing repeatedly arc-welded and unstress-relieved shoulder pads at the ends of the gage length, was found totally free of graphite after 7002 hr at 950F under 10,000 psi which is $1^1/_4$ times the permissible working stress.

Case 5. A creep-test specimen made of straight silicon-killed McQuaid-Ehn normal- and coarse-grained carbon-molybdenum steel (grain size Nos. 2-4) in the 1650 F annealed (slow-furnace-cooled) condition and containing a standard multiple-pass arc weld, stress-relieved at 1200 F, in the center of the gage length, was found free of graphite after 5800 hr at 1000 F under 10,000 psi which is twice the working stress permitted by the Boiler Code.

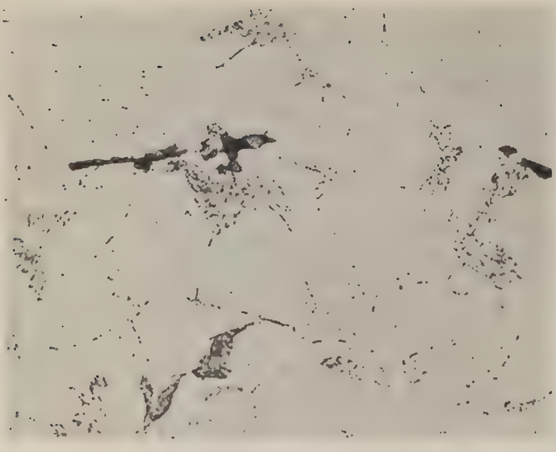
Case 6. An identical creep-test specimen as just described, but with an unstress-relieved weld in the center of the gage length, was found to contain scattered medium-sized graphite nodules in both the weld-affected and weld-unaffected metal after 13,420 hr at 1000 F under 10,000 psi, which again is twice the allowable working stress, Fig. 22 (*a* and *b*).

Case 7. Another creep-test specimen of the same annealed straight silicon-killed carbon-molybdenum steel as previously described (cases 5 and 6) and again containing a 1200 F stress-relieved standard multiple-pass arc weld in the center of the gage length, also displayed scattered medium-sized graphite nodules in both the weld-affected and weld-unaffected metal after 13,440 hr at 1000 F under 10,000 psi. Number and size of the graphite nodules were not markedly different from those in the unstress-relieved weld specimen of case 6, Fig. 23 (*a* and *b*).

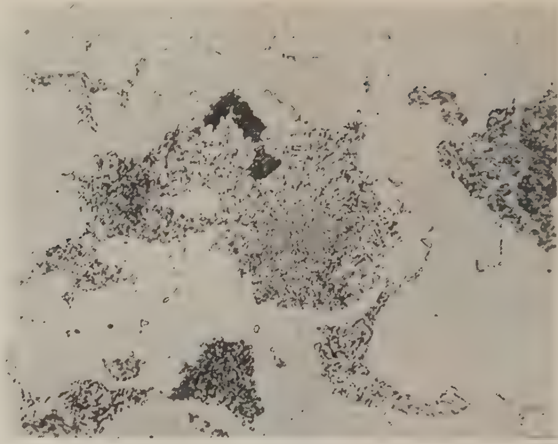
Case 8. An unwelded section of a refinery tube of straight silicon-killed McQuaid-Ehn normal and coarse-grained carbon-molybdenum steel was found free of graphite after 39,420 hr service at 1050 F under 2140 psi. The steel of this tube contained also 0.61 per cent nickel.

Case 9. Another straight silicon-killed carbon-molybdenum steel refinery tube, containing 0.45 per cent nickel, likewise was found free of graphite after 30,660 hr of service at 1050 F to 1070 F under 2080 psi.

Case 10. A third, straight silicon-killed, carbon-molybdenum



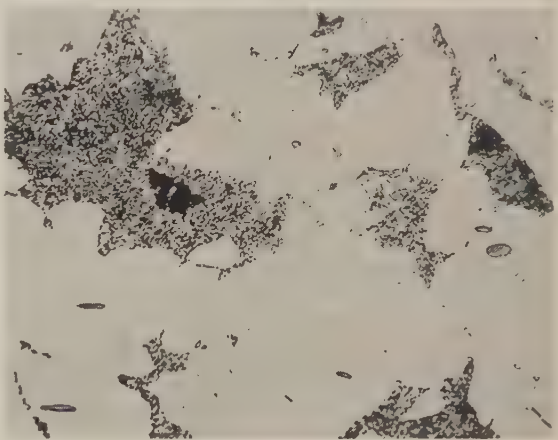
(a) Low-temperature end of weld-affected zone; ×500



(b) Weld-unaffected metal; ×500

FIG. 22 CREEP-TEST SPECIMEN CASE 6.

(a) Low-temperature end of weld-affected zone; ×500



(b) Weld-unaffected metal; ×500

FIG. 23 CREEP-TEST SPECIMEN CASE 7

steel refinery tube with 0.48 per cent nickel also was found free of graphite after 30,660 hr of service at temperatures of 1080 to 1100 F under 1860 psi.

These examples permit some interesting deductions. A study of cases 1 and 3 shows that susceptibility to graphitization is not a direct function of the amount of aluminum added in deoxidization practice and that two steels may possess similar McQuaid-Ehn case characteristics and yet display a marked difference in susceptibility to graphitization.

A comparison of cases 2 and 3 demonstrates that weld-affected metal in the stress-relieved condition is decidedly more susceptible to graphitization than normalized and stress-relieved metal.

Cases 5 to 7 show that straight silicon-killed McQuaid-Ehn coarse-grained and normal carbon-molybdenum steel is fundamentally unstable and will graphitize under extreme conditions.

The absence of graphite in the three nickel-containing refinery tubes raises the question whether the resistance to graphitization of these tubes is due to the nickel in the steel. The authors have expressed the opinion that nickel, dissolved in the ferrite, may prevent graphitization by interfering with the carbon diffusion due to distortion of the ferrite lattice and by forcing more molyb-

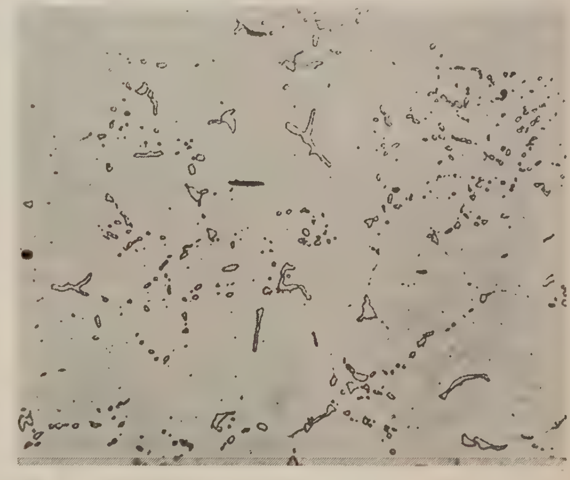


FIG. 24 MICROSTRUCTURE OF REFINERY-TUBE, CASE 9; ×500

denum into the carbide. The microstructure of the three refinery tubes, an example of which is depicted in Fig. 24 of this discussion, shows a high degree of carbide diffusion, spheroidization, and coalescence. One can readily imagine that a low rate of carbon diffusion may retard the beginning of graphite formation and the growth of the graphite cells formed, but in the case of a steel which has undergone fargoing carbide diffusion and coalescence, the rate of carbon diffusion does not seem to have been a factor in preventing graphitization. It also appears to us to be doubtful that nickel conferred stability to the ferrite against carbon rejection in the form of graphite by mechanically strengthening the ferrite, i.e., by increasing the work to be done by the volume expansion resulting from the carbide dissociation, since another element, silicon, which is a much more powerful ferrite strengthener than nickel, is known to promote carbide dissociation. However, the possibility that nickel dissolved in the ferrite forces more molybdenum into the carbide, as suggested by the authors, remains.

One may also wonder if the addition of an element which dissolves in the ferrite, such as nickel, does not modify the electrostatic balance in the ferrite lattice in a manner which is less favorable to carbon rejection in the form of graphite. It all goes to show that there is much to be learned yet before a full understanding of the phenomenon of graphitization is attained. The authors have presented some significant facts and have indicated a new approach to the solution of this difficult problem for which they deserve appreciation and thanks.

L. H. Carr.[6] In the Edward laboratories, we have been making extensive graphitization tests for the past year and a half. We have obtained a certain amount of test data which does not differ in its essential details from most of that which has been presented already. The tests show up many of the same anomalies which have already been pointed out. They seem to show that we all know a number of factors which definitely have to do with a tendency toward graphitization, but that we do not know for sure of an economical solution which can be applied to steelmakers the country over which will guarantee the absence of graphite for service times of 5 to 25 years.

For example, we have found such conflicting data as the following: Samples of cast carbon-molybdenum steel from one foundry source deoxidized with 2 lb of aluminum per ton will develop graphite in every test. Samples of the same material again deoxidized with 2 lb of aluminum per ton from another foundry source have never developed graphite in any of our tests. On the other hand, the foundry source whose material graphitizes with high aluminum addition, has made a number of test bar heats of either no aluminum or $^1/_2$ lb per ton. None of these specimens has graphitized in our tests, but other investigators have found graphite developing in steels of low aluminum additions. We have tested a good many heats of steel graphitized when stress-relieved at 1200 F after welding, and none of them showed graphite when stress-relieved at 1300 F after welding. Again, however, other investigators have reported this as not being a foolproof preventive. We have tested a number of heats of cast carbon-molybdenum steel containing from 0.40 to 0.60 per cent chromium for test periods as long as 10,000 hr at 1025 F. Again none of these specimens has shown graphite, although we have heard from other investigators that they have found graphite in some steels containing chromium.

Again, we might call attention to a number of pipe investigations wherein graphite has been found distributed in a number of random locations away from any apparent welding or heating. It would seem therefore that perhaps graphitization is a little bit like creep in that at high temperatures we will always have some, and that the engineering and metallurgical problem is to minimize this tendency toward graphitization, rather than to try to erase it 100 per cent in all cases.

In view of the fact that different steelmakers, using nominally the same practice, produce carbon-molybdenum steels of widely differing graphitization tendencies, it would seem that too much weight could not be given to the results of any one graphitization investigator. Furthermore, it is felt that too many suggested solutions to the whole problem have been discarded because of just one or two graphite spots widely distributed in a sample. Since no simple cure-all for the matter has been developed, it would seem that the best way to give engineers the information they need today would be to summarize all available test information in some sort of a statistical score sheet. This might show, for example, that of several thousand specimens of high-aluminum carbon-molybdenum steel 30 per cent would show graphitization at the end of a certain testing time. It might show in the case of several thousand other low-aluminum carbon-molybdenum steels that 3 or 4 per cent would show graphitization under similar time and temperatures. Again, it might show that of specimens stress-relieved at 1300 F, a certain per cent would show graphitization. Several hundred specimens of chromium containing carbon-molybdenum steels might show that only one or two had developed any graphite. It would also be desirable to work out some simple method of classifying the degree of graphitization. A simple statement of "Yes, it is present," or "No, it is not," leads to the condemnation of many test samples which are structurally quite sound, but which show one or two isolated graphite spots. Samples in which two out of three fields of view show no graphite in the critical area should perhaps receive a separate classification.

It is believed that a large amount of data of this sort is now available in laboratories throughout the country, which will represent the widest possible range of steel manufacture. A table of this sort should enable engineers to select materials for today's problems on a much more rational basis than merely listening to the opinions of any one or two investigators.

Authors' Closure

The authors are indebted to Mr. Eberle and Mr. Carr for their discussions and interest. The cases cited by Mr. Eberle are significant in illustrating the profound influence of minor variations of treatment on graphitizing tendency. In our own work we have found that in certain samples which were presumed to be homogeneous, since they were cut out of larger parts which had been treated as a whole, a particular region would display a marked shower of graphite particles while the remainder of the sample would be devoid of visible graphite (on the same polished section). This made for great tedium in the microscopic examination for the failure to find graphite at the first examination gave no assurance graphite was not present and in significant quantity. Particular conditions of divorcement of ferrite and carbide were suspect, as brought out in the paper and if graphite was not found in the first instance, the sample was ground down and repolished through successive layers until there was reasonable assurance that graphite either was or was not present.

This critical examination was required by the nature of the Springdale failure and by the findings on the graphitized valve body described in the paper. The graphite found on a particular section may not be significant in revealing the degradation of the whole member. This was brought out by Blumberg in the general discussion of graphitization at the December, 1943, meeting of the A.S.M.E.

The foregoing comments will partially serve in reply also to Mr. Carr. The compilation and statistical analysis of graphitiza-

[6] Director Research, Edward Valves, Inc., East Chicago, Ill. Mem. A.S.M.E.

tion data suggested by Mr. Carr would possess a certain interest but would afford small satisfaction to the design engineer. To lay out a power plant with the knowledge that 30 per cent or even 3 per cent of the steels involved would graphitize within the normal service life, perhaps seriously, would not be very comforting. It is quite true that we shall always have a little creep at these high temperatures but creep is something that can be measured and predicted and loading values are selected so that total creep over service expectancy is kept small in amount. However, the action through which controlled creep takes place does not lead to marked degradation of the physical properties. Graphitization involves degradation and may in some instances lead to serious failure. We therefore believe that erasing of graphitization 100 per cent *is* the metallurgical problem and to fail to strive for that end is not sound.

Question has been raised regarding usage of 1100 F as a graphitization test temperature. The results presented at this session of the Joint E.E.I.-A.E.I.C. Investigation supports our position in this. Although we made no detailed attempt to measure graphitization quantitatively, qualitatively it appeared from our results that time was the principal factor affected by using higher temperatures. Furthermore, 1100 F is a directly significant temperature when it is considered that in some newer installations operating swings will closely approach it.

We affirm our position that for increasing the relative stability the judicious use of alloys offers the safest approach. Also, the mechanisms are such that simple concepts, for example that thermodynamic stability of carbides is the only factor, may be not only misleading, but also dangerous. In support of our findings on WC-4 steel other investigators working within the chemistry of this specification have found samples with weld metal deposited thereon resistant to graphitization in the weld-affected zone over periods up to 10,000 hours at 1100 F. If the chart on graphitization rates prepared for the Summary Report of the Joint E.E.I.-A.E.I.C. may be presumed correct this would be the equivalent of 100,000 hours at 925 F.

Influence of Heat-Treatment Upon the Susceptibility to Graphitization of High-Aluminum-Deoxidized Carbon-Molybdenum Steel

By F. EBERLE,[1] BARBERTON, OHIO

Susceptibility to graphitization of carbon-molybdenum steels may be influenced by heat-treatment of the material prior to installation. Tests are reported in this paper on high-aluminum-deoxidized pipe with welded upset ends, the upsetting being carried out at 2000 F without subsequent heat-treatment. Graphitization in certain zones of the pipe were in evidence. Subsequently the influence of prior heat-treatment or weld heat effects upon susceptibility to graphitization was explored. The procedure and results attained are recorded in this paper.

Introduction

A STUDY of the behavior of carbon-molybdenum steels in long-time service at steam-plant operating temperatures led to the observation that susceptibility to graphitization may be influenced by the heat-treatment of the material prior to installation. For example, in high-aluminum-deoxidized pipe with welded upset ends, the upsetting being carried out at 2000 F without subsequent heat-treatment, the following conditions might be encountered:

(*a*) High-temperature end of weld-affected upset material: not graphitized.

(*b*) Low-temperature (1300–1400 F) zone of weld-affected upset material: graphitized.

(*c*) Weld-unaffected upset part of pipe: not graphitized.

(*d*) Transition zone from the upset to the unupset part of the pipe which had been heated to and cooled from a temperature of 1300–1400 F: graphitized.

(*e*) Heat-unaffected unupset body of the pipe: graphitized or not graphitized, depending on the heat-treating temperature and rate of cooling prior to installation.

This influence of prior heat-treatment or weld heat effects upon susceptibility to graphitization was subsequently explored with a series of experiments designed to examine the following factors:

1 Rapid cooling from very high temperatures as sometimes encountered in upset pipe ends and in the high-temperature zone of weld-affected metal.

2 Slow cooling from very high temperatures, sometimes found in pipes and heavy castings.

3 Rapid cooling from temperatures above but in the vicinity of the A_3 point of the steel as employed in standard normalizing treatment.

4 Slow-cooling from temperatures above but in the vicinity of the A_3 point of the steel as met in "normalized" heavy castings.

5 Rapid and slow cooling from temperatures just above the A_1 point of the steel after heat-treatments (1) to (4) as may occur in the low-temperature zone of weld-affected metal ("contact zone").

6 Quenching of normalized material from temperatures just above and just below the A_1 point of the steel as may be encountered in the contact zone between weld-affected and weld-unaffected metal.

7 Annealing at 1200 F and 1300 F, respectively, as employed for stress-relieving.

No attempt was made in these experiments to reproduce exact fabrication conditions, i.e., the rate of cooling in the normalized samples was sometimes faster, the quenching of the quenched samples more drastic, and the cooling of the annealed samples slower than in fabricated material. This was done purposely in order to establish fundamental trends and, if possible, to shed more light on the mechanism of graphitization.

Experimental Procedures

For some of the tests only unwelded specimens were used, consisting of $^3/_4$-in. cubes machined from the heat-treated material so as to eliminate oxidized and decarburized surface metal. In general, the heat-treating was done in a salt or lead bath. Some tests with unwelded material were run in duplicate, one set of specimens remaining in the heat-treated and machined condition, while identical specimens of a second set were 50 per cent cold-compressed in a tensile machine prior to the soaking in the lead bath at 1000 F. This severe cold deformation had been found generally to promote or accelerate graphitization and helped to induce the formation of graphite in some conditions of heat-treatment where it would otherwise not occur, at least not within the chosen duration of the tests.

A second set of experiments was conducted with special spot welds consisting of 1-in-diam × $^1/_8$-in-thick disks which were prepared by placing them between the water-cooled electrode tips of a spot-welding machine and applying sufficient current to fuse the core of the disks from cold to fusion within $^1/_{15}$ sec, $^2/_{15}$ sec, and $^1/_5$ sec, respectively, the disks then cooling to room temperature within 1 sec. These tests were intended to give information on the effect of extremely rapid cooling under conditions of welding.

A third series of tests was carried out with regular multiple-pass arc welds prepared by joining two plates which had been water-, air-, or slow furnace-cooled from 2200 F and 1650 F, respectively, prior to the welding, water-cooled plate being welded to water-cooled plate, and air-cooled plate to air-cooled plate, etc. Each

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[1] Research Metallurgical Engineer, The Babcock & Wilcox Company.

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of these test welds was then cut into three sections, one section remaining in the as-welded condition, a second section being standard stress-relieved at 1200 F and the third section annealed at 1300 F for 20 hr. These tests were intended to furnish information on the effect of various heat-treatments under actual fabricating conditions.

All graphitization tests were conducted in a charcoal-covered lead bath held at 1000 F.

Materials

Two types of carbon-molybdenum steel were employed: Two high-aluminum deoxidized steels of great susceptibility to graphitization (steels A and B) and a high-aluminum-deoxidized steel of low susceptibility to graphitization (steel C). The deoxidation practice, McQuaid-Ehn case characteristics, and chemistry of these steels are given in Tables 1 to 6 which also contain summaries of the test results.

TABLE 1 TEST SERIES A—TESTS WITH UNWELDED SPECIMENS IN VARIOUS CONDITIONS OF HEAT-TREATMENT

(Specimens 3/4-in. cubes machined from heat-treated material)

Steel "A": Si-Al Killed (1.8 lb. Al/Ton), McQuaid-Ehn Grain Size 7 (Abnorm.)
.19C, .48Mn, .16Si, .018P, .024S, .52Mo, .039 Al, .009 Al_2O_3

Specimen	Heat Treatment	Results of 3000 hr Tests At 1000F	Illustration
B1	1 hr 2200F/Air	No Graphite	
B2	1 hr 2200F/Air +50% Cold Reduction	No Graphite	
B3	1 hr 2200F/Air +1 hr 1400F/Fce (1°/hr)	Very Few Scattered Very Small Nodules	
B4	1 hr 2200F/Air +1 hr 1400F/Fce (1°/hr) +50% Cold Reduction	Scattered Small Nodules	Fig. 1
E1	1 hr 1650F/Air	No Graphite	
E2	1 hr 1650F/Air +50% Cold Reduction	Scattered Small Nodules	Fig. 2
E3	1 hr 1650F/Air +1 hr 1400F/Fce (1°/hr)	No Graphite	
E4	1 hr 1650F/Air +1 hr 1400F/Fce (1°/hr) +50% Cold Reduction	Few Scattered Small Nodules	
E5	1 hr 1650F/Air +1 hr 1400F/Air	Very Few Scattered Very Small Nodules	
E6	1 hr 1650F/Air +1 hr 1400F/Air +50% Cold Reduction	Few Scattered Small Nodules	
E7	1 hr 1650F/Air +1 hr 1400F/Water	Few Scattered Very Small Nodules	
E8	1 hr 1650F/Air +1 hr 1400F/Water +50% Cold Reduction	Scattered Very Small Nodules	
C1	1 hr 2200F/Fce (50°/hr)	Scattered Large Nodules	Fig. 3
C2	1 hr 2200F/Fce (50°/hr) +50% Cold Reduction	Increased Number Of Scattered Large Nodules	
C3	1 hr 2200F/Fce (50°/hr) +1 hr 1400F/Fce (1°/hr)	Very Few Scattered Small Nodules	
C4	1 hr 2200F/Fce (50°/hr) +1 hr 1400F/Fce (1°/hr) +50% Cold Reduction	Grain Boundary Chain Graphite And Scattered Medium Sized Nodules	Fig. 4
F1	1 hr 1650F/Fce (50°/hr)	Segregated Graphite At Surface Only	
F2	1 hr 1650F/Fce (50°/hr) +50% Cold Reduction	Scattered Small Nodules	
F3	1 hr 1650F/Fce (50°/hr) +1 hr 1400F/Fce (1°/hr)	Few Scattered Small Nodules	
F4	1 hr 1650F/Fce (50°/hr) +1 hr 1400F/Fce (1°/hr) +50% Cold Reduction	Scattered Small Nodules	Fig. 5
F5	1 hr 1650F/Fce (50°/hr) +1 hr 1400F/Air	Very Few Scattered Small Nodules	
F6	1 hr 1650F/Fce (50°/hr) +1 hr 1400F/Air +50% Cold Reduction	Scattered Small Nodules	
F7	1 hr 1650F/Fce (50°/hr) +1 hr 1400F/Water	Few Scattered Small Nodules	
F8	1 hr 1650F/Fce (50°/hr) +1 hr 1400F/Water +25% Cold Reduction	Scattered Small Nodules	Fig. 6

TABLE 2 TEST SERIES B—TESTS WITH UNWELDED SPECIMENS IN VARIOUS CONDITIONS OF HEAT-TREATMENT

(Specimens 3/4-in. cubes machined from heat-treated material)

Steel "B": Si-Al Killed (1.8 lb. Al/Ton), McQuaid-Ehn Grain Size 6-7 (Abnorm.)
.11C, .45Mn, .14Si, .012P, .026S, .53Mo, .036 Al, .006 Al_2O_3

Specimen	Heat Treatment	Results Of 3000 hr Tests At 1000F
SR1	4 hr 2200F/Fce (50°/hr)	No Graphite
SR2	4 hr 2200F/Fce +4 hr 1200F/Fce	Localized Nodules At Surface Only
SR3	4 hr 2200F/Fce +4 hr 1300F/Fce	Some Scattered Medium Nodules At Surface Only
SR4	4 hr 2200F/Fce +50% Cold Reduction	Numerous Small To Medium Nodules
SR5	4 hr 2200F/Fce +50% Cold Red. +4 hr 1200F/Fce	Numerous Small To Medium and Some Large Nodules
SR6	4 hr 2200F/Fce +50% Cold Red. +4 hr 1300F/Fce	Medium Amount Of Small To Medium And Some Large Nodules
SR7	4 hr 2200F/Water	No Graphite
SR8	4 hr 2200F/Water +4 hr 1200F/Fce	No Graphite
SR9	4 hr 2200F/Water +4 hr 1300F/Fce	No Graphite
SR10	4 hr 2200F/Water +50% Cold Reduction	No Graphite
SR11	4 hr 2200F/Water +50% Cold Red. +4 hr 1200F	No Graphite
SR12	4 hr 2200F/Water +50% Cold Red. +4 hr 1300F	No Graphite

TABLE 3 TEST SERIES C—TESTS WITH UNWELDED SPECIMENS IN VARIOUS CONDITIONS OF HEAT-TREATMENT

(Specimens $^3/_4$-in. cubes machined from heat-treated material)

Steel "A": Si-Al Killed (1.8 lb. Al/Ton), McQuaid-Ehn Grain Size 7 (Abnorm.)
.19C, .48Mn, 16Si, .018P, .024S, .52Mo, .039 Al, .009 Al_2O_3

Specimen	Heat Treatment	Results Of 1500 hr Tests At 1000F
G1	1 hr 2200F/Air + 1 hr 1300F/Fce	No Graphite
G2	1 hr 2200F/Air + 1 hr 1300F/Fce +50% Cold Reduction	Small Nodules At Surface Only
G3	1 hr 1650F/Air + 1 hr 1300F/Fce	No Graphite
G4	1 hr 1650F/Air + 1 hr 1300F/Fce +50% Cold Reduction	Few Small Nodules At Surface Only
G5	1 hr 1650F/Air +20 hr 1300F/Fce	No Graphite
G6	1 hr 1650F/Air +20 hr 1300F/Fce +50% Cold Reduction	Few Small Nodules At Surface Only
H1	1 hr 2200F/Fce (50°/hr) + 1 hr 1300F/Fce	Some Small To Medium Nodules At Surface Only
H2	1 hr 2200F/Fce (50°/hr) + 1 hr 1300F/Fce +50% Cold Red.	Numerous Small Nodules
H3	1 hr 1650F/Fce (50°/hr) + 1 hr 1300F/Fce	Some Small Nodules At Surface Only
H4	1 hr 1650F/Fce (50°/hr) + 1 hr 1300F/Fce +50% Cold Red.	Few Small Nodules
H5	1 hr 1650F/Fce (50°/hr) +20 hr 1300F/Fce	Some Small Nodules At Surface Only
H6	1 hr 1650F/Fce (50°/hr) +20 hr 1300F/Fce +50% Cold Red.	Some Small Nodules At Surface Only

TABLE 4 TEST SERIES D—TESTS WITH UNWELDED SPECIMENS QUENCHED FROM TEMPERATURES JUST ABOVE AND JUST BELOW THE A_1 POINT OF THE STEEL

($^1/_2 \times {}^1/_4 \times {}^1/_4$ in. specimens, normalized at 1650 F)

Steel "B": Si-Al Killed (1.8 lb. Al/Ton), McQuaid-Ehn Grain Size 6-7 (Abnorm.)
.11C, .45Mn, .14Si, .012P, .026S, .53Mo, .036 Al, .006 Al_2O_3

Specimen	Heat Treatment	Results of 10000 hr Tests At 1000F
K1	½ hr 1400F/Ice-Brine	Uniformly Scattered Graphite
K2	½ hr 1350F/Ice-Brine	Less Uniform And Less Graphite Than In K1
K3	½ hr 1300F/Ice-Brine	As In K2
K4	½ hr 1250F/Ice-Brine	As In K2

TABLE 5 TEST SERIES E—TESTS WITH DISKS CONTAINING SPOT-WELD-FUSED CORES

(Specimens 1 in. diam × $^1/_8$ in. thick, machined from material normalized at 1700 F and heated between the water-cooled electrode tips of a spot-welding machine)

Steel "B": Si-Al Killed (1.8 lb. Al/Ton), McQuaid-Ehn Grain Size 6-7 (Abnormal)
.11C, .45Mn, .14Si, .012P, .026S, .036 Al, .006 Al_2O_3

Specimen	Heating Time To Fusion	Cooling Time To Room Temp.	Results of 10000 hr Tests At 1000F: Fused Core	Contact Z. Weld-Aff./Weld-Unaff. M.	Weld-Unaff. Metal
L1	1/15 sec.	1 sec.	No Graphite	Few Scattered Nodules	Few Scattered Nodules
L2	2/15 sec.	1 sec.	No Graphite	Few Scattered Nodules	Few Scattered Nodules
L3	1/5 sec.	1 sec.	No Graphite	Few Scattered Nodules	Few Scattered Nodules

TABLE 6 TEST SERIES F—TESTS WITH STANDARD MULTIPLE-PASS ARC WELDS

(Specimens 5 × 5 × 1-in. plate sections in various conditions of heat-treatment, welded together and post-weld-treated as indicated)

Steel "C": Si-Al Killed (1.5 lb. Al/Ton), McQuaid-Ehn Grain Size 5-7 (Slight To Medium Degree of Abnormality)
.18C, .83Mn, .23Si, .016P, .023S, .49Mo, .041 Al, .029 Al_2O_3

			Results Of 6000 hr Tests At 1000F			
			2200F Treated Material		1650F Treated Material	
Specimen	Heat Treatment Of Plate Pair	Postweld Treatment	Weld-Unaff. M.	Heat-Aff. Z. (Contact Z.)	Weld-Unaff. M.	Heat-Aff Z. (Contact Z.)
M1	2200F/Water—1650F/Water	None	No Graphite	No Graphite	No Graphite	No Graphite
M2	2200F/Water—1650F/Water	1 hr 1200F	No Graphite	No Graphite	No Graphite	No Graphite
M3	2200F/Water—1650F/Water	20 hr 1300F	No Graphite	No Graphite	No Graphite	No Graphite
M4	2200F/Air—1650F/Air	None	No Graphite	No Graphite	Some Nodules In Some Cold Worked Surface Zones	No Graphite
M5	2200F/Air—1650F/Air	1 hr 1200F	No Graphite	No Graphite	Some Nodules In Some Cold Worked Surface Zones	No Graphite
M6	2200F/Air—1650F/Air	20 hr 1300F	No Graphite	No Graphite	No Graphite	No Graphite
M7	2200F/Fce*—1650F/Fce*	None	A Few Isolated Nodules	Some Widely Scattered Medium Nod.	Some Nodules In Cold Worked Surface Zones Only	Some Scatt. Small To Med. Nodules
M8	2200F/Fce*—1650F/Fce*	1 hr 1200F	A few Widely Scatt. Nodules	Some Scatt. Medium Nod.	Some Large Nod In Surface Zones Only	No Graphite
M9	2200F/Fce*—1650F/Fce*	20 hr 1300F	No Graphite	No Graphite	No Graphite	No Graphite

* Cooled 50° per hour

Test Results

Evaluation of the test results led to the following conclusions:

1 Carbon-molybdenum steels deoxidized with the same nominal amount of aluminum, possessing practically identical chemistry and displaying practically identical abnormal McQuaid-Ehn case characteristics may show differences in susceptibility to or rate of graphitization.

2 Annealing at very high temperatures followed by rapid cooling seems to inhibit or retard graphitization at the contemplated temperatures. This was found to be true also for specimens which had been 50 per cent cold-reduced by compression prior to the subcritical annealing at 1000 F.

3 Annealing at temperatures above, but near the A_3 point of the steel followed by rapid cooling appears to be much less effective in retarding graphitization than rapid cooling from very high temperatures. When severely cold-deformed, such material may form graphite within 1500 to 3000 hours at 1000 F.

4 Annealing at very high temperatures followed by slow cooling seems to promote or accelerate graphitization. In this condition relatively large scattered graphite nodules are formed within 1500 hours at 1000 F.

5 Annealing at temperatures above, but near the A_3 point followed by slow cooling renders the steel less susceptible to graphitization than slow cooling from very high temperatures, but more susceptible than air cooling from temperatures above but near the A_3 point.

6 Reheating steel which had been rapidly cooled from very high temperatures and thereby made insensitive to graphitization, to temperatures just above the A_1 point renders it again susceptible, irrespective of the subsequent rate of cooling.

7 Heating to just above the A_1 point (1400 F) followed by water-quenching renders the steel more susceptible to graphitization than slow furnace-cooling from the same temperature. On the other hand, slow-cooling from just above the A_1 point favors graphitization more than air-cooling. In this connection it should be remembered that quenching from very high temperatures inhibited the formation of graphite more effectively than any other heat-treatment. The high susceptibility to graphitization resulting from quenching from just above the A_1 point therefore deserves particular attention.

8 Steel which previously had been heated to very high temperatures appears to be rendered slightly more susceptible to graphitization when subsequently reheated to just above the A_1 point than steel which previously had been heated to temperatures above but in the vicinity of the A_3 point.

9 Material which had been slow furnace-cooled from temperatures above the A_3 point appears to be slightly more susceptible to graphitization when subsequently reheated to temperatures just above the A_1 point than previously fast-cooled material. Steel slowly cooled from very high temperatures, reheated to above the A_1 point and then severely strained, shows a tendency to form continuous grain-boundary chain graphite.

10 Quenching from temperatures just above the A_1 point seems to favor graphitization more than quenching from temperatures just below the A_1 point.

11 Annealing at 1200 F as used in conventional stress-relieving appears to have little effect upon the formation of graphite unless accompanied by a reduction or elimination of stresses. Under like conditions, annealing at 1300 F for the same length of time seems to reduce the number of graphite nodules, but does not seem to prevent their formation.

12 Long-time annealing at 1300 F considerably retards the formation of graphite and also tends to reduce the number of nodules formed.

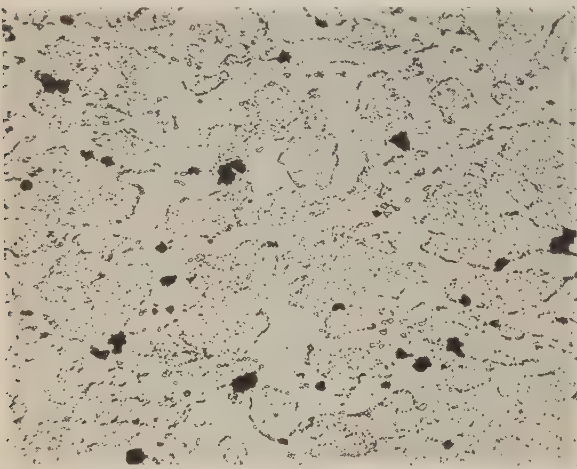


Fig. 1 Specimen B4 After 3000 Hr at 1000 F; ×500
(Steel A, 1 hr 2200 F/Air plus 1 hr 1400 F/Fce (1 deg per hr) plus 50 per cent cold reduction.)

Fig. 2 Specimen E2 After 3000 Hr at 1000 F; ×500
(Steel A, 1 hr 1650 F/Air plus 50 per cent cold reduction.)

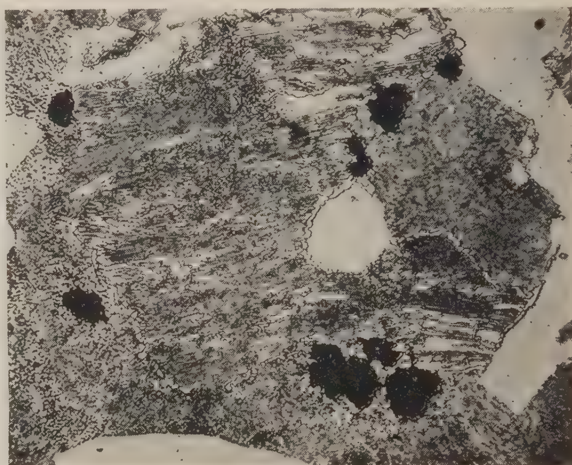
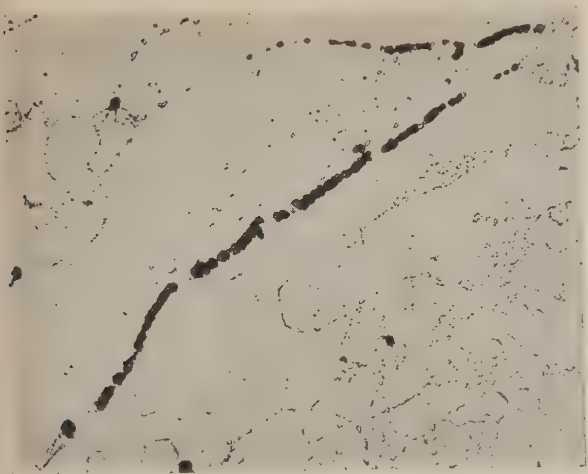


Fig. 3 Specimen C1 After 3000 Hr at 1000 F; ×500
(Steel A, 1 hr 2200 F/Fce (50 deg per hr).)



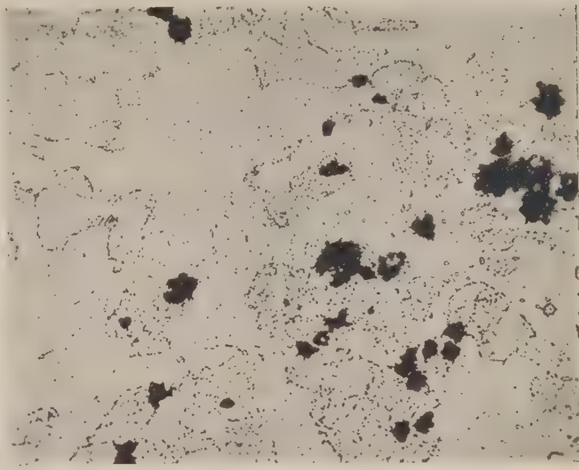


FIG. 4 SPECIMEN C4 AFTER 3000 HR AT 1000 F; ×500
(Steel A, 1 hr 2200 F/Fce (50 deg per hr) plus 1 hr 1400 F /Fce (1 deg per hr) plus 50 per cent cold reduction.)

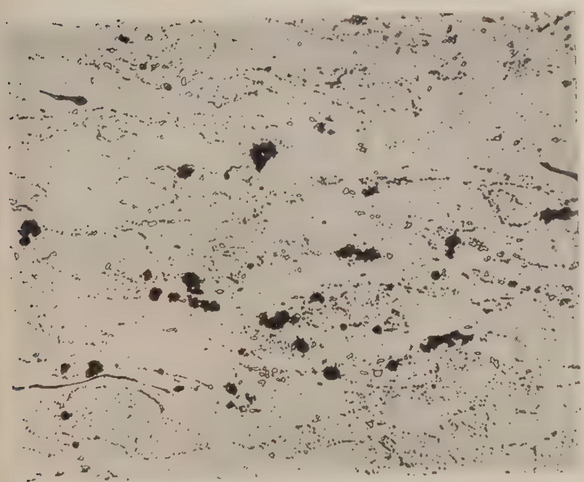


FIG. 5 SPECIMEN F4 AFTER 3000 HR AT 1000 F; ×500
(Steel A, 1 hr 1650 F/Fce (50 deg per hr) plus 1 hr 1400 F/Fce (1 deg per hr) plus 50 per cent cold reduction.)

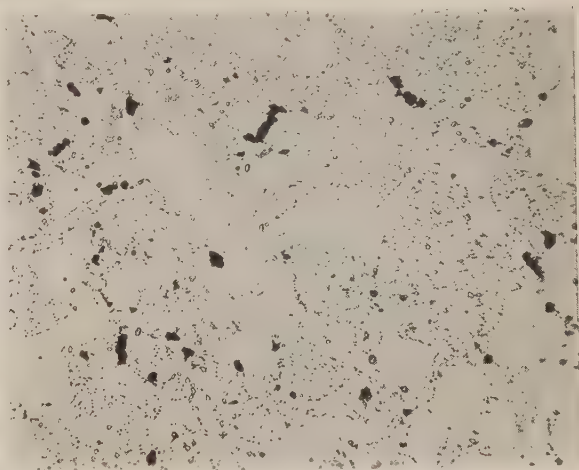


FIG. 6 SPECIMEN F8 AFTER 3000 HR AT 1000 F; ×500
(Steel A, 1 hr 1650 F/Fce (50 deg per hr) plus 1 hr 1400 F/water plus 25 per cent cold reduction.)

General Deductions

The pattern of susceptibility to graphitization obtained with the various heat-treatments preceding the isothermal graphitization tests at 1000 F clearly indicates the involvement of a submicroscopic precipitation phenomenon as suggested by Austin and Fetzer.[2] There seems to be a substance, elements or compounds, which upon precipitation in a certain state promote carbide instability. The solution temperature of this substance seems to be rather high and its precipitation from solution relatively sluggish. Air-cooling from temperatures above 2000 F appears to suffice to retain these elements or compounds in solution even in the extremely sensitive high-aluminum-deoxidized steels and in this condition the steel will not graphitize within 10,000 hr at 1000 F, as demonstrated by the behavior of the spot-weld-fused samples. Heating the steel to intermediate temperatures above but in the vicinity of the A_3 point apparently does not create complete solution, and diffusion of the submicroscopic phase and abnormal steel air-cooled from such temperatures will therefore graphitize under favorable conditions, as for instance, when severely cold deformed. Slow cooling from very high and from intermediate temperatures precipitates the phase and renders the steel susceptible to graphitization. Complete solution and diffusion of the phase as obtained at very high temperatures seems to result in a more effective state of precipitation than incomplete solution and diffusion as obtained at intermediate temperatures.

If steel which had been rapidly cooled from high temperatures is subsequently reheated to temperatures just above the A_1 point, the dissolved phase will at least partly precipitate and render the steel again susceptible to graphitization. It is suspected that the allotropic transformation at the A_1 point, particularly when combined with straining, aids the precipitation of the submicroscopic phase. The preferential occurrence of graphite in the low-temperature zone of weld-affected metal may be similarly explained. Short-time annealing at 1200 F and 1300 F apparently

[2] "Factors Controlling Graphitization of Carbon Steels at Subcritical Temperatures," by C. R. Austin and M. C. Fetzer. Trans. A.S.M., vol. 39, 1945, pp. 485–535.

fails to bring about effective precipitation of the phase. On the other hand, once the phase is precipitated, annealing at 1200 F and 1300 F tends to render the phase ineffective, presumably by coalescence or agglomeration. At 1200 F, this effect seems to be very slight and even at 1300 F a relatively long annealing time is required to produce results of practical significance.

Practical Conclusions

Applying the foregoing deductions to the practical aspect of graphitization in welded structures, the following conclusions suggest themselves:

1 Normalized material is slightly less susceptible to graphitization than annealed material.

2 Normalizing at temperatures close to the A_3 point offers a better safeguard against the formation of dangerous chain graphite than normalizing at very high temperatures.

3 Welding conditions should be chosen so as to minimize strain in the weld-affected metal by suitable preheating.

4 Long-time stress relieving at 1300 F is more advantageous as a safeguard against graphitization than conventional stress-relieving at 1200 F.

Acknowledgment

The author gratefully acknowledges the inspiration and help received from the late Mr. H. J. Kerr, executive assistant of The Babcock and Wilcox Company, in carrying out this work.

General Discussion of Graphitization Papers

S. H. WEAVER.[1] The General Electric graphite furnace heats bars with a lengthwise weld bead to 900 F at one end and 1300 F at the other end. All bars were notched at the 1100 F location for examination after 3000 hr. The 6200-hr study has been partially completed.

Fig. 1 of this note, for items 1 to 7, Table 1, gives the graphitic growth by chemical analysis in 3000 hr for different temperatures upon 0.15 per cent carbon steel with different deoxidization. The 1200 to 1300 F points were omitted because of decarburization of the steel. The furnace was then changed to dry nitrogen atmosphere.

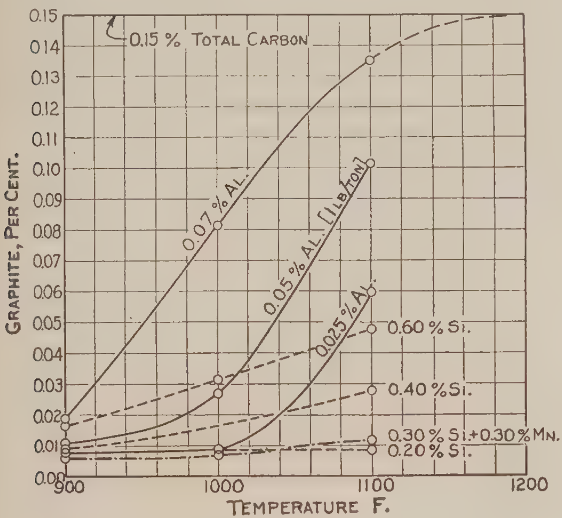


FIG. 1 GROWTH OF GRAPHITE AT 3000 HR IN 0.15 PER CENT CARBON STEEL

In the alloy series in Table 2 of this note, items 8 to 16 have no microscopic graphite after 6200 hr at 1100 F.

Table 3 lists nine conditions of pipe to A158-36P1 with 2 lb per ton of aluminum where at 1100 F in 3000 and 6200 hr all contained small nodular graphite except the 2300 F upset-end sample.

The four items of pipe in Table 4 to A206-42T with 1 lb per ton of aluminum, and the four items in Table 5, with 0.4 lb per ton of aluminum have shown no graphite at 1100 F. No examination has been made at other temperatures.

There are 19 other bars in the furnace including samples of 1 per cent Mo; 1 per cent Mo, 0.2 per cent V; 0.5 per cent Cr, 0.5 per cent Mo, in each case with varying practices as regards heat-treatment and deoxidization practice. No graphite has been observed after the first 1700 hr. A longer soak will be required before any conclusion can be drawn from these latter test bars.

[1] Turbine-Generator Engineering Division (retired), General Electric Company, Schenectady, N. Y.

TABLE 1 INDUCTION FURNACE HEATS, DIFFERENT DEOXIDIZATION

Heat Treatment: 1700 F—2 hr air cool
1200 F—2 hr furnace cool

Item	C	Al	Si	Mn
1	0.15	0.025	..	..
2	0.15	0.050	..	..
3	0.15	0.075	..	..
4	0.15	..	0.20	..
5	0.15	..	0.40	..
6	0.15	..	0.60	..
7	0.15	..	0.30	0.30

NOTE: Graphite nodules in 3000 hr. See Fig. 8 for graphite chemical analysis with temperature.

TABLE 2 INDUCTION FURNACE HEATS, EFFECT OF ALLOYS

Heat Treatment: 1700 F—2 hr air cool
1200 F—2 hr furnace cool

Item	C	Si	Mn	Mo	Cr
8	0.15	0.30	0.70	..	..
9	0.15	0.30	1.30	..	..
10	0.15	0.30	0.30	0.50	..
11	0.15	0.30	0.70	0.50	..
12	0.15	0.30	0.90	0.50	..
13	0.15	0.30	..	0.50	0.30
14	0.15	0.30	..	0.50	0.70
15	0.15	0.30	..	0.50	0.90
16	0.15	0.30	..	0.50	1.30

NOTE: No graphite in 6200 hr at 1100 F but fully spheroidized.

TABLE 3 PIPE SPECIFICATION A.S.T.M., A158-36P1

(2 lb al per ton)

1 "As received;" 1200 F- 2 hr, furnace cool
2 "As received;" 1300 F- 2 hr, furnace cool
3 Upset, 2300 F- 1 hr, furnace cool
4 1600 F- 2 hr, water quench, 1200 F- 1 hr, furnace cool
5 1700 F- 2 hr, air cool, 1200 F- 2 hr, furnace cool
6 1700 F- 2 hr, air cool, 1100 F- 2 hr, furnace cool
7 1700 F- 2 hr, air cool, 1000 F- 2 hr, furnace cool
8 1700 F- 2 hr, air cool, no draw
9 575 F weld cycle, 1200 F- 2 hr, furnace cool

NOTE: Except upset item all had small nodules of graphite at 3000 and 6200 hr at 1100 F.

TABLE 4 PIPE SPECIFICATION A.S.T.M., A206-42T

(1 lb al per ton)

1 "As received;" 1200 F- 2 hr, furnace cool
2 "As received;" 1300 F- 6 hr, furnace cool
3 Upset, 2300 F- 1 hr, air cool
4 575 F weld cycle, 1200 F draw

NOTE: No graphite at 1100 F in 3000 or 6200 hr.

TABLE 5 PIPE SPECIFICATION A.S.T.M., A206-42T

(0.4 lb al per ton)

1 "As received;" 1200 F- 2 hr, furnace cool
2 "As received;" 1300 F- 6 hr, furnace cool
3 Upset, 2300 F- 1 hr, air cool
4 575 F weld cycle, 1200 F draw

NOTE: No graphite at 1100 F in 3000 or 6200 hr.

Some Stress-Corrosion Studies on Austenitic Cast Irons

By J. B. Urban,[1] J. W. Bolton,[1] and A. J. Smith[1]

Experimentation is the most adequate approach to stress-corrosion problems and should involve various mechanical loadings of a material under corrosive conditions. When corrosive conditions are held constant in respect to concentration and temperature, useful threshold stresses if existent can be ascertained and suitable safety factors adopted. The procedure for studying austenitic cast irons on this basis is discussed, the investigation indicating that at very high stresses, such irons are subject to stress corrosion in strong hot caustic. However, threshold stresses have been found to be relatively high, markedly exceeding stresses contemplated in design and application. Thus the desirable properties of such material can be utilized to good advantage, with margin to assure adequate safety in this corrosive medium.

Problems of stress corrosion have concerned engineers for many years. "Season-cracking" in brasses and embrittlement of ductile boiler-plate steel in presence of alkaline media are familiar examples. A material may possess apparently adequate strength and adequate corrosion resistance, as ascertained by separate conventional tests. Yet some materials are known to fail prematurely when exposed to certain combinations of these factors.

It is known that triaxiality of stresses may bring about acceleration of failure, or failure at lower loading than would be expected from conventional formulas for mono- and biaxial loading. Somewhat similarly, "corrosive stress" may be considered as an additional stress imposed upon a normally loaded system, giving the effect of higher stress order than contemplated by formulas. The safest approach to stress-corrosion problems is experimentation on various mechanical loadings under corrosive conditions. When corrosive conditions are held constant in respect to concentration and temperature, useful threshold stresses if existent can be ascertained and suitable safety factors adopted.

Since stress corrosion is initiated as a surface effect, it is essential that surface stress intensities (as imposed by loading) be determined accurately, so that the result of imposition of a corrosive condition can be evaluated. The resistance-wire (SR-4) type strain gage has proved quite useful in this respect.

Strong hot caustic (sodium hydroxide) is a powerful corrosive toward many metals, and produces embrittlement in some. Suitable grades of gray cast iron have fairly good resistance toward the corrosive effect and also are relatively free from stress-corrosion deterioration. Some of the high-nickel austenitic cast irons have excellent corrosion resistance toward strong caustics, even at elevated temperatures. This suggests their consideration[2] where contamination of caustic must be held to a minimum and where maximum life under such corrosive conditions is desired.

The present investigation indicates that at very high stresses the austenitic cast irons are subject to stress corrosion in strong hot caustic. However, threshold stresses have been found to be relatively high, markedly exceeding stresses contemplated in design and application. Thus the desirable properties of such material can be utilized to good advantage, with margin to assure adequate safety in this corrosive medium.

Experimental Procedures

Compositions of the materials examined are given in Table 1. Samples of austenitic-type cast iron were immersed in 50 per cent caustic solutions and boiled, using reflux condenser, for periods up to 6 months. These were broken in impact test. The test results were equal to those of companion samples not exposed to caustic. This showed that caustic embrittlement does not result when stress is absent.

TABLE 1 COMPOSITION OF MATERIALS

	Austenitic cast iron A, per cent	Austenitic cast iron B, per cent	Austenitic cast iron C, per cent	Austenitic cast iron D, per cent	No. 25 cast iron, per cent
Carbon	2.44	2.10	2.66	2.75	3.25
Graphite	1.98	2.01	...	...	...
Silicon	1.84	1.77	1.88	1.97	2.75
Manganese	0.90	0.74	0.92	0.98	0.58
Phosphorus	0.050	0.049	0.14	0.13	0.33
Sulphur	0.020	0.015	0.028	0.025	0.055
Nickel	19.82	20.61	19.86	19.09	...
Chromium	1.60	None	1.65	1.75	...
Molybdenum	0.74	0.90	0.63	0.62	...
Copper	4.10	0.06	4.55	4.33	...

Other samples were loaded under standard creep-test conditions.[3] These tests showed that the materials involved withstand stresses to 15,000 psi at 350 F, without measurable flow or "creep." After removal from the creep test and breaking, no differences in strength and ductility from companion unstressed samples were observed. These and other tests indicate that even at relatively high stresses the material is not adversely affected by mechanical loadings, even at temperatures appreciably above 350 F. The question then is whether the combined actions of stress and exposure to the corrosive medium produce effects not indicated by these factors operating independently.

The corrosive medium and condition chosen was 50 per cent by weight commercial sodium hydroxide (aqueous solution) at 280 F.

Three methods of stress application utilized were as follows:

[1] Metallurgical Department, The Lunkenheimer Company, Cincinnati, Ohio.

Contributed by the Metals Engineering Division and presented at the Fall Meeting, Cincinnati, Ohio, October 2–3, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Austenitic cast irons also are resistant to many other corrosive agents, both alkaline and acid, wherein long experience has shown that stress corrosion is not a factor.

[3] "Recommended Practice for Conducting Long-Time High-Temperature Tension Tests of Metallic Materials," A.S.T.M. Standards, 1941, E22-41.

(*a*) tensile loading, (*b*) cantilever loading, and (*c*) beam loading.

All specimens were cast in green sand and tested in the unmachined condition. (This simulates the surface conditions of cast product as commonly used.)

Tensile Loading Method

The arrangement of equipment is shown in Fig. 1. The test specimen was mounted between high-nickel-alloy pulling bars supported at the top with a ball-and-socket joint and a dead load applied to the bottom. The bottom bar passed through the base of a high-nickel-alloy container with a watertight seal, the container being filled with solution of commercial sodium hydroxide. A heating element around the container maintained the desired temperature of 280 F, through a potentiometer controller and thermocouple. (This temperature is very near the boiling point.) Water lost through evaporation was replaced continuously through a simple gravity-feed mechanism holding the solution concentration relatively constant. The average stress intensity over the affected area was established by the sum of the weights of container and contents, lower pulling bar, and applied weights.

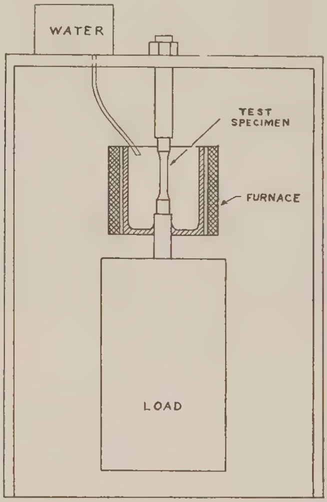


Fig. 1 Stress-Corrosion Test Setup for Tensile Loading

Simple- and Cantilever-Beam Loading Methods

These systems are shown in Figs. 2 and 3. The ends of the test specimens were attached to the load through a wire passing over a ball-bearing-mounted pulley of low frictional resistance. By means of this arrangement stresses on the specimens were independant of flexural changes in the supports, within reasonable limits of flexure of specimens.

Tests were carried out both on round specimens 0.5 in. diam, and specimens of rectangular cross section measuring 1 × 0.4 in. Specimens were $8^1/_2$ in. in length in the cantilever test; in the simple-beam tests the outer knife-edges were $9^1/_2$ in. apart, the center knife-edge $3^1/_2$ in. from the lower fixed edge.

In conventional engineering practice stress intensities (as rupture moduli) usually are calculated by the beam formulas. These formulas are based upon the assumptions of elasticity and equivalent stress-strain moduli in both tension and compression. Cast irons do not comply with these assumptions. As shown by the authors' company[4] and others, the stress-strain curve of cast irons combines both elastic and plastic components. Also, properties are not equivalent in tension and compression. Thus the neutral stress axis does not coincide with the geometric axis.

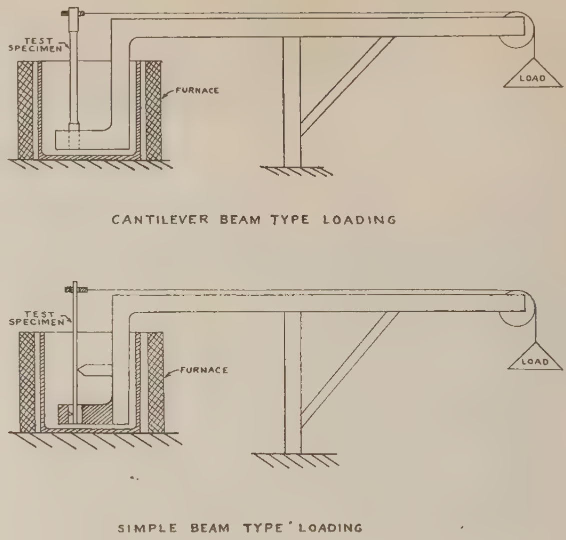


Fig. 2 Stress-Corrosion Test Setup for Beam-Type Loading

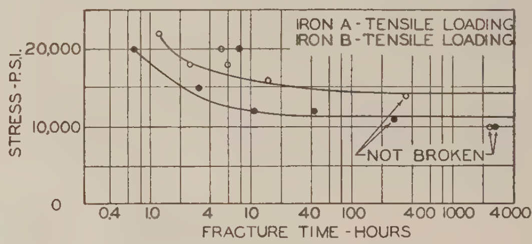


Fig. 3 Stress-Corrosion Rate for Austenitic Cast Irons; Tensile Loading

TABLE 2 SUMMARY OF TENSILE TESTS

Tensile stress, psi	Time for fracture, hr: Austenitic cast iron A	Austenitic cast iron B	No. 25 cast iron
25000	...	...	284[a]
22000	...	1.25	...
20000	0.7 and 7.7	5.2	...
18000	...	2.5 and 6	...
16000	...	15	...
15000	3.1	350[a]	...
12000	11 and 43	...	...
11000	270[a]	...	...
10000	2600[a]	2600[a]	2600[a]

[a] Not broken.

Since stress-corrosion phenomena are (at least at their incidence) influenced by surface conditions, accurate measurement of fiber stress is necessary for evaluation. Strain-resistance (SR-4) gages are quite useful for this purpose. SR-4 gages and instrumentation have been described in detail in recent literature. Many applications have been to elastic materials. In such applications the measured strain is interpreted in terms of stress by introduction of Young's modulus.

Since the materials studied are but quasi-elastic, a modified procedure was required. Relationship of strain to stress not being constant, strain readings must be compared to a separately determined stress-strain curve to ascertain the corresponding stress intensities. The details of procedure and the various calibrations are described in the Appendix. The SR-4 strain-measurements'

[4] "Some Notes on Mechanism of Deformations in Grey Iron," by J. W. Bolton, Proceedingsof the A.S.T.M., vol. 32, part 2, 1932, p. 477.

accuracy compared very favorably with measurements by other extensometric devices and results were readily reproducible. The facility with which these gages can be applied to locations and shapes where strains are not measurable by other devices extends the field of testing engineering. SR-4 gages can be used not only on simple test shapes but also on products and other complicated shapes subjected to stress and strain.

Test Results

Tensile Tests. These are summarized in Table 2 and are shown graphically in Fig. 3. The gray cast iron sustained a tensile load of 25,000 psi for 284 hr and 10,000 psi for 2600 hr without breaking. The solution discolored and the bars showed evidence of surface corrosion but there was no evidence of corrosion being accelerated by the stress, or suggestion that embrittlement was effected.

Austenitic iron failed in a comparatively short time under a load of 20,000 psi, 0.7 hr on one bar, 7.7 hr on another, both from the same heat. Decrease of stress to 12,000 psi increased the time to fracture by almost 3 times. At a stress of 10,000 psi load was sustained for 2600 hr without fracture. The curves indicate that there is a limiting stress below which failure will not occur in caustic solutions. (This is confirmed by long-time service experience with the condition and material.) Above the limiting stress very slight increases in stress profoundly shorten the time to failure. This indicates the desirability of suitable choice of design stress, and measurements under operational conditions to reveal it.

The tensile strength of iron A, cast as a 0.5-in-diam specimen was 38,500 psi. The specimen removed unbroken after 270 hr in hot caustic under a stress of 11,000 psi was subsequently broken at room temperature and showed a tensile strength of 37,900 psi. A piece of the test specimen which failed after 43 hr at 12,000 psi was broken at room temperature, giving a tensile strength of 38,600 psi. It would appear that the caustic is effective only in lowering the time to fracture under stress when material is under the corrosive and temperature influence.

Beam Loadings. Results obtained from beam loadings of specimens are recorded in Table 3, and plotted in Fig. 4. The upper curves show the tensile fiber stress as calculated from the beam formula, the lower from conversion of actual strain measurements according to the method described in the Appendix. The marked error occasioned by use of the beam formula is evident.

These results are in good agreement with those found for tensile loading, the limiting or threshold stress being about 10,000 psi.

Conclusion

These studies show that for the corrosive and temperature considered there is a limiting stress below which the effects of stress corrosion on austenitic irons are either negligible or nonexistent. This stress is relatively high in respect to contemplated stresses for cast-iron products. Thus the good corrosion resistance of these metals in hot caustic can be advantageously utilized as sound engineering practice. This confirms field experience.

As shown by these studies (and described in the Appendix), the SR-4 resistance-strain-gage method is a valuable tool for the metallurgist and test engineer in their evaluation of quasi-elastic materials.

TABLE 3 RESULTS OF BEAM-LOADING SPECIMENS

Calculated stress from beam formula, psi	Stress from stress-strain curve, psi	Fracture time, hr: Austenitic cast iron C, simple beam	Austenitic cast iron C, cantilever beam	Austenitic cast iron D, cantilever beam
49200	26300	...	...	0.1
46800	26000	...	0.5	...
41400	24200	...	1.8	...
39600	23700	...	...	0.7
31100	20800	...	...	1.2
25800	18400	...	12.4	...
24900	18000	...	...	5.0
23200	17000	8.1	...	...
22600	16700	...	...	7.1
22000	16400	...	236	...
20400	15400	...	...	15.5
19650	14800	243[a]	...	...
18200	13900	...	...	152
16370	12600	...	...	468

[a] Not broken.

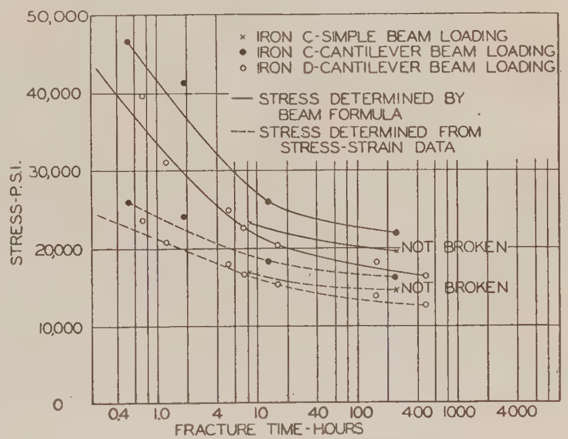


Fig. 4 Stress-Corrosion Rate for Austenitic Cast Irons; Transverse Loading

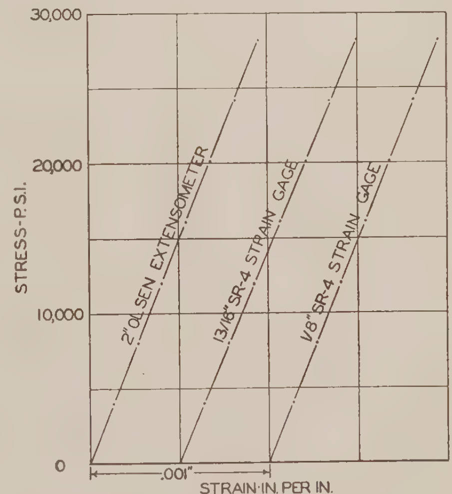


Fig. 5 Comparison of Strain Gages

(Gages mounted on steel tensile-test specimen. Readings were taken simultaneously on all gages.)

Appendix

The foregoing paper indicates the necessity of correct stress evaluation in studies of stress-corrosion effect.

Stresses due to tensile loading are determined readily and directly. Those due to various transverse loadings are not so easily determined. This is especially so in case of quasi-elastic materials, such as cast irons, where the premises of beam formulas do not apply, and strain measurements by conventional extensometric devices are not readily made.

SR-4 resistance-type strain gages were used to determine the

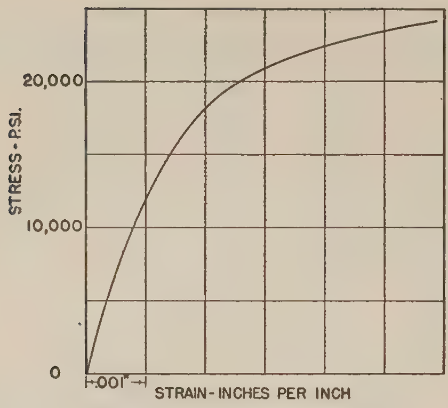


Fig. 6 Tensile Stress-Strain Curve for Austenitic Cast Iron

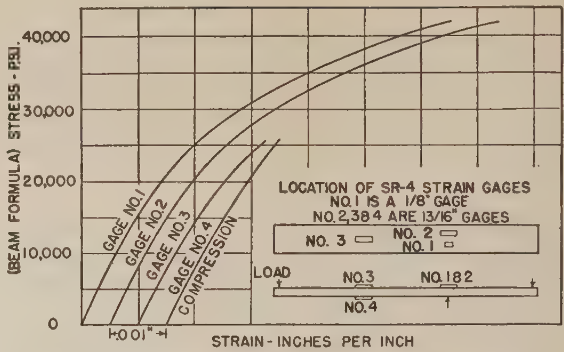


Fig. 7 Stress-Strain Curve on Austenitic Cast Iron
(Simple beam loading.)

strain readings for the transverse loading of the stress-corrosion tests. To check the accuracy and sensitivity of the SR-4 strain gages a steel tensile-test specimen was made long enough to accommodate an Olsen extensometer (2 in. gage length), a $^{13}/_{16}$-in. SR-4 strain gage, and a $^{1}/_{8}$-in. SR-4 strain gage. Simultaneous readings of strains and applied load were taken. The results given in Fig. 5 show good agreement, especially when it is borne in mind that different gage lengths are involved.

A typical tensile stress-strain curve for austenitic cast iron is shown in Fig. 6. Four SR-4 strain gages were mounted on a specimen as shown in Fig. 7, and the specimen was subjected to simple loading in the setup shown in Fig. 2. The applied loads were converted to stress as indicated by the beam formula and plotted against the corresponding strain as shown in Fig. 7. The apparent moduli by beam-formula fiber stress versus strain are considerably higher than those obtained from tensile stress-strain data especially at higher stresses. This emphasizes the fact that stresses determined from beam formulas deviate considerably from the true stresses. The stress-strain data shown in Fig. 7 were obtained on as-cast specimens not subjected to previous stressing. Cast irons are quasi-elastic materials. Therefore stressing will result in plastic deformation and alter the stress-strain characteristics of the material. When strain determinations on a quasi-elastic material are converted to stress, it is imperative that the stress-strain data used in the conversion be obtained on material of similar prior stress history.

Modern Methods in the Heat-Treatment of Steel

By E. R. Mertz,[1] Burbank, Calif.

The construction of time-temperature-transformation curves (S-curves) and their use in developing the interrupted-quench processes for steels are treated in this paper. Of these processes, martempering, austempering, and cyclic annealing are explained in detail. Martempering results in greatly reduced stresses in quenched parts, as compared to the conventional quench and temper method of hardening steel, and hence much less cracking and distortion is apparent. Austempering, an isothermal process producing bainite, develops surprisingly high impact properties in the region of 250,000 psi or 50 Rockwell C. Cyclic annealing is an elevated-temperature isothermal annealing process to produce a predetermined degree of softness, and saves much valuable time in the annealing of the higher alloy steels. The use of hardenability curves in establishing size limitations for particular heats of steel and in selecting accurate tempering temperatures is demonstrated.

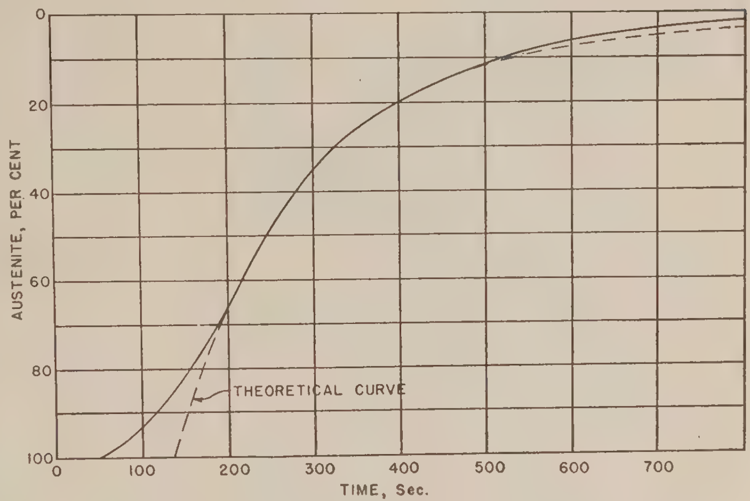


Fig. 1 Reaction Rate of Austenite Compared to Theoretical First-Order Reaction Curve

Introduction

Most of the processes involved in the heat-treatment of steel depend upon the decomposition of austenite into one or more of its transformation products, and the physical properties produced depend upon the nature of the product or products formed. Austenite is a solid solution of carbon in iron which is stable (for most steels) only at elevated temperatures, and when it is cooled below a certain critical temperature the normal solubility of carbon in iron is reduced to a negligible amount. As a result, a reaction occurs which involves a change in the crystallographic structure of the iron, and in most cases the formation of iron carbide and/or other carbides. Bain and Davenport (1)[2] demonstrated that the product formed from the decomposition of austenite was dependent upon the temperature at which the reaction occurred rather than the speed at which the austenite was cooled.

Austenite Decomposition; S-Curves

Fig. 1 illustrates how closely the isothermal transformation of austenite follows a first-order chemical reaction in which the rate of reaction at any instant is proportional to the amount of reacting material remaining. However, there are several notable deviations from a true first-order reaction. Such a reaction begins immediately at a maximum rate and continues at a constantly diminishing rate. Theoretically, it never reaches completion.

Austenite apparently goes through an incubation period before decomposition starts, does not reach its maximum rate until the reaction has progressed an appreciable amount, and does reach completion in a finite time.

Bain and Davenport (1) studied isothermal transformation rates of austenite at various subcritical temperatures and plotted curves of temperature versus time for beginning and end of transformation. The result was the now familiar "S-curve."

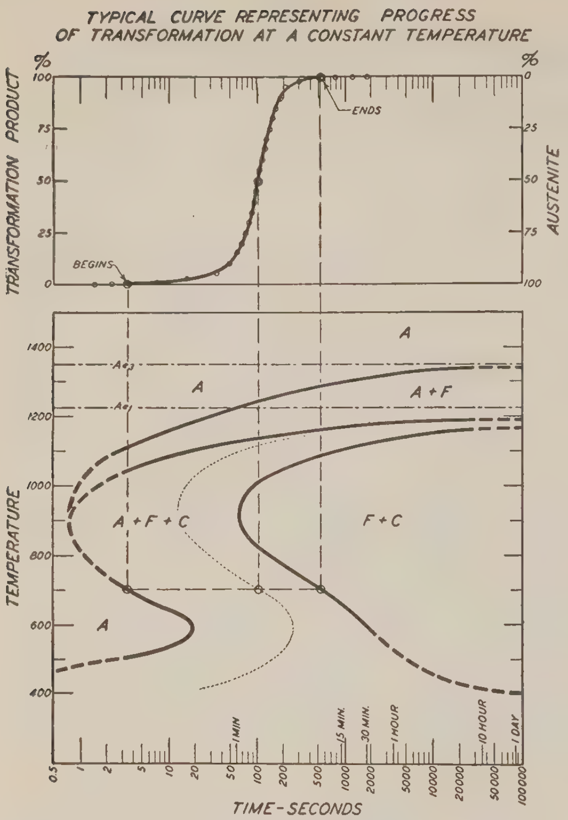
Fig. 2 illustrates (2) how these curves were constructed. The progress of transformation at a chosen temperature was followed by quenching numerous thin samples from an austenitizing temperature into a molten metal bath maintained at the chosen temperature. Periodically, specimens were drastically quenched from the constant-temperature bath so that any untransformed austenite would be converted to martensite, the product formed from austenite at low temperatures. Microscopic examination of such specimens properly etched revealed the dark-etching pearlite or bainite in contrast to the lighter-etching martensite.

Thus accurate data on the beginning and virtual end of transformation of austenite to pearlite or bainite were established, but the data on austenite to martensite transformation remained in doubt. Inasmuch as the formation of martensite apparently involves only a space-lattice change, it does not require time for diffusion. Hence many metallurgists were of the opinion that time was not a function of the process, and temperature was the controlling factor.

[1] Associate Professor in General Engineering, University of Southern California, Los Angeles, Calif. Formerly Research Engineer, Lockheed Aircraft Corporation.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Metals Engineering Division and presented at the Aviation War Conference, Los Angeles, Calif., June 11–14, 1945, of The American Society of Mechanical Engineers.



TYPICAL ISOTHERMAL TRANSFORMATION DIAGRAM (2)

Fig. 2

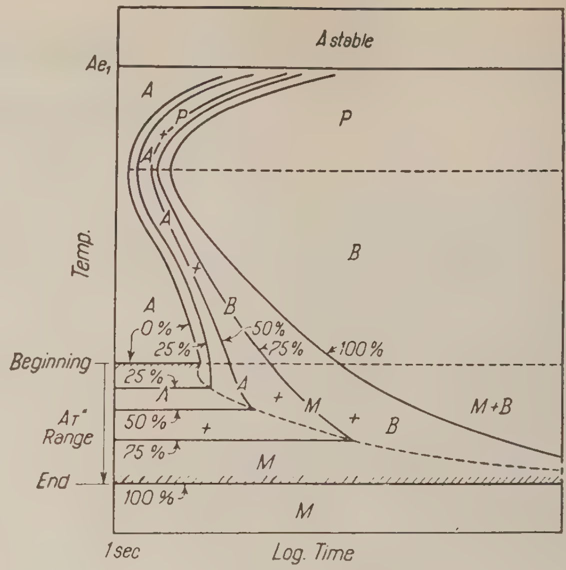


Fig. 3 Modified S-Curve Diagram

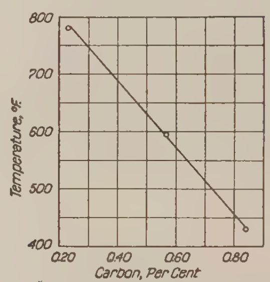


Fig. 4 Effect of Carbon on the M_s of Plain-Carbon Steel

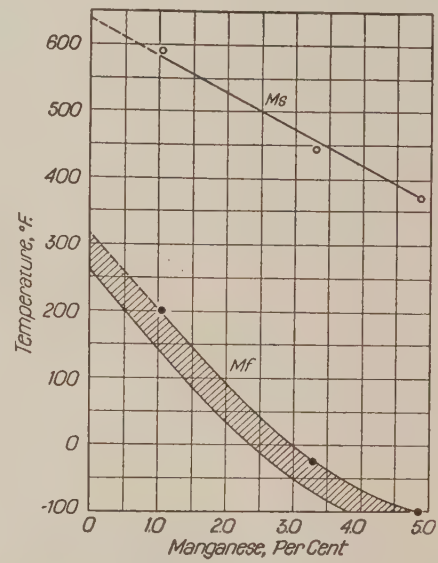


Fig. 5 Effect of Manganese on the M_s and M_f of 0.50 Per Cent Carbon Steel

Martensite Formation; M_s and M_f Points

In a recent investigation Greninger and Troiano (3) were able to follow the progress of austenite to martensite formation by tempering for a short time at a higher temperature the specimens quenched into a low-temperature isothermal bath. The tempering treatment caused any martensite formed to etch darker than freshly quenched martensite, and microscopic examination revealed it in sharp contrast to that formed by quenching after the tempering treatment. The lighter-etching martensite formed by quenching after tempering, represented the untransformed austenite from the original isothermal quench. The results of this investigation established that martensite formation was essentially dependent upon temperature alone. At a given temperature a certain percentage of martensite was formed almost instantaneously and no more was formed unless the temperature was lowered. A constant-temperature product began to appear after a considerable time lag, which amounted to days or weeks at room temperature. This latter product was no doubt a result of diffusion, and was bainitic in structure.

Fig. 3 represents a modified S-curve, or time-temperature-transformation (TTT) curve constructed by Cohen (3), in which the "timeless" nature of martensite formation is incorporated. The dotted line represents the beginning of formation of the isothermal product, which Cohen includes with bainite.

The temperature indicating the beginning of formation of martensite has been designated M_s, and that representing the end, M_f. Fig. 4 is a curve constructed by Greninger (4) showing that the M_s point decreases linearly with increasing carbon content.

Alloys have a similar effect on both the M_s and M_f point, as indicated in Fig. 5 by Payson and Savage (5). They have submitted the following formula for calculating the M_s point, provided, of course, that the austenitizing temperature used is one at which all carbides will be completely in solution in the austenite.

$$M_s \text{ in deg F} = 930 - 570\,C - 60\,Mn - 50\,Cr - 30\,Ni - 20\,Si - 20\,Mo - 20\,W$$

In many steels of high-carbon and high-alloy content the M_f point is well below room temperature, and considerable amounts of austenite are retained by conventional heat-treating procedures. The distortion of accurate gages after long storage may be explained by the eventual transformation of such retained austenite to the isothermal product formed at the storage temperature. Subzero treatment of such steels often transforms much of the retained austenite to martensite, giving a more uniform product, and in many cases a much harder one.

Interrupted-Quench Processes

Complete hardening of a steel part requires the formation of essentially 100 per cent martensite, necessitating a cooling rate at the center fast enough to miss the "nose" of the S-curve where much softer products are formed. Fig. 6 illustrates (6) the conventional quenching process. With rapid quenching the temperature gradient is sufficient to allow the surface of the steel to be almost wholly martensitic while the center is still wholly austenitic. Thus when the center does transform, the expansion due to the austenite-martensite transformation produces tensile stresses in the rigid outer layer of martensite which are often sufficient to produce cracking or at least severe distortion. Such conditions may be minimized by quenching into an isothermal bath above the M_s point and holding at temperature long enough to equalize the temperature throughout the cross section. Air-cooling from this temperature will produce complete hardening, and the thermal gradient will be small enough to allow martensite formation to take place more uniformly from surface to center, minimizing residual stresses. This process is called "martempering," and is illustrated in Fig. 7 (6). The quenching is followed by the usual tempering treatment to produce desired properties.

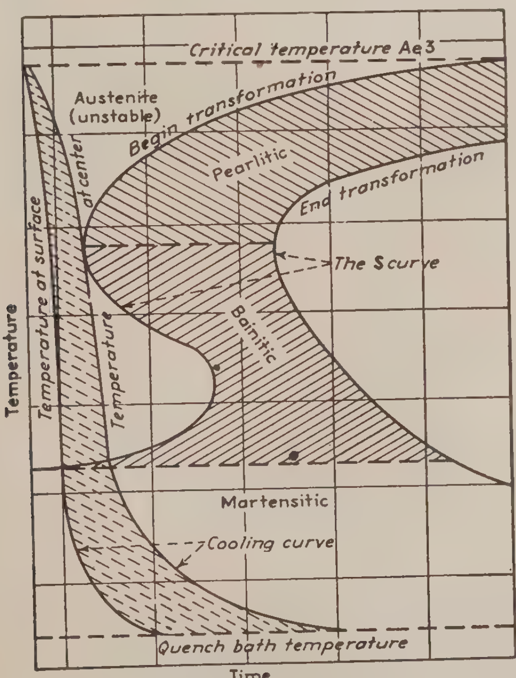


Fig. 6 Cooling Curves in Conventional Quench and Temper Method in Relation to the TTT Curve

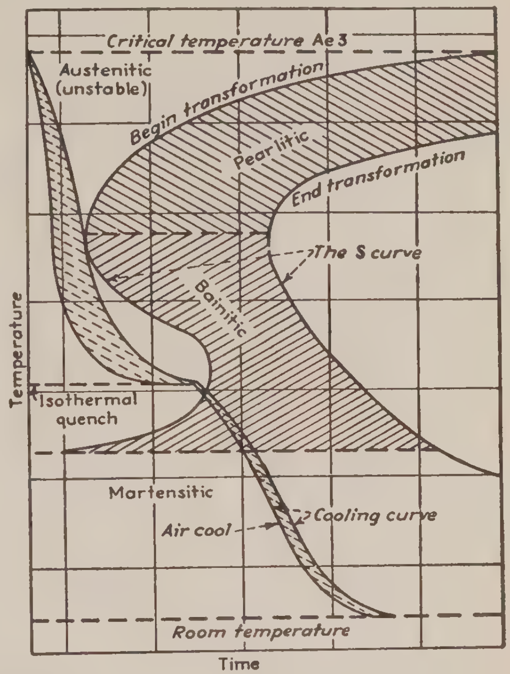


Fig. 7 Cooling Curves in Martempering Method in Relation to the TTT Curve

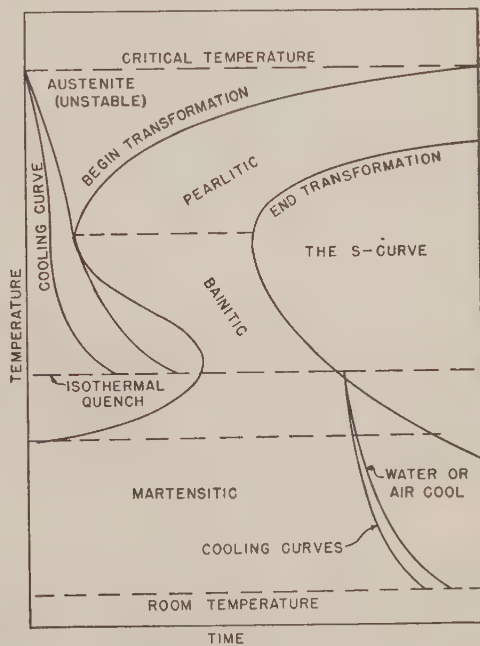


Fig. 8 Cooling Curves in Austempering Method in Relation to the TTT Curve

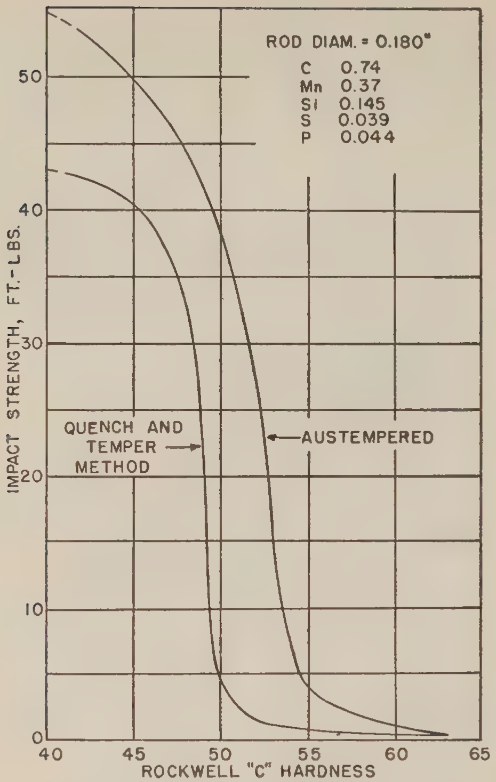


Fig. 9 Comparison of Impact Strengths at Various Hardnesses of Austempered and Quenched-and-Tempered Steels

Fig. 8 illustrates (6) a method of heat-treatment which eliminates the need for tempering. The process, known as "austempering," consists of quenching the steel into a bath held at a temperature where bainite is formed and allowing the austenite to transform isothermally to bainite. The selection of bath temperature determines the hardness of the final product. Steels treated in this manner show considerable improvement in impact resistance, especially in the vicinity of 50 Rockwell C hardness or 250,000 psi, as illustrated in Fig. 9 (7). This process has been applied successfully in the heat-treatment of certain springs, knives, chisels, and other parts where toughness combined with hardness is of importance.

The annealing of steels by furnace-cooling to produce a soft, machinable structure is often a very slow process, especially in the higher-alloy grades which require very slow cooling to avoid some hardening. Much valuable time may be saved by the use of the cyclic annealing process illustrated in Fig. 10 (6). An air furnace operating at constant subcritical temperature may often be substituted for the isothermal quench with little loss of time. The selection of isothermal temperature is dictated by a compromise between time consumption and desired hardness. Higher temperatures produce softer structures, but require longer time.

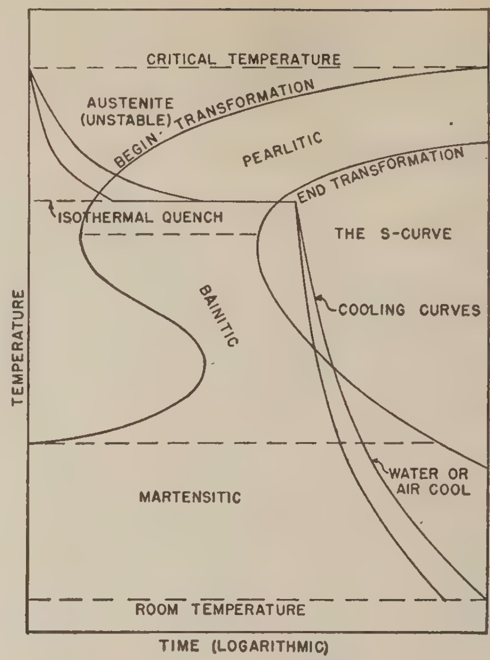


Fig. 10 Cooling Curves in Cyclic Annealing Method in Relation to the TTT Curve

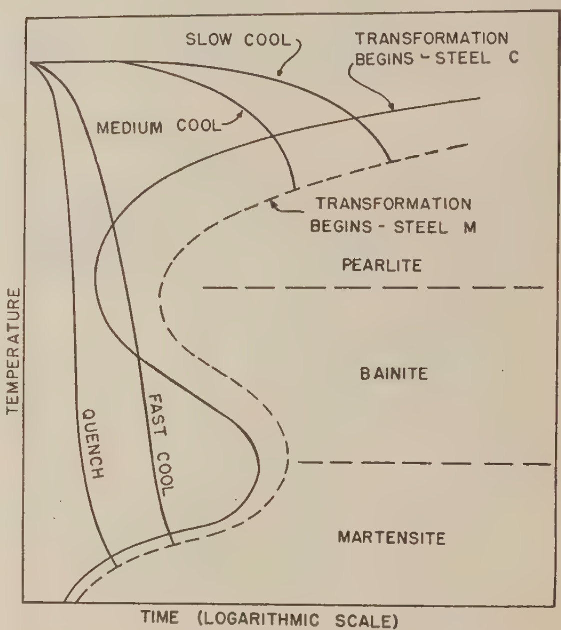
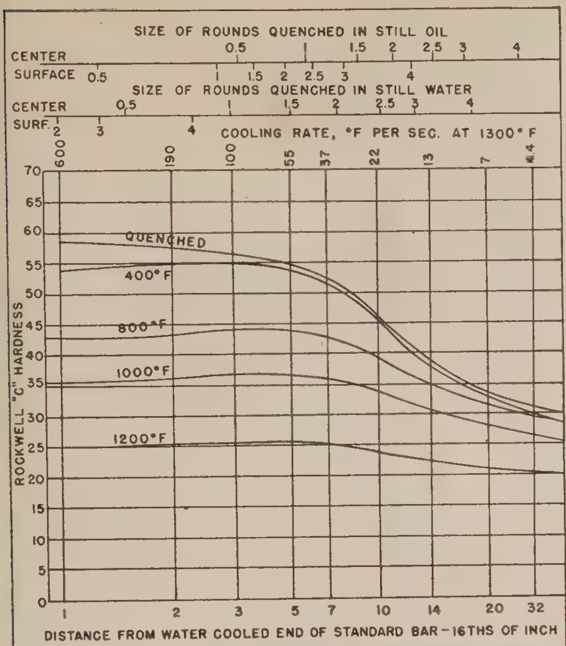


Fig. 11 Comparison of Cooling Rates to Obtain Desired Structures for Two Steels

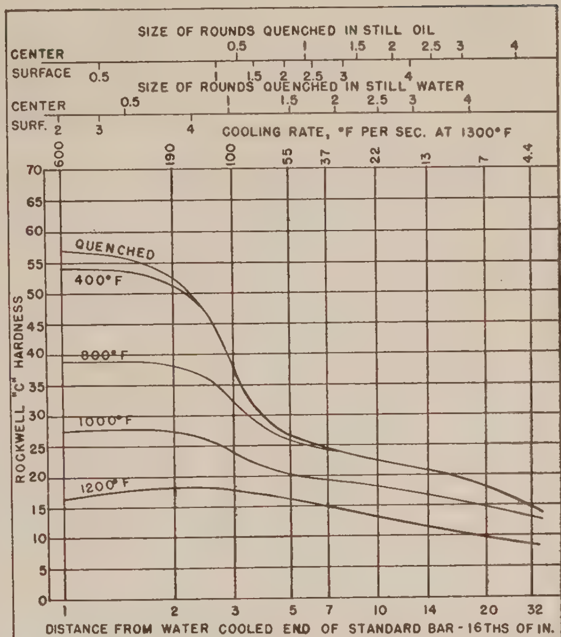
Hardenability

The severity of quench required for complete hardening of a steel depends upon the reaction rate of that steel at the "nose" of the S-curve. If the cooling curve crosses the nose some pearlite and bainite will be formed, and a softer structure obtained. Another steel with more alloy might be completely hardened by the same quench, indicating that its reaction rate at the nose of the S-curve was slower. Fig. 11 illustrates (7) this comparison. The fast cool completely hardens steel M, but incompletely hardens steel C, inasmuch as it crosses the nose of the S-curve. The construction of an S-curve for a given heat of steel is very time-consuming, so other methods of determining the hardenability of steels are used. Most popular is the end-quench hardenability



STEEL	C	Mn	P	S	Si	Ni	Cr	Mo	QUENCHING TEMP.	HT. NO.	GRAIN SIZE
8740	.40	.88	.024	.027	.27	.62	.53	.24	1575 °F	—	8

Fig. 12 Standard End-Quench Hardenability Curves for an NE-8740 Steel



STEEL	C	Mn	P	S	Si	Ni	Cr	Mo	QUENCHING TEMP.	HT. NO.	GRAIN SIZE
1040	.43	.91	.017	.040	.20	—	—	—	1575 °F	—	6-7

Fig. 13 Standard End-Quench Hardenability Curves for an S.A.E. 1040 Steel

test devised by Walter E. Jominy. This test makes use of the fact that a heated steel bar, quenched only on the end by a jet of cold water, will have a very rapid cooling rate at the point of impingement of the water jet, and a continually decreasing cooling rate at increasing distances from the quenched end. Inasmuch as such cooling rates have been established, a longitudinal hardness traverse of an end-quenched bar will determine what cooling rates are necessary to produce specific hardnesses in the steel from which the bar was made. By correlation of these cooling rates with known cooling rates in fully quenched bars, the hardnesses in such bars may be predicted.

Figs. 12 and 13 show end-quench hardenability curves for two steels. These are supplemented by curves for the same steels tempered at various temperatures.

Correlation of Hardenability Data to Cross-Sectional Hardness Data

Figs. 14 and 15 are correlation charts to be used in conjunction with hardenability curves to convert rapidly end-quench hardness values to cross-sectional hardness values. These charts were constructed by plotting established cooling rates (8) as abscissas against section size as ordinates. Cooling rates are necessarily

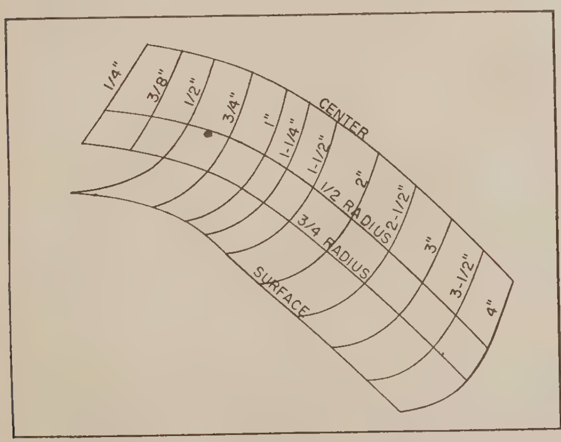


Fig. 14 Correlation Chart; End Quench to Section Hardness; Oil Quench

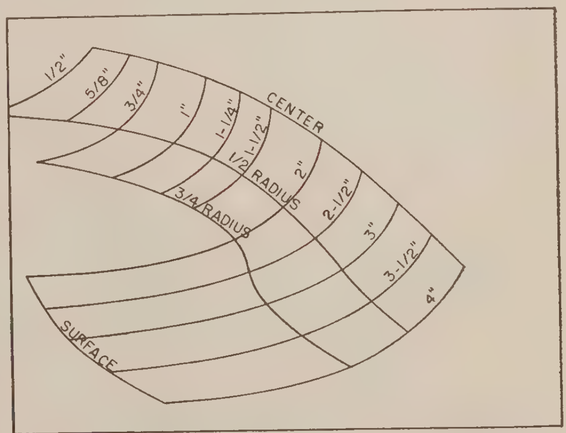


Fig. 15 Correlation Chart; End Quench to Section Hardness; Water Quench

on the same scale as those which appear in Figs. 12 and 13. Construction lines are deleted to avoid confusion. For use, the correlation charts are constructed on transparent paper or copied on photographic film. The use of the water-quench correlation chart is illustrated in Fig. 16. The chart is superimposed on Fig. 13 so that the vertical index line at the left falls on the left border line of Fig. 13. By sliding the chart up or down (keeping the vertical lines indexed) hardness values are read at the intersection of the hardenability curve and the selected bar size and section-location curves. In Fig. 16 point A indicates that the center of a 1-in. round of this heat of S.A.E. 1040 steel would have a hardness of approximately 38 Rockwell C when quenched in water. To find the anticipated hardness halfway between the surface and the center, the correlation chart is moved up until the junction of the half-radius and 1-in. curves intersects the hardenability curve. Here a hardness of 48 Rockwell C is obtained. The oil-quench correlation chart is used in a similar manner.

These correlation charts permit the rapid translation of hardenability data to cross-sectional hardness data for various tempering temperatures as well as the as-quenched values by using tempering curves like those appearing in Figs. 12 and 13. They also permit the rapid determination of hardness depth for incompletely hardened parts and the establishment of limiting size of a given steel for a particular quench when a minimum acceptable hardness has been decided upon.

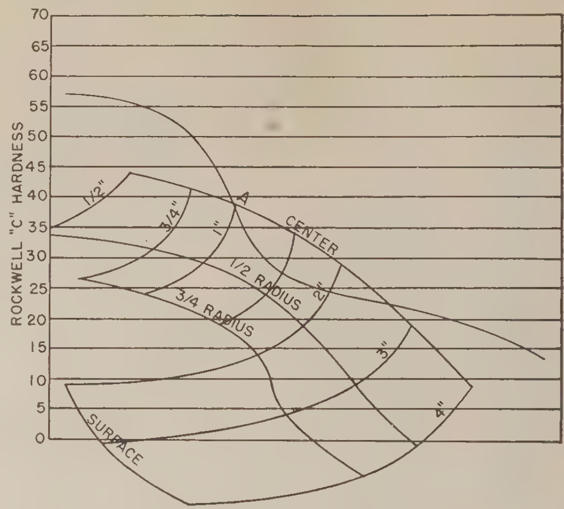


Fig. 16 Use of Water-Quench Correlation Chart on End-Quench Hardenability Curve for an S.A.E. 1040 Steel

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Elevated-Temperature Stretch-Flanging of Some Aluminum Alloys

By E. G. Thomsen,[1] D. M. Cunningham,[2] and J. E. Dorn[3]

Many aircraft parts consisting of a flat web and a concave flange are produced by rubber-die-forming on a hydropress. The forming action consists of bending the flange around the die radius and stretching the outer fiber of the flange from its original length to that which is achieved when it contacts the die. Several types of failures and defects are encountered, the most important being a radially directed fracture initiated at the outer fiber of the flange. When this fracture occurs without necking, the maximum per cent stretch that may be produced approximates the per cent local ductility in the tension test. If, however, fracture is preceded by necking, the permissible per cent stretch is intermediate between the uniform and local per cent strain in tension. Increasing the forming temperature permits greater per cent stretches for some of the more-ductile alloys.

Introduction

Many aircraft parts consist of a flat web with a concave flange. Such parts are usually produced by rubber-die-forming on a hydropress at atmospheric temperature. Although many aircraft materials exhibit ample deformation to permit successful forming of concave flanges, other materials, particularly those in the aged condition, have such limited deformability that satisfactory flanges cannot be produced. The Western Aircraft War Production Council recommended that the War Metallurgy Committee investigate the mechanics of concave flanging and the possibilities of improving this operation by forming at elevated temperatures. This report is based on investigations conducted by Project NRC-548 for the Office of Production Research and Development of the War Production Board.

The rubber-die hydropress-forming of concave flanges has been described in several papers.[4,5,6] A blank of radius R_b, as illustrated in Fig. 1, is placed on a die block having a contour radius R_c somewhat greater than R_b. As the rubber contained in the head of the press is impressed against the die, the blank is bent around the die radius along the die contour line until the formed flange touches the die. Since the extreme fiber of the flange is stretched from a radius R_b to the larger radius it assumes when contact is achieved, the operation is commonly called stretch-flanging.

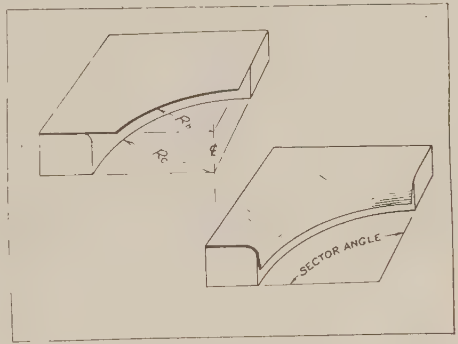

Fig. 1 Blank and Flanged Part of Die

Experimental Procedure

Scope of Investigation. Tests were planned to determine the stretch-flanging limits of a number of aluminum-alloy sheet materials at forming temperatures of 70 F, 300 F, 400 F, and 450 F. The maximum temperature of 450 F was selected because of rapid deterioration of the rubber pad at higher temperatures and possible detrimental changes in physical properties of some of the precipitation-hardened alloys. The general survey of the effect of temperature on the stretch-flange formability was carried out on a 4-in-contour-diam die with blanks having 360-deg flange sectors. Spot tests were made on other variables, such as other contour diameters, sheet thicknesses and sector lengths, in order to observe the major trends introduced by these variables.

Materials. The standard aluminum sheet materials that were investigated included 3S-O, 52S-O, 24S-O, 61S-O, 75S-O, 24S-T, 24S-T86, 61S-T, R301-T, and 75S-T covering thicknesses from 0.020 in. to 0.081 in.

Equipment. The hydropress consisted of a modified 200-ton aircraft-type press. The platen was 15 in. wide and 28 in. long arranged with electric-resistance heaters and automatic temperature control.

The normal platen pressure was 1050 psi, but pressures as high as 7000 psi were obtained with special platens and rubber heads. The mild-steel die blocks had contour diameters of 3, 4, 6, and 8 in.; $^3/_{16}$-in. bend radii were provided on all dies in order to eliminate bend failures.

Specimens. The specimens were sheared from sheet stock with die-locating holes and blank pilot center hole jig-drilled. The desired blank-hole diameter was machined with a fly cutter to an accuracy of 0.010 in. of the nominal blank-hole diameter. All blank holes were polished with 2/0 Behr Manning polishing paper after machining, in order to provide uniformly controlled edge conditions.

Experimental Technique. The specimen blanks were preheated to the forming temperature for 10 min in an air furnace and rapidly transferred to the die block. A light layer of asbestos powder was screened over the blank in order to prolong the useful

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[1] Assistant Professor of Mechanical Engineering, University of California, Berkeley, Calif.

[2] Lecturer in Mechanical Engineering, University of California.

[3] Associate Professor of Physical Metallurgy, University of California.

[4] "Forming Methods—Forming by Single Action Hydraulic Press," International Textbook Company, Bureau of Aeronautics, Navy Department, Washington, D. C., 1943.

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Presented at the National Meeting of the Aviation Division, Los Angeles, Calif., June 3–6, 1946, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

life of the rubber pad. The press head was then moved against the platen to a predetermined pressure which permitted complete forming of the blank. A series of blanks were thus formed with blank-hole diameters varying in steps of 0.10 in. (flange widths varying in 0.05-in. steps). The critical blank diameter was evaluated as the minimum diameter that was successfully formed. Frequently in the region of the critical diameter a specimen would fracture, whereas a duplicate specimen would form successfully. In this case the critical value of blank diameter was taken as the larger and more conservative value.

In order to obtain forming temperatures in the blank approaching those of the die it was necessary to hold the blank under light pressure for 1 min between rubber pad and heated die before forming the stretch flange. All temperatures reported herein are those of the heated die but represent the blank temperature to within 10 deg F.

Geometry of Stretch Flanges. The simplest type of stretch flange consists of a 360-deg sector produced on a block having a 90-deg die angle. For this type flange the per cent stretch of the outer fiber of the flange is simply

$$\text{Per cent } S = \frac{(R_e - R_b)}{R_b} \times 100 \ldots\ldots\ldots\ldots [1]$$

As smaller values of R_b are employed the per cent stretch of the outer fiber of the flange increases. Finally a critical value is reached resulting in fracture of the flange. The maximum permissible stretch is given by

$$\text{Per cent } S_{max} = \left(\frac{R_e}{R_{b\ min}} - 1\right) \times 100 \ldots\ldots [2]$$

and the corresponding true strain is

$$\phi_{max} = \ln \frac{(R_e)}{R_{b\ min}} \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

If the sector angle is less than 360 deg, another geometrical factor enters the analysis. The induced circumferential tensile stresses causing stretching vanish at the sector edges and the per cent stretch that is obtained becomes less than the value calculated from Equation [1]. The actual stretch is therefore less than the theoretical stretch, and consequently flanges of greater heights may be produced without encountering failures.

For small sector angles, the arc length of the flange approaches the corresponding chord length. Since the energy that is required to form a straight flange is less than that required for a curved flange of the same contour length, flanges of small sector angle frequently form straight flanges along the chord.

In order to produce a part having a flange at 90 deg to the web, the part must be slightly overformed so that the desired angle is obtained after springback. The per cent stretch for dies having angles that differ from 90 deg is derived in Appendix 1.

Experimental Results

Types of Fracture. The forming of stretch flanges is limited by several forming defects and failures which are shown in Fig. 2, and are briefly described as follows:

(*a*) *Stretch Rupture.* Stretch rupture, illustrated in Fig. 2(A), is the most common type of forming failure. This type of fracture was encountered in most of the 360-deg flanges that were tested. Two types of fracture were observed. The annealed alloys at all temperatures and some of the precipitation-hardened alloys at 450 F exhibited local necking preceding fracture; some of the precipitation-hardened alloys fractured without necking at the lower temperatures.

(*b*) *Bulging Rupture.* The second type of rupture which limits the formability is shown in Fig. 2(B), and was named a bulging rupture. This type of rupture was obtained when small blank-hole diameters were employed. Instead of forming a flange the blank bulged in a manner similar to that obtained in the absence of a central hole. Fracture occurred along the die-contour radius before the outer fiber reached its limiting ductility. This type of fracture was not obtained when the hole diameter was increased to the critical value for stretch rupture.

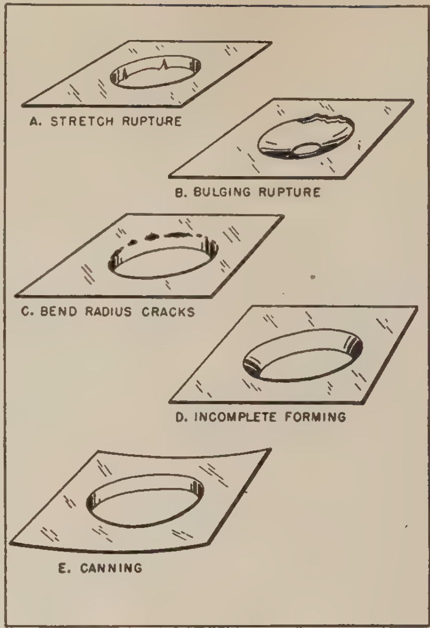


Fig. 2 Forming Defects

(*c*) *Bend Cracks.* If the die bend radius is too small the strains of the fiber on the top surface of the blank may exceed the permissible strain before the outer fiber of the flange reaches its critical stretch value, resulting in fracture of the upper surface of the blank along the die bend radius. This type of fracture is shown in Fig. 2(C). In the present investigation the die bend radii were maintained sufficiently large to prevent the formation of bend cracks.

(*d*) *Incomplete Forming.* In forming stretch flanges the blank is bent as well as stretched. Consequently the pressure required for forming stretch flanges is greater than that required for straight flanges. If the pressure is not great enough an incompletely formed flange results, as shown in Fig. 2(D). The pressures for the present investigation were maintained sufficiently large to insure contact of the outer fiber of the blank and the die.

(*e*) *Canning.* Canning, as shown in Fig. 2(E), results from springback after forming. No attempt was made to evaluate completely the springback after forming, but it was observed that the springback for stretch flanges is slightly less than that for straight bends for the same conditions of forming temperature, sheet thickness, and bend radius.

Maximum Per Cent Stretch. The effect of temperature on the maximum per cent stretch that may be achieved in a number of aluminum alloys is shown in Fig. 3. It is evident from these curves that increased stretch-flange formability is achieved at temperatures above 300 F for all alloys investigated. Each alloy, however, exhibits its own peculiar maximum per cent stretch-temperature curve.

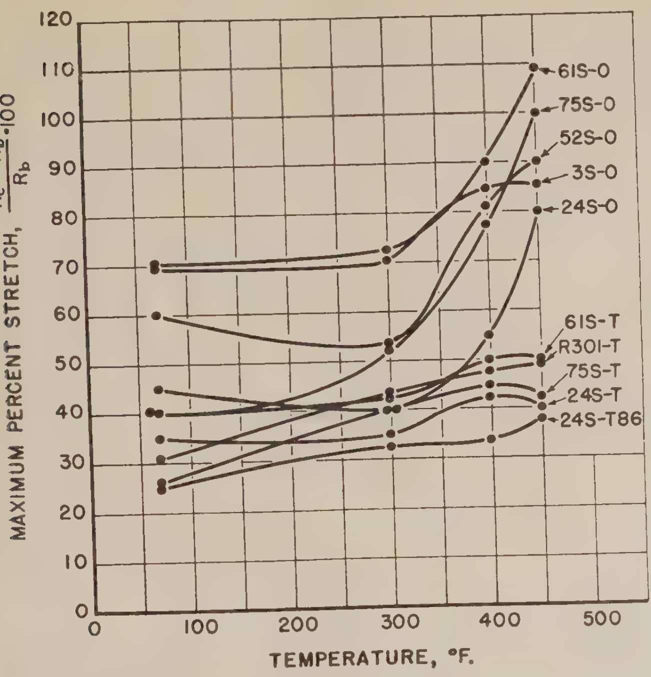


FIG. 3 EFFECT OF TEMPERATURE ON MAXIMUM PER CENT STRETCH

A few tests were made to evaluate the effect of contour radius on the maximum per cent stretch. Materials which necked during flanging yielded decreasing per cent stretch with increasing contour radii.

The effect of thickness was studied for 24S-O and 61S-T formed at atmospheric temperature on a 6-in-contour-diam die. Slightly greater per cent stretch was observed as the thickness was increased from 0.020 to 0.081 in., but results were too varied to warrant a more definite statement.

The effect of the angle of sector of the blank on the maximum per cent stretch is given in Fig. 4 for atmospheric and 450 F forming temperatures. The plain open and solid circles refer to maximum per cent stretch limited by stretch fracture, whereas the crossed symbols refer to maximum per cent stretch limited by fracture at the edge of the segmented blanks near the die bend radius. For the two alloys investigated the maximum per cent stretch with atmospheric-temperature forming rises moderately with decreasing sector angle to a maximum at approximately 90 deg. Blanks with sector angles of 30 deg could not be fractured at all forming temperatures. At 450 F alloy 24S-O could not be fractured with sector angles of 90 deg or less.

Discussion of Results. A major objective of investigations on the formability of metals is the development of a useful philosophy that will permit the prediction of forming limits from a few simple, easily obtainable test data. Most metal-forming processes, however, are so complex that this objective has not been completely achieved; stretch flanging is not an exception. The results of this investigation, however, show that the forming limit for stretch flanges can be approximated in a few simple cases.

The fracture causing a stretch-rupture failure during forming starts at the outer fiber of the flange. Although, in some alloys, local plastic action precedes fracture in this critical region, in other alloys fracture occurs without evidence of necking. Many alloys that neck when tested in simple tension do not neck when stretch-flanged.

An estimate of the maximum per cent stretch is possible for alloys that do not neck when stretch-flanged. Since the outer fiber of the flange is known by grid analyses to be essentially stressed by simple circumferential tension, it is logical to assume that stretch rupture is initiated when the circumferential tension stress equals the true fracture stress in tension. This assumption implies that the maximum true strain for stretch-flange forming should approximate the true local strain in tension defined by

$$\phi_{\max} = \ln \frac{A_0}{A_f}$$

where

A_0 = original cross-section area
A_f = cross-section area at fracture

A comparison of the maximum true stretch with tension data,[7] obtained from standard A.S.T.M. specimens, shown in Fig. 5, illustrates the agreement that may be expected. Materials 24S-T86 at all temperatures, R301-T and 75S-T up to 300 F, and 24S-T up to 400 F exhibited stretch ruptures without evidence of necking. For these conditions the maximum true stretch approximated the true local strain in tension. In all other cases necking preceded fracture, and the maximum true stretch that could be achieved was intermediate between the true uniform and the true local strains in tension. Thus the maximum true stretch that can be achieved in the absence of necking approaches the true local strain at fracture in tension.

No satisfactory analysis is available for predicting the stretch-flange-forming limits of materials under conditions where necking is obtained; the maximum true stretch is significantly greater than the true uniform strain in tension. Since necking is known to begin when the resistance to the forming load passes a maximum value, it is logical to suspect that necking in stretch-flanging requires a complete analysis of the stress and strain distribution in the entire flange. Thus changes in contour diameter may affect

[7] "Tensile Properties Affecting the Formability of Aluminum-Alloy Sheet at Elevated Temperatures," by A. E. Flanigan, L. F. Tedsen, and J. E. Dorn. To be published in the *Journal of the Aeronautical Sciences.*

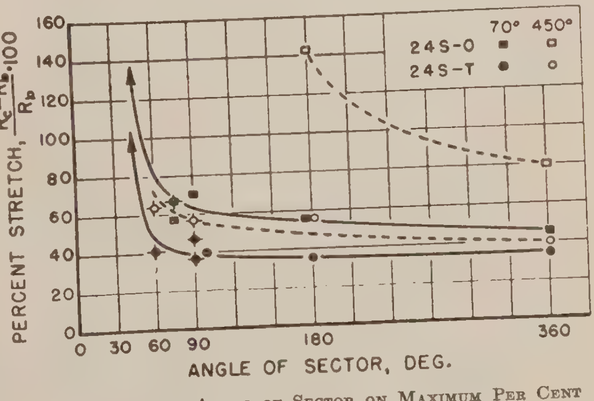
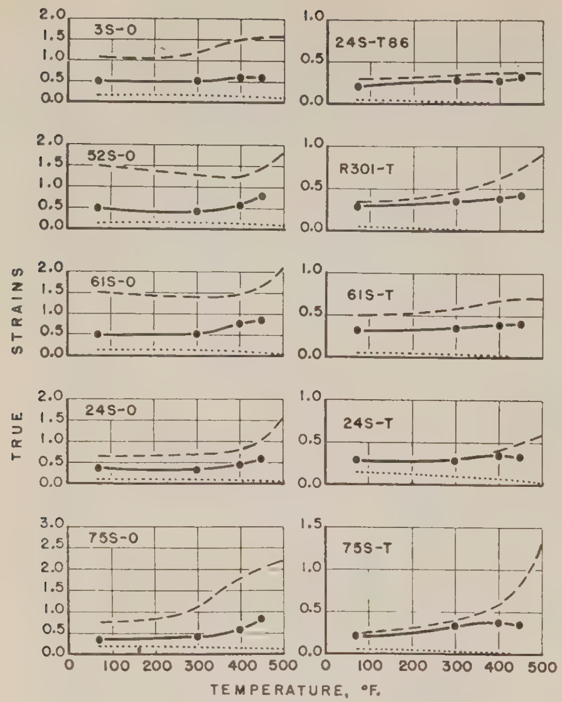


FIG. 4 EFFECT OF ANGLE OF SECTOR ON MAXIMUM PER CENT STRETCH



$\ln(A_0/A)$ = True strain at fracture in the tensile test.

$\ln(l_u/l_0)$ = True maximum uniform strain in the tensile test.

$\ln(R_c/R_b)$ = Maximum stretch in stretch flanging.

FIG. 5 COMPARISON OF STRETCH-FLANGING DATA WITH TENSILE DATA

the per cent stretch. The complex distribution of stresses in the flange are not yet known and therefore complete knowledge of stretch-forming limits will have to await the development of this involved analysis.

SUMMARY AND CONCLUSIONS

1 Forming of 360-deg-sector stretch flanges in a number of aluminum alloys is limited by stretch rupture, radius-bend cracks, or available press pressure.

2 Materials 24S-T, 24S-T86 and 75S-T fractured without evidence of necking when formed at 70 F to 300 F; for all remaining materials and temperatures investigated, fracture was preceded by necking except R301-T at 70 F.

3 The maximum per cent stretch obtained with 24S-T, 24S-T86, and 75S-T agrees closely with the local ductility in tension when formed at temperatures of 70 F to 300 F; between 300 F and 450 F some necking preceded fracture.

4 The maximum per cent stretch for 3S-O, 52S-O, 24S-O, 61S-O, 75S-O, and 61S-T falls between the local and the uniform per cent strain in tension for all temperatures investigated.

5 Higher stretch-flange-formability may be achieved with a 3-in. than with an 8-in-contour-diam die at a forming temperature of 70 F for alloys 24S-O and 61S-T.

6 Increasing thickness from 0.020 in. to 0.064 in. results in higher stretch-flange-formability for alloys 24S-O and 24S-T when formed on a 6-in-contour-diam die and 70 F.

7 Sectors may be formed to a greater flange height than 360-deg flanges for 24S-O and 24S-T.

8 Springback of stretch flanges is slightly less than for straight bends for the same die radius and sheet thickness.

ACKNOWLEDGMENT

The authors wish to acknowledge the support of the Office of Production Research and Development of the War Production Board on this investigation. They are particularly indebted to the Western Aircraft War Production Council for suggestions and assistance in planning this investigation. The authors also wish to thank Messrs. T. Robinson, V. Gudmendsen, and Miss Barbara Roney, for their assistance on the experimental phases of the investigation.

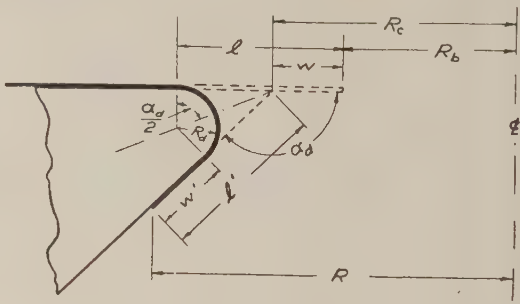


FIG. 6 GEOMETRY OF STRETCH FLANGE

Appendix 1

PER CENT STRETCH FOR 360-DEG FLANGES

Assuming that the changes in thickness and length are minor, an approximate calculation of the effect of the die angle on the stretch is readily obtainable from Fig. 6.

Let

$$l' = w' + R_d \tan\frac{\alpha_d}{2}$$

$$w' = l - \alpha_d R_d$$

$$l' = l - \alpha_d R_d + R_d \tan\frac{\alpha_d}{2}$$

$$l = w + R_d \tan\frac{\alpha_d}{2}$$

$$l = R_c - R_b + R_d \tan\frac{\alpha_d}{2}$$

$$l' = R_c - R_b + R_d \tan\frac{\alpha_d}{2} - \alpha_d R_d + R_d \tan\frac{\alpha_d}{2}$$

$$R = R_c - l'\cos\alpha_d = R_c - \left[R_c - R_b + R_d\left(2\tan\frac{\alpha_d}{2} - \alpha_d\right)\right]\cos\alpha_d$$

$$\text{Per cent } S = \left(\frac{R - R_b}{R_b}\right)\cdot 100 = \left[\left(\frac{R_c}{R_b} - 1\right)(1 - \cos\alpha_d) - \frac{R_d}{R_b}\left(2\tan\frac{\alpha_d}{2} - \alpha_d\right)\cos\alpha_d\right]\cdot 100$$

Accordingly, the true strains of the outer flange fiber are

$$\phi = \ln\frac{R}{R_b} = \ln\left[\frac{R_c}{R_b} - \left\{\frac{R_c}{R_b} - 1 + \frac{R_d}{R_b}\left(2\tan\frac{\alpha_d}{2} - \alpha_d\right)\right\}\cos\alpha_d\right]$$

Actually, the length l' is somewhat smaller than assumed in the foregoing analysis owing to simultaneous thinning of material and shortening of the flange during stretching. Usually this factor may be neglected, as it is generally a small correction term.

Folding in Tube-Sinking

By GEORGE SACHS[1] and W. M. BALDWIN, JR.[2]

An experimental study of the folding tendency of tubes during the process of sinking was made. It was determined that tubes with a fold-free point, having a wall thickness of less than 1 per cent of the diameter, can be sunk without folding. However, if it is attempted to sink tubes with points containing folds beyond a certain critical reduction, the fold contained in the point will be perpetuated throughout the length of the tube. This critical reduction increases almost linearly with the ratio of thickness to diameter of the tube, and is little affected by the die angle, the metal, and its temper. The length of the fold during its formative period is found to increase with increasing reduction, with decreasing relative thickness, and to be relatively unaffected by die angle. The stress required to sink a tube with folds was found to increase only slowly beyond the limiting stress for sinking such a tube without folding to the smallest possible diameter.

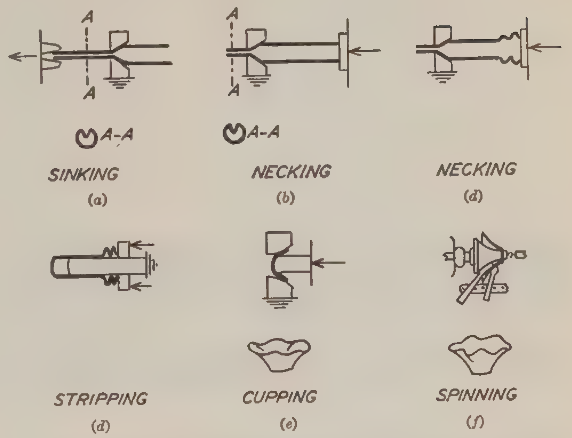


Fig. 1 Various Industrially Important Plastic-Forming Operations in Which Folding Can Occur

Nomenclature

The following nomenclature is used in the paper:

A = cross-sectional area of tube
D = mean diameter of tube
L = gage length of tube
P = force required to sink tube
s_1 = average longitudinal tension in sinking
t = wall thickness of tube

Note: o as a superscript refers to original dimensions. Primed symbols refer to final dimensions.

Introduction

Certain metal-forming operations are performed in such a manner that the metal is not completely constricted in its flow by the forming tools, with the result that the metal may escape the desired or intended shaping by buckling or folding. Of these operations, the following are industrially important:

"Sinking," where a tube is drawn through a die without support on its internal surface. Circumferential or hoop stresses may cause the tube to fold along its length if the wall of the tube is thin, Fig. 1 (*a*).

"Necking," "tapering," and "nosing," where a tube is pushed through a die without internal support. In this case the tube may fold along its length, Fig. 1 (*b*), but may also develop transverse folds under the action of longitudinal stresses, Fig. 1 (*c*).

"Stripping," where a shell or cup is removed from a draw punch. The cup is supported on its internal surface by the punch but not on its exterior surface. The longitudinal compressive stresses may therefore develop transverse folds, Fig. 1 (*d*).

"Cupping," or "deep drawing," where a blank is pushed by a punch through a die, forming a cup or shell. The sheet cannot always be supported sufficiently by holddown pads to prevent radial wrinkles or creases caused by compressive hoop stresses, Fig. 1 (*e*).

"Spinning," where a sheet is rotated in a lathe while forming tools work the sheet onto a forming block to produce a cup. Circumferential stresses again can develop radial wrinkles, Fig. 1 (*f*).

These folds and wrinkles appear to be specific cases of the general class of buckling failures and as such are characterized by two features: (*a*) they occur in "slender" articles, i.e., in thin metal, and (*b*) they are caused by compressive stresses lying in the plane of the sheet or the wall of the tube and acting in a direction normal to that in which the fold occurs.

The only one of the phenomena described which has been made the subject of theoretical considerations is the wrinkling of a deep-drawn cup (1).[3] However, this analysis has been applied principally in predicting the number of waves into which the flange would buckle rather than to the limiting conditions under which the cupping operations could be successfully conducted.

In the present work the factors involved in the folding or collapsing of tubes being sunk will be studied experimentally. The variables studied will be the alloy, its temper, the wall thickness of the tube, and the die contour.

Materials and Procedure

For the experimental investigation tubes of four different materials were used: (*a*) hard phosphorus-deoxidized copper (Rockwell 30-T, 59–66); (*b*) soft phosphorus-deoxidized copper (annealed 1 hr at 1000 F in a forced-convection furnace); (*c*) soft tube brass (66.5 per cent copper, 0.5 per cent lead, balance zinc, annealed 1 hr at 1000 F); and (*d*) soft aluminum (2S, annealed 15 min at 600 F).

The tubes were drawn commercially to different diameters ranging from 0.580 to 1.375 in., and possessed a wall thickness

[1] Professor of Physical Metallurgy, Case School of Applied Science, Cleveland, Ohio. Mem. A.S.M.E.

[2] Metallurgical Engineer, Chase Brass and Copper Company, Inc., Midwestern Division, Euclid, Ohio.

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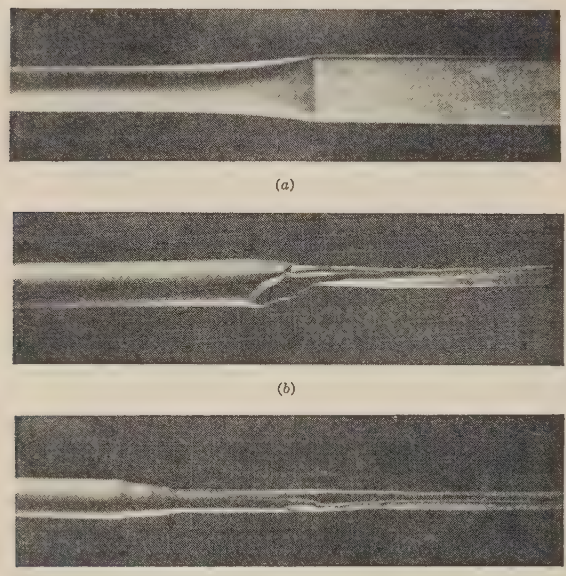
[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

varying between 0.009 to 0.050 in. In the case of the hard copper, the reduction in cross-sectional area after the last anneal (but before sinking) was approximately 50 per cent.

Tube specimens, approximately 40 in. long, were drawn through three different sets of conical steel dies, possessing half die angles of 7, 14, and 27 deg, respectively, to an outside diameter ranging from 0.375 to 1.300 in. The polish on the 7-deg die was inferior to that of the 14- and 27-deg dies.

Two types of points were put on the tubes, i.e., a folded point and a fold-free point. The first type of point was obtained by first flattening the tube, and then folding the flattened section under as shown in Fig. 2 (a and b). The fold-free point was obtained by sinking a tube progressively with a folded point in sufficiently small steps for a short length of the tube to eliminate the fold, Fig. 2 (c).



(a)

(b)

(c)

Fig. 2 Procedure of Pointing Tube

(a, Flattened tube end. b, Flattened section folded under. c, Tube drawn in a series of small reductions which eliminates folds and produces "fold-free" point.)

The sinking was performed on either an Olsen 60,000-lb, or a Baldwin Southwark 100,000-lb tensile-testing machine, at a speed of approximately 0.5 fpm. For this purpose a special die holder was attached above the upper jaw of the testing machine and the point of the tube was gripped and pulled with the regular lower jaw. A commercial soap-base lubricant with a high free-fat content (Nopco No. 6) was employed.

The load required to sink the tube was generally found to remain constant within ± 5 per cent over the entire length of each tube.

Micrometer measurements were made on both the initial and the drawn tube in order to compute the sinking stress s_1, and the reduction in mean diameter, D. The sinking stress was calculated from the formula

$$s_1 = \frac{P}{A'} = \frac{P}{\pi t' D'}$$

The significant strain for the strain-hardening of the metal in tube-sinking is the reduction in mean diameter for those cases where the wall thickness increases during the operation. This was the case for all tubes investigated since in no instance did the ratio of the wall thickness to mean diameter exceed 0.05, at which value, according to Baldwin and Howald (2), the wall thickness increases for reductions in outside diameter as high as 50 per cent.

This reduction in mean diameter R was computed by means of the equation

$$R = 1 - \frac{D'}{D}$$

The relative thickness of a tube is defined in the present work as the ratio of the wall thickness of the tube to the mean diameter of the tube, t^0/D^0.

If no restriction were put on the angle at which the tube entered the die, the tube would assume a cant, especially with the heavier drafts, when a fold was being perpetuated, Fig. 3. In general, the cant was toward the fold when the tube was sunk through wide-angle dies, whereas the tube assumed a cant away from the fold in the case of the narrow-angle die, although there were a few exceptions to this general rule. Experiments conducted with a restricting cage to force the tube into an axial position, however, showed that this factor had a negligible effect on the critical reductions that could be made.

Theory of Folding

The folding during tube-sinking and the closely related process of necking (primarily observed in cartridge cases) are, according to all external appearances, phenomena of plastic buckling, Figs. 12 to 14, inclusive. The stress responsible for folding is clearly the compressive hoop stress. This hoop stress varies for a fold-free tube along the length of the tube portion in the die, being determined for a given material primarily by the reduction from the original to the considered cross section.

If certain unknown conditions, regarding the magnitude and distribution of the hoop stresses required to draw a fold-free tube, exist the tube wall would be expected to collapse. It was therefore believed at the beginning of this investigation, that folding during tube-sinking might be closely related to the collapsing of a tube under external pressure. In any buckling process one of the decisive factors is the metal thickness; and the expected effect of wall thickness was also found to occur in folding, as discussed later. In the case of collapsing, increasing length of the portion exposed to pressure reduces the unit collapsing pressure (3, 4, 5, 6). The conclusion to be drawn from this fact, that a sunk tube drawn with an acute die (i.e., a long contact length) should collapse after a smaller reduction than a tube drawn through a large-angle die, has not been verified by the following experimentation.

In addition, it was found that no folds could be obtained within the limits of the investigated dimensions without the presence of a preformed fold. Therefore the phenomenon of folding probably is not related to a true buckling process, but comprises a new problem of plastic flow. The results of this investigation may help to reveal the nature of this problem, which, of course, must be recognized before any theoretical analysis can be attempted.

Experimental Results

Sinking of Tubes With a Fold-Free Point. As already mentioned, it was not found possible to collapse any tube with a fold-free point. The circular contour appears to be highly resistant to buckling, even in tubes with the thinnest wall and subjected to the highest possible reduction. An extreme example was soft-copper tube with $D^o = 1$ in. and $t^o = 0.009$ in. (i.e., a ratio $t^o/D^o = 0.009$) and a reduction in mean diameter of approximately 50 per cent.

(a) (b) (c)

Fig. 3 Tubes Showing a Cant When Sunk With Folds
(a, Tube in a 27-deg die. b, Tube removed from 7-deg die showing cant away from fold. c, Tube removed from 27-deg die, showing cant toward fold.)

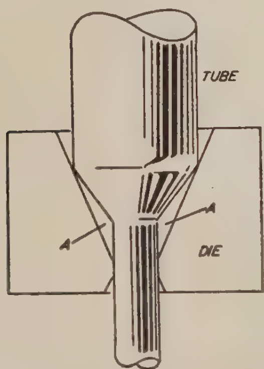


Fig. 4 Entrapment of Lubricant Between Tube and Die
(If the tube was prepared with a fold-free point whose angle was greater than that of the die, the lubricant could be entrapped in the cavities A-A. Being incompressible, it would force folds into the tube being sunk.)

Such fold-free tubes, however, are obtained only if the tubing is free from dents, and seams. In addition, it was observed that any lubricant entrapped between the die and tube surface caused folding, as illustrated schematically in Fig. 4.

The same facts were found also to exert a considerable influence in the necking of cartridge cases. A high percentage of rejections encountered in regular production was eliminated in one instance by avoiding any denting on handling the mouth of annealed cases, and in another case by designing the die contour such that no lubricant was entrapped in successive tapering operations.

Conditions for the Development of Folds in Sunk Tubing. Either one of two cases occurred when drawing a tube with a folded point: When subjected to small reductions in diameter, the resulting tube was fold-free, while beyond a certain reduction it folded along its entire length. Close to this limiting reduction the fold was eliminated only after the tube was drawn for a considerable length, occasionally as much as 12 in.

In the case of a tube which contained folds when sunk, the reduction in mean diameter possesses no physical significance. In order to evaluate the data, however, it is necessary to know what reduction could have been attained if folding had not occurred.

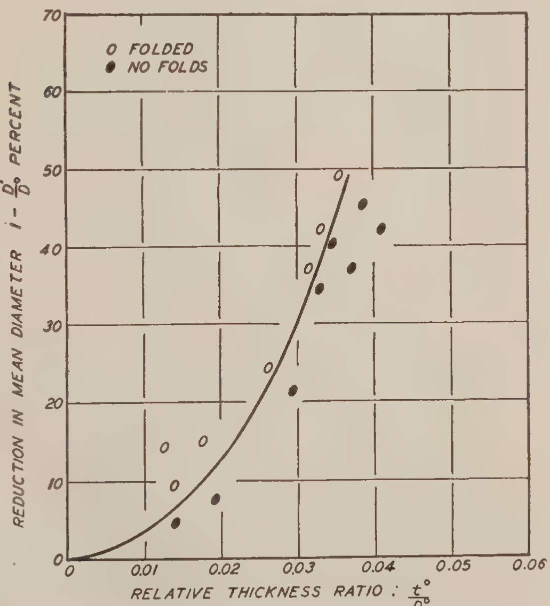


Fig. 5 Typical Set of Data on Folding
(For soft copper sunk through a polished steel die having a half-angle of 14 deg.)

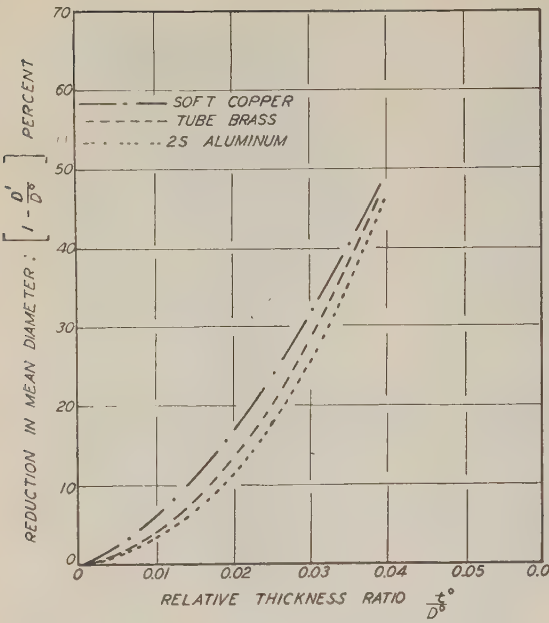


FIG. 6 LIMITING REDUCTION TO WHICH TUBES OF DIFFERENT METAL MAY BE SUNK USING FOLDED POINTS
(Data for 7-deg dies.)

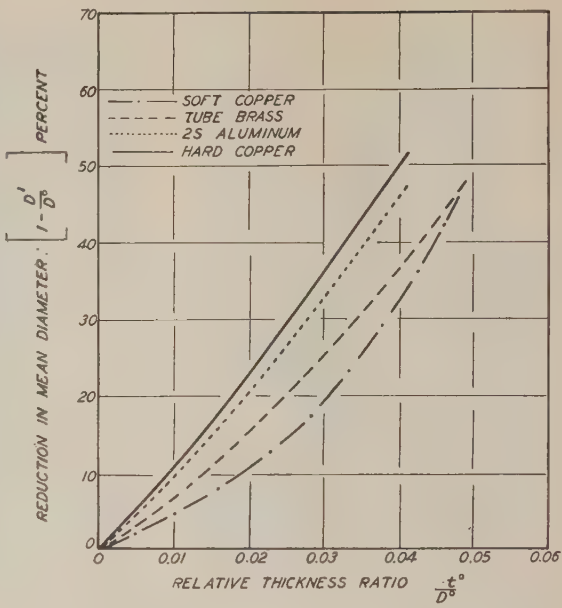


FIG. 8 LIMITING REDUCTION TO WHICH TUBES OF DIFFERENT METALS MAY BE SUNK USING FOLDED POINTS
(Data for 25-deg dies.)

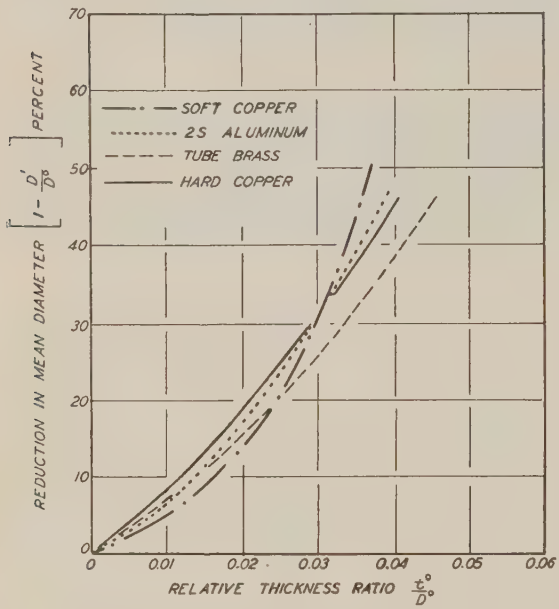


FIG. 7 LIMITING REDUCTION TO WHICH TUBES OF DIFFERENT METALS MAY BE SUNK USING FOLDED POINTS
(Data for 14-deg dies.)

This intended reduction was obtained by interrupting the sinking process, which produced folds; preparing the tube with a fold-free point; and then resuming the sinking process without folding. The (intended) reduction was then taken as that of the fold-free tube.

A typical set of experimental data is represented in Fig. 5. It assembles the results obtained on tubes of varying wall thickness of a particular alloy and drawn through a particular die. A solid line has been drawn separating the conditions where a fold was perpetuated in the sinking process from those conditions where folds are eliminated. This graph illustrates the degree of accuracy in determining the dividing line between folded and fold-free tubing. In general, the position of this line appears to be accurate within approximately ±3 per cent reduction.

In Figs. 6 to 8, inclusive, all the experimentally determined boundary lines are shown for the three different dies, the various curves in each graph belonging to the different alloys. These graphs illustrate that the reduction in mean diameter, up to which a tube can be sunk fold-free depends primarily upon the relative thickness of the tube. In general, it increases slightly faster than proportional to the reduction, for all conditions investigated. Any one of the investigated alloys could be sunk with any one die to the commercial reduction limit of approximately 50 per cent, if the wall thickness is larger than 5 per cent of the outside diameter. On the contrary, no tube investigated possessing a wall thickness of approximately 1 per cent of its diameter could be drawn fold-free (with a folded point) by more than approximately 10 per cent reduction.

This relation was found to be comparatively independent of either the alloy, according to Figs. 6 to 8, inclusive, or the die angle, according to Figs. 9 to 11, inclusive. A possible effect of the alloy, exceeding the accuracy limits of experimentation, was observed only for the wide-angle 27-deg die. Contrary to expectations for a buckling phenomenon, hard copper could be sunk fold-free to higher reductions than soft copper.

Also, regarding any effect of die angle, it could have been expected that a large-angle die, because of its length of contact, would permit larger reductions than an acute die. Numerous investigations on the collapse of tubes when subjected to external pressure reveal a profound effect of specimen length on collapse pressures (3, 4, 5, 6). It would normally be expected that this effect would apply to tube-sinking if the fold development was a phenomenon of buckling.

Observations on Fold Development. This anomalous behavior

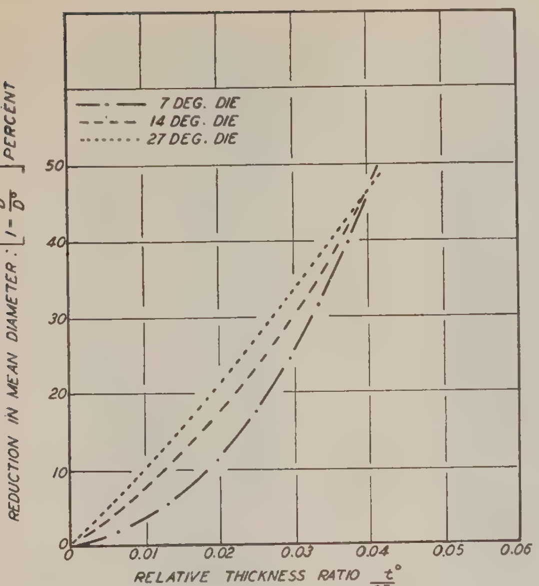


Fig. 9 Limiting Reduction to Which Soft 2S Aluminum May Be Sunk Using Folded Points

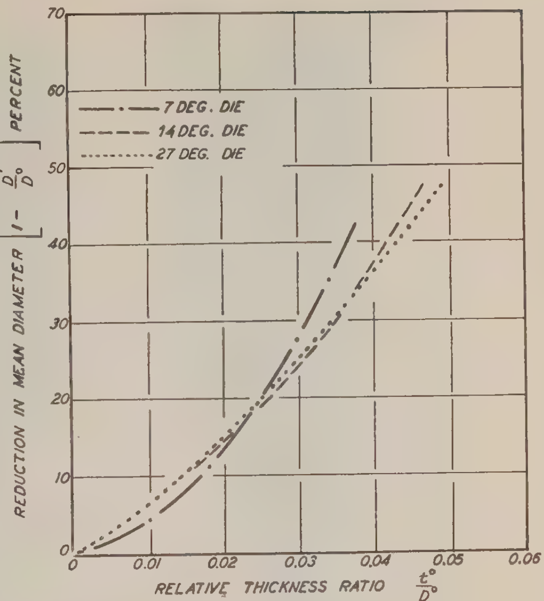


Fig. 10 Limiting Reductions to Which Soft Tube Brass May Be Sunk Using Folded Points

is more readily understood, however, if the geometry of the fold is studied. The manner in which the fold is perpetuated along the length of the tube is illustrated in Figs. 12 to 14, inclusive. The point of inception of the fold is shallow, but as the metal proceeds through the die, the fold deepens generally, first becoming broader and then narrower. When the metal leaves the die the edges of the fold are brought together. This cycle occurs for all reductions, heavy or light. The length of the fold depends primarily upon two factors, i.e., the intended reduction Fig. 12, and the relative thickness, Fig. 13. It is, however, practically independent of the die angle, other factors being constant, Fig. 14. Consequently, the point of inception of the fold may be either inside or outside of the die. It appears interesting that the die contour, and consequently the length along which the forces act, which cause the reduction in diameter, has practically no effect on the fold development.

The length of the fold was measured for a number of soft copper tubes drawn through the 27-deg die, Fig. 15. The critical reductions for the various relative thickness ratios have been also entered in Fig. 15, as arrows. It is seen that the length of fold at the critical point is roughly equal to the diameter of the initial outside diameter.

In fold-free sinking the reduction in diameter causes the length of the tube to increase, while the thickness remains practically constant, as mentioned previously. If the formation of a fold were merely a matter of the tube wall being folded into a lobe as the tube was drawn through the die, little change in the cross-sectional area of the original tube would be anticipated. The reduction in diameter would then cause primarily a bending in the area containing the fold, while the remaining portion of the cross section would be only slightly curved. Such a process would result in no change in length of the tube. Some measurements, however, reveal that during the sinking of a folded tube the cross-sectional area is definitely reduced and that, consequently, the entire cross section is subjected to plastic flow.

To determine any increase in length, 2-in. intervals were scribed along the length of tubes. The tubes were sunk with folds, and the segments were then measured. From the assumption that the volume of the metal remains constant during plastic deformation, the ratio of the initial length to the final length of a tube section yields the reduction in cross-sectional area affected by the process

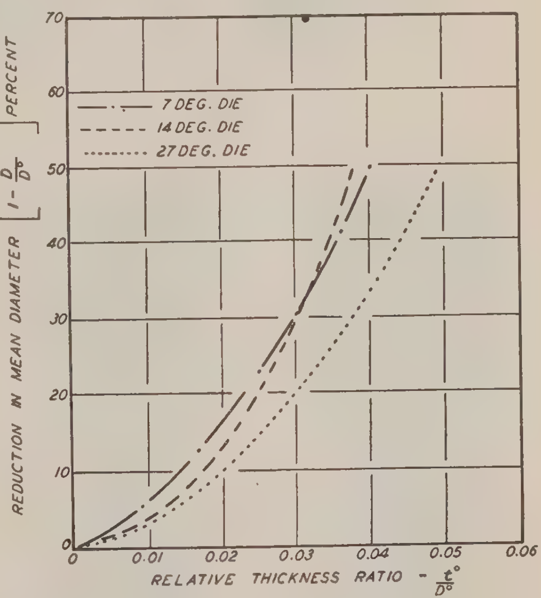


Fig. 11 Limiting Reductions to Which Soft Copper May Be Sunk Using Folded Points

$$R' = 1 - \frac{L^\circ}{L'}$$

In Fig. 16 the reduction in area determined in this manner is plotted as a function of the intended reduction (in mean diame-

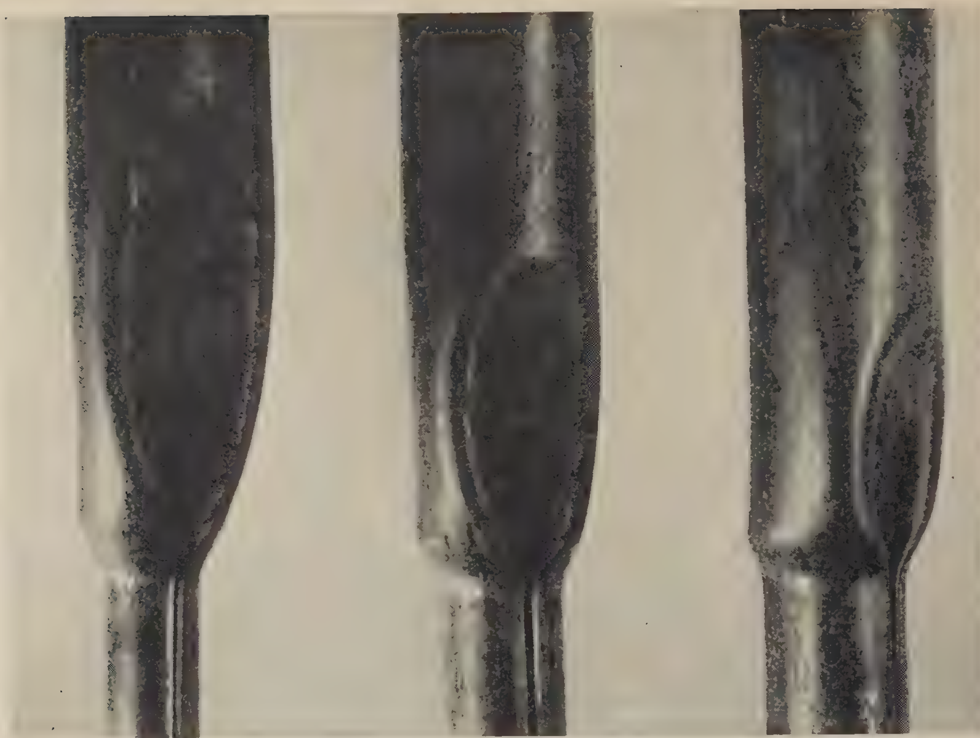


Fig. 12 Length of Fold During Formative Period as a Function of Intended Reduction in Mean Diameter

(All tubes were 1 in. × 0.020 in. soft copper [$t^o/D^o = 0.021$] sunk through 27-deg dies. Left to right: sunk to 0.500 in. diam, sunk to 0.650 in. diam, sunk to 0.750 in. diam, respectively. Slightly larger than natural size.)

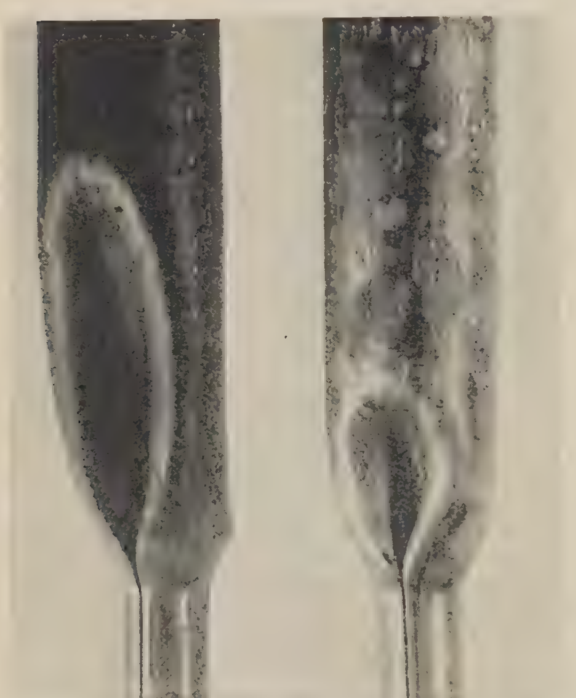


Fig. 13 Length of Fold During Formative Period as a Function of Relative Thickness

(Left-hand tube is a 1-in. × 0.020-in. soft copper tube [t^o/D^o — 0.021] sunk to 0.500 in., showing relatively long fold. Right-hand tube is a 1-in. × 0.038-in. soft copper tube [t^o/D^o — 0.040] sunk to 0.500 in. showing relatively short fold. Both tubes sunk through a 27-deg die. Slightly larger than natural size.)

Fig. 14 Length of Fold During Formative Period as a Function of Die Angle

(Both tubes were 1 in. × 0.020 in. soft copper [t^o/D^o — 0.021], sunk to 0.500 in. diam. Left-hand tube sunk through 27-deg die. Right-hand tube sunk through a 7-deg die. Slightly larger than natural size.)

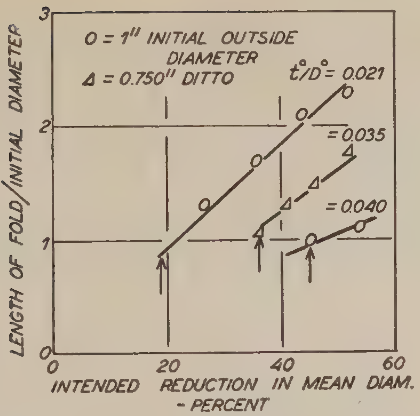


FIG. 15 GRAPH SHOWING LENGTH OF FOLD AS A FUNCTION OF INTENDED REDUCTION AND RELATIVE THICKNESS RATIO

(Data are for soft copper through 27-deg die. Critical reductions are indicated by arrows.)

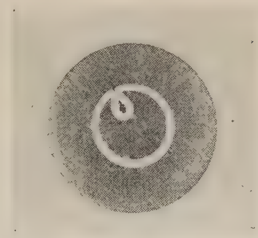


FIG. 17 CROSS SECTION OF TUBE SUNK WITH FOLD

(Tube wall is variable in thickness, being thickest at point diametrically opposite the fold.)

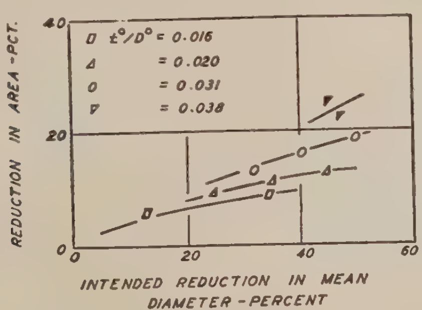


FIG. 16 REDUCTION IN CROSS-SECTIONAL AREA OF TUBES SUNK WITH FOLDS

(Data are for soft tube brass sunk through a 14-deg die.)

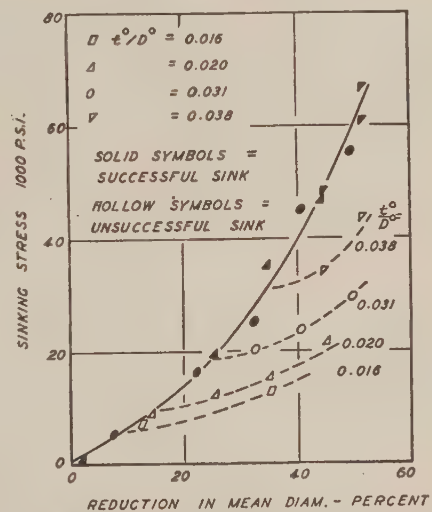
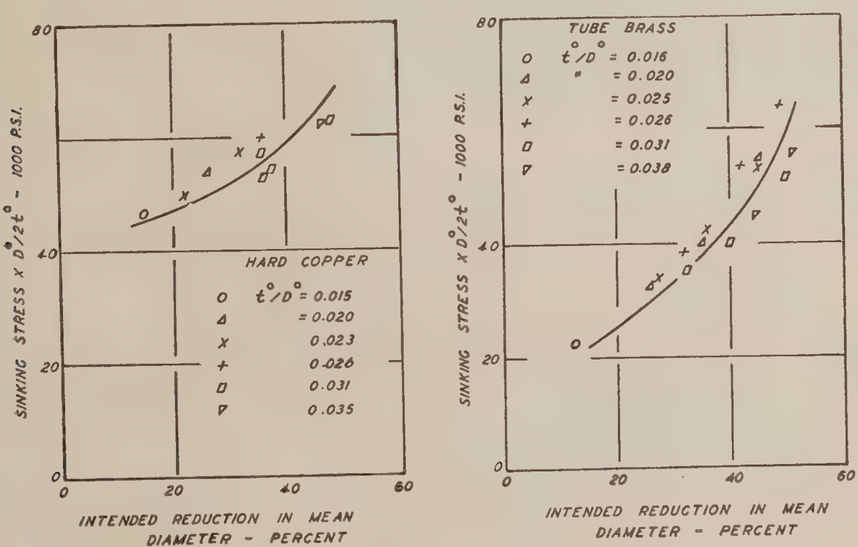


FIG. 18 STRESSES REQUIRED TO SINK TUBE WITH FOLDS

(Data are for soft tube brass sunk through a 14-deg die.)



(a) (b)

FIG. 19 STRESS REQUIRED TO SINK TUBE WITH FOLDS DIVIDED BY RELATIVE THICKNESS RATIO

(a, Data for hard copper and 14-deg die. b, Data for soft tube brass and 14-deg die.)

ter). It is apparent that an increase in both the intended reduction and the relative thickness ratio of the tube leads to an increase in the actual reduction in area.

Thickness measurements revealed that no thickness changes occur before the folded tube enters the die. Within the die, however, the wall thickness increases. As shown in Fig. 17 this deformation is not uniform around the circumference as the wall thickens to the greatest extent at a point diametrically opposite to the fold.

Stresses to Sink Folded Tubes. The unit stress s_1, required to pull the folded tube through the die is determined by the equation

$$s'_1 = \frac{P}{A'} = \frac{P}{A^\circ}\frac{L'}{L^\circ}$$

In Fig. 18 the stresses required to sink (with folds) brass tubes of various initial relative thickness ratios, through a 14-deg die, are plotted as a function of the intended reduction. It is to be noted that the curves so obtained intersect the curve for sinking without folds, reported previously (7), at the critical reduction, for the relative thickness ratio in question.

The curves for various relative thickness ratios can be further reduced to a single master curve if the values for the individual stresses are divided by the relative thickness ratio. This has been carried out in Figs. 19 and 20 for tube brass and hard copper, respectively.

Conclusions

The results of this investigation cannot be reconciled with the conventional theory of buckling or collapsing. The factors which determine the perpetuation or elimination, respectively, of a fold in tube-sinking are different from those which determine definite buckling phenomena such as the collapsing of a tube.

The only factor which is conducive to both collapsing and fold formation is a small thickness-to-diameter ratio. Other factors, however, such as the length along which pressure is applied and the elastic and plastic metal properties which are important in collapsing, have only a minor influence on folding.

The predominant effect of the reduction in diameter cannot be satisfactorily explained. However, it has been observed that a fold is enlarged only if the circumferential force is smaller for this process than for fold-free sinking, assuming that the differences in the respective sinking forces also indicate proportional differences in the circumferential forces, other conditions being equal. This would also explain the observation that if the sinking forces for folded tubes equal those for fold-free tubes, the tendency to fold disappears. In other words, a fold will be perpetuated only if the resistance against bending into the fold is smaller than the resistance against a uniform compression of the diameter. No theoretical approach to this phenomenon has been developed as yet.

It has been also observed that it is much more difficult to obtain a fold on sinking a circular cross section than on sinking a section with a preformed fold. This result may be compared with the buckling forces required to deform permanently bars under concentric and eccentric loading, respectively.

Acknowledgment

Acknowledgment is made to Prof. K. H. Donaldson, head of the Metallurgical Engineering Department of Case School of Applied Science, and to Dr. H. P. Croft, director of technical control and research, Midwestern Division, Chase Brass and Copper Company, Inc., for their encouragement and help. The authors are also indebted to B. H. Higgins, Linderme Tube Company, Cleveland, Ohio, who kindly supplied the aluminum tube used in this research.

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Stress Analysis of Tube-Sinking

By GEORGE SACHS,[1] AND W. M. BALDWIN, JR.[2]

A theoretical stress analysis was made of the tube-sinking operation, equations being set up to yield the sinking stress as a function of the reduction in mean diameter. The equations were confirmed by experimentation on annealed and full hard copper and on annealed tube brass for conical dies of various tapers. The coefficients of friction existing during tube-sinking have been determined and are found to be of the same magnitude as those determined previously for wire drawing.

Nomenclature

The following nomenclature is used in the paper:

A = cross-sectional area of tube

B = parameter $= \dfrac{\tan\alpha + f}{\tan\alpha}$

D = mean diameter of tube

f = friction coefficient

k = flow stress (or yield strength in tension)

k_0 = $1.1\,k$

P = force required to sink tube

s_1 = algebraically largest principal stress, assumed to be average longitudinal tension in sinking

s_2 = intermediate principal stress, assumed to be die pressure

s_3 = algebraically smallest principal stress, assumed to be average hoop stress

t = wall thickness of tube

x = longitudinal co-ordinate

α = angle between die wall and tube axis = half die angle of conical die

NOTE: o as a superscript refers to original dimensions. Primed symbols refer to final dimensions.

Introduction

The analysis of the stresses occurring in a number of forming processes, in which the metal is continuously subjected to deformation, follows in each case a similar pattern. The processes to which such an analysis has been applied so far, wire-drawing (1, 2),[3] strip-rolling (3, 4, 5, 6, 7), and tube-drawing with a moving mandrel (8) have one feature in common, namely, that the mean stresses acting on each cross section can be derived from the equilibrium of the longitudinal forces.

An analysis of tube-sinking has not been attempted previously. In a paper dealing with the stress and strain relations in tube-sinking, it was assumed that the stresses in this process are identical with those occurring in wire-drawing (9).

The following discussion presents a theoretical and experimental analysis of the tube-sinking process. In order to develop a differential equation for the stresses occurring in the process, it was found necessary to consider not only the equilibrium of the longitudinal but also that of the transverse forces. This results from the fact that in contrast with the processes just mentioned the largest and smallest principal stresses in tube-sinking do not occur in a longitudinal plane but in a circumferential plane.

In the experimental analysis, the high tendency of the tube being sunk to develop folds was eliminated by a special pointing procedure. This yielded a fold-free point, which permitted sinking operations with tubes with the smallest available wall thickness to maximum reductions without folding, and thus to measure the sinking forces over a wide range of conditions. The problem of folding will be considered in a subsequent publication.

Materials and Procedure

For the experimental investigation, tubes of three different materials were used: (*a*) hard phosphorus-deoxidized copper (Rockwell 30-T, 59–66); (*b*) soft phosphorus-deoxidized copper (annealed 1 hr at 1000 F in a forced-convection furnace); and (*c*) soft tube brass (66.5 per cent copper, 0.5 per cent lead, balance zinc, annealed 1 hr at 1000 F). The tubes were drawn commercially to different diameters, ranging from 0.580 to 1.375 in., and possessed a wall thickness of 0.015 to 0.050 in. In the case of the hard copper, the reduction in cross-sectional area after the last anneal (but before sinking) was approximately 50 per cent.

Tube specimens, approximately 40 in. long, were drawn through three different steel dies, possessing half die angles of 7, 14, and 27 deg, respectively, to an outside diameter of 0.50 in. The polish on the 7-deg die was inferior to that of the 14- and 27-deg dies.

The sinking was performed on either an Olsen 60,000-lb, or a Baldwin Southwark 100,000-lb tensile-testing machine, at a speed of approximately 1/2 fpm. For this purpose, a special die holder was attached above the upper jaw of the testing machine, and the point of the tube was gripped and pulled with the regular lower jaw. A commercial soap-base lubricant with a high free-fat content (Nopco No. 6) was employed.

The load required to sink the tube was generally found to remain constant within ±5 per cent over the entire length of each tube.

Micrometer measurements were made on both the initial and the drawn tube in order to compute the sinking stress s_1', and the reduction in mean diameter. The sinking stress was calculated from the formula

$$s_1' = \frac{P}{A'} = \frac{P}{\pi t' D'}$$

The significant strain for the strain-hardening of the metal in tube-sinking is the reduction in mean diameter for those cases where the wall thickness increases during the operation. This was the case for all tubes investigated, since in no instance did the ratio of the wall thickness to mean diameter exceed 0.05, at which value, according to Baldwin and Howald (10), the wall thickness increases for reductions in outside diameter as high as 50 per cent.

The reduction in mean diameter R was computed by means of the equation

$$R = 1 - \frac{D'}{D^o}$$

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[1] Professor of Physical Metallurgy, Case School of Applied Science, Cleveland, Ohio. Mem. A.S.M.E.

[2] Metallurgical Engineer, Chase Brass and Copper Company, Inc., Midwestern Division, Euclid, Ohio.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Metals Engineering Division and presented at the Spring Meeting, Chattanooga, Tenn., April 1–3, 1946, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Stress-strain curves by means of tensile tests were determined for the various metals before and after sinking in the conventional manner. The reduction in area is the significant strain in the tensile test, being equivalent to the reduction in mean diameter in tube-sinking. The significant strain of a sunk tube subjected to a tensile test is therefore the sum of the two strains referred to in the initial cross-sectional area.

Theoretical Analysis

The method of analysis used here assumes (*a*) that a pressure normal to the working tool-metal interface operates on the interface of tube and die, and that, neglecting the presence of the shear stress, this pressure is one of the principal stresses; (*b*) that because of friction, a shear stress operates at the interface; (*c*) that transverse sections are free of shear stresses; (*d*) that the normal stress acting on the transverse sections is uniformly distributed over the cross section and that it is another principal stress; (*e*) that the wall thickness of the tube is small in comparison to the tube diameter; and (*f*) that the wall thickness of the tube remains constant through the process. This last assumption is not strictly valid but has been found to be true within ±5 per cent (9, 10).

With these assumptions, two equations can be derived from Fig. 1, using the equilibrium of the forces in the longitudinal and radial directions. Summation of the longitudinal forces supplies one equation

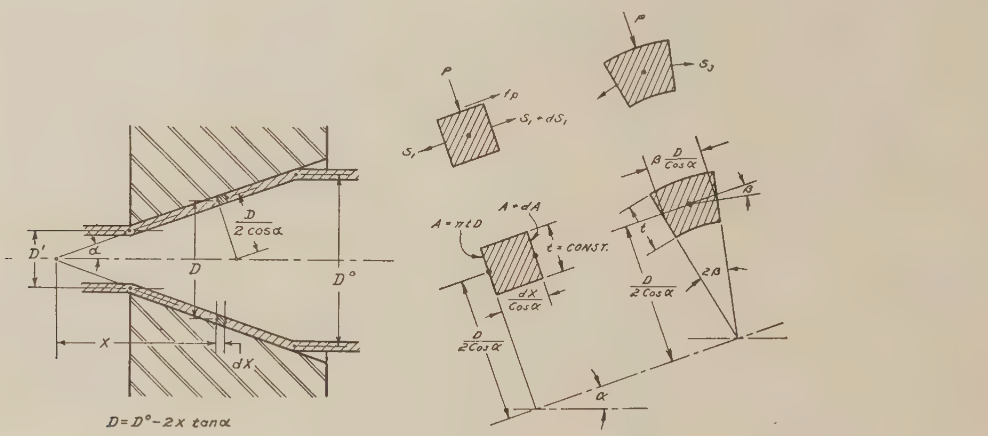

Fig. 1 Stresses in Tube-Sinking

$$\cos\alpha[(s_1 + ds_1)(A + dA) - s_1A] + p\sin\alpha\,\frac{dx}{\cos\alpha}\,\pi D + fp\cos\alpha\,\frac{dx}{\cos\alpha}\,\pi D$$

$$= \cos\alpha\, d(s_1A) + pdx\pi D\,(\tan\alpha + f) = 0 \ldots\ldots\ldots\ldots [1]$$

Using the geometrical relations, Fig. 1

$$D = 2x\tan\alpha \qquad dD = -2dx\tan\alpha \ldots\ldots\ldots\ldots [2]$$

$$A = \pi tD \quad dA = \pi tdD = 2\,\pi t\,dx\tan\alpha \ldots\ldots\ldots\ldots [3]$$

transforms Equation [1] to

$$\pi td(s_1D)\cos\alpha\,\pi DdD\,p\,\frac{\tan\alpha + f}{2\tan\alpha}$$

$$= \pi td(s_1D)\cos\alpha\,{}^1/_2\,\pi DdD\,pB = 0 \ldots\ldots\ldots\ldots [4]$$

where

$$B = \frac{\tan\alpha + f}{\tan\alpha} \ldots\ldots\ldots\ldots [5]$$

Summation of the radial components of forces yields

$$p2B\,\frac{D}{2\cos\alpha}\,\frac{dx}{\cos\alpha} + 2s_3Bt\,\frac{dx}{\cos\alpha} = 0$$

or

$$p = -s_3\,\frac{2t\cos\alpha}{D} \ldots\ldots\ldots\ldots [6]$$

Substituting Equation [6] into Equation [4] results in the general differential equation of tube-sinking

$$d(s_1D) = Bs_3dD \ldots\ldots\ldots\ldots [7]$$

Equation [7] is of the same form as that derived for rolling (3, 4, 5, 6, 7), wire-drawing (1, 2), or tube-drawing (8), the only difference being that the parameter B takes on different values for the various forming operations.

To relate the quantities s_1 and s_3 in Equation [7], some condition of plasticity must be invoked. The most accurate available is the strain-energy condition

$$(s_1 - s_2)^2 + (s_2 - s_3)^2 + (s_1 - s_3)^2 = 2k^2 \ldots\ldots\ldots [8]$$

where the quantities s_1, s_2, and s_3, are the three principal stresses in algebraically decreasing order of magnitude and k is the flow stress or yield strength of the metal as determined in simple tension or compression. However, the use of this condition involves considerable calculations. Under most general conditions Equation [8] sets certain limits on the values that $s_1 - s_3$ can assume, namely

$$k \leqq (s_1 - s_3) \leqq 1.15\,k \ldots\ldots\ldots\ldots [8a]$$

For the present case a constant average value may be taken as sufficiently accurate to represent the condition of plasticity throughout the sinking process, namely

$$s_1 - s_3 = 1.1\,k = k_0 \ldots\ldots\ldots\ldots [8b]$$

Substitution of Equation [8*b*] in Equation [7] and making the variable D dimensionless by multiplying the equation with D^o finally yields the linear differential equation

$$\frac{ds_1}{d(D/D^o)} + (1 - B)\frac{s_1}{D/D^o} = -B\frac{k_0}{D/D^o}$$

If the parameter B, is considered constant throughout the sinking process (i.e., if the die angle and coefficient of friction are constant) this equation may be integrated as follows

$$s_1 = -B\left(\frac{D}{D^o}\right)^{B-1}\int_1^o k_0\frac{d(D/D^o)}{(D/D^o)^B} + s_1^o\left(\frac{D}{D^o}\right)^{B-1} \quad \ldots\ldots [9]$$

where s_1^o is the stress for zero reduction, $D = D^o$.

The parameter B passes to unity as the coefficient of friction goes to zero. If the metal is further assumed to possess no strain-hardening (i.e., k_0 is constant) then Equation [9] becomes, neglecting s_1^o

$$s_1 = k_0 \log_e \frac{D^o}{D} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9a]$$

This equation is commonly associated with ideal (i.e., frictionless) conditions, and has been deduced, for example, for wire-drawing, not only by using a method analogous to the present one (1, 2), but also by using stress-analysis methods based on the balance of the work consumed (11).

Another special case of the general Equation [9] is that in which friction exists but no work-hardening occurs. In this case k_0 is independent of D; and Equation [9] may be integrated to yield the following equation

$$s_1 = \frac{Bk_0}{B-1}\left[1 - \left(\frac{D}{D^o}\right)^{B-1}\right] + s_1^o\left(\frac{D}{D^o}\right)^{B-1} \quad \ldots\ldots [9b]$$

This equation yields the longitudinal stress at any position D, for a given D^o, and consequently also the sinking stress, $s_1 = s_1'$, at the exit, $D = D'$

$$s_1' = \frac{Bk_0}{B-1}\left[1 - \left(\frac{D'}{D^o}\right)^{B-1}\right] + s_1^o\left(\frac{D'}{D^o}\right)^{B-1} \quad \ldots [10]$$

Experimental Results

Hard Copper. The comparison of experimental data with the theory may be first made for hard copper. This material was selected for the investigation primarily because it represents a practical example of the conditions assumed to develop Equation [9b], namely, a material which does not strain-harden during the sinking process. In Fig. 2 the yield strength k, of the hard copper is shown as a function of the (mean) diameter change, after sinking through the 14-deg (half-angle) die. It will be noticed that an average value of k = 60,000 psi would describe the yield strength over the entire range of reduction, within ±3 per cent. A family of curves for the sinking stress s_1, obtained by substituting the value $k_0 = 1.1\ k$ = 66,000 psi, and a series of values for the parameter B, in Equation [9b] are also represented in Fig. 2.

The value of the integration constant C was determined experimentally by sinking a tube with the negligibly small reduction of 0.25 per cent, approximately; the measured value of 3000 psi agrees well with that extrapolated from the experimental sinking stresses for larger reductions.[4]

[4] Abundant experimental evidence may be found in the literature where a wire-drawing stress extrapolates to a definite positive stress value for zero reduction, see Thompson (12), Lueg and Pomp (13), Davis and Dokos (2). In discussing this positive value, Thompson describes it as "the extra pull required merely to overcome the frictional resistance in the die." Poeschl (14) and Davis and Dokos (2) attribute this positive value to a small transitional region of elasticity. Poeschl has attempted a mathematical analysis for this elastic region. In the present paper such positive values for soft copper or tube brass were found to be negligible whereas for hard copper they were found to be considerably higher, thus tending to confirm Poeschl's theory.

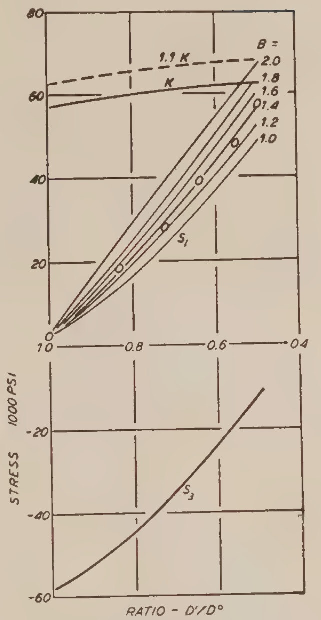

Fig. 2 Stress-Strain Diagram and Sinking-Stress Data for Hard Copper

(Experimental sinking-stress data for a 14-deg die half-angle have been entered as circular symbols.)

The experimental data agree rather closely with the theoretical curve for which the parameter B has a value of 1.4, approximately.

The hoop stress, $s_3 = s_1 - k_0$ Equation [8b] is also shown in Fig. 2, for a value of B = 1.4. The curve for the hoop stress represents its distribution over any length of contact, as its value is determined only by the local reduction of the considered cross section, irrespective of the total reduction of any particular draw.

Annealed Copper and Brass. For any soft material such as annealed copper and brass, the yield strength is primarily a function of the reduction. The stress-strain curves for these two metals derived from tensile tests are shown in Figs. 3 and 4.

The theoretical sinking stress can be obtained by graphical integration of Equation [9]. In Figs. 5 and 6, families of curves for the sinking stress, for a series of values for the parameter B, are given for soft copper and brass, respectively.

In comparing these theoretically derived curves with experimental data, it should be borne in mind that the stress-strain curve for a tube which had been sunk differs from that derived from a soft tube, as illustrated for one example in Fig. 7. This factor can be taken into account by a method employed by Sachs (15) for wire-drawing. From a comparison of the stress-strain curves in tension with those of sunk tubes as given in Fig. 7, a chart giving the effective reduction as a function of the actual reduction can be constructed.[5] Such a chart is shown in Fig. 8 for tubes sunk through the three different die angles included in this research. It will be noted that for large reductions the ideal and effective reductions become approximately equal. It is for this reason that this correction was not required for hard copper.

Figs. 9 to 13, inclusive, show the theoretical sinking stresses

[5] The difference between effective and actual reduction is attributed to the presence of shear stresses and shear strains. These have been demonstrated by means of grids on the cross section of wire, in both soft and hard copper, by Taylor and Quinney (16), and by Siebel (17).

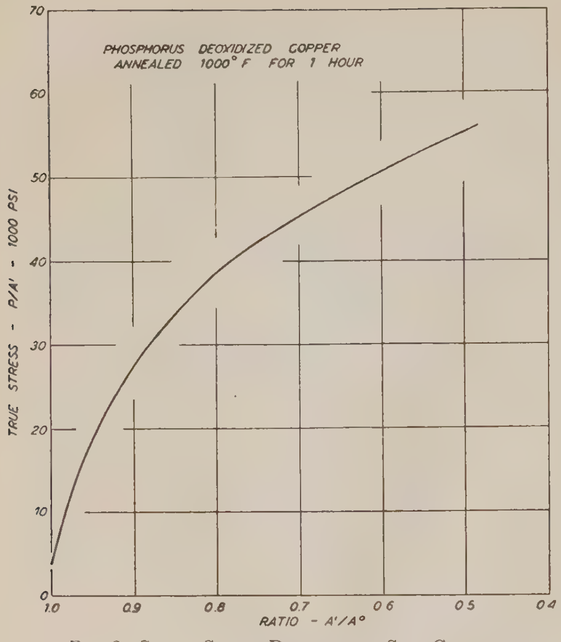

Fig. 3 Stress-Strain Diagram for Soft Copper

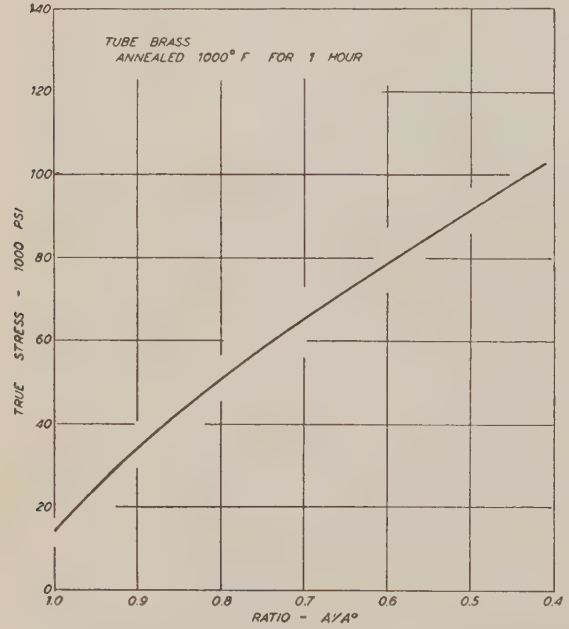

Fig. 4 Stress-Strain Diagram for Soft Brass

transformed by means of Fig. 8 from functions of the actual reductions, Figs. 5 and 6, into functions of the effective reductions. Each graph now applies only for a given alloy and a given die angle; and the range of parameter values B, represented in each graph covers the range of experimental scattering.

The experimental sinking stresses are shown in Figs. 9 to 13, inclusive, as open circles. It will be noticed that the scatter is greatest in the case of the 7-deg die and progressively lessens in

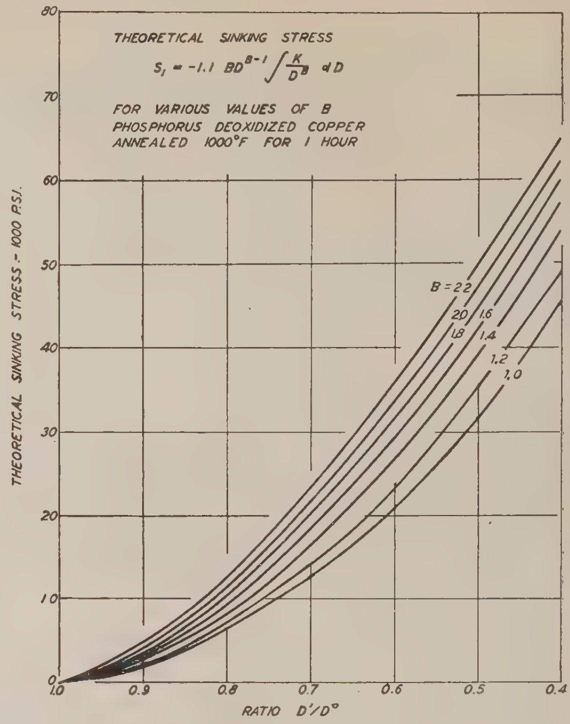

Fig. 5 Theoretical Sinking-Stress Curves for Soft Copper for Various Values of Parameter B

the case of the 14- and 27-deg die. These observations indicate that some variation in friction was encountered in the experiments.

In general, however, there is a fair agreement between the experimental value and a theoretical curve for a particular value of the parameter, B.

Coefficients of Friction. From the parameter B, and the die angle α, the frictional coefficient $f = \tan \rho$, can be readily determined by means of Equation [5]

$$\frac{\tan \alpha + f}{\tan \alpha} = B$$

In Fig 14 the parameter B is plotted as a function of the coefficient of friction, $\tan \rho$, for the three die angles, $\alpha = 7$, 14, and 27 deg employed in this investigation.

In Table 1 the parameters B are tabulated for the various metals and die angles investigated. The corresponding values of the coefficient of friction, f, are also included in Table 1.

TABLE 1 PARAMETERS, DIE ANGLES, AND COEFFICIENTS OF FRICTION OF METALS INVESTIGATED

Material	Die angle, deg	Parameter, B	Coefficient of friction, f
Hard copper.........	14	1.4	0.10
Soft copper..........	14	1.4	0.10
	27	1.3	0.15
Tube brass...........	7	2.6	0.20
	14	1.5	0.13
	27	1.3	0.15

These coefficients of friction are for both the 14- and 27-deg die of the same order of magnitude as those obtained by other investigators for wire-drawing (1, 2). However, for the 7-deg die, which was not as finely polished as the 14- or 27-deg die, the friction was found to be approximately twice as great.

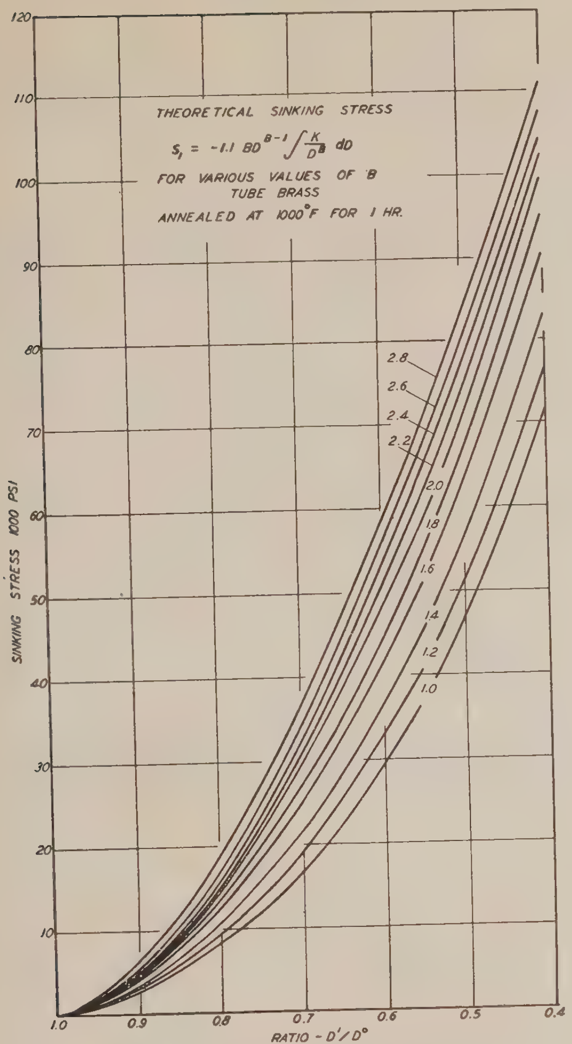

FIG. 6 THEORETICAL SINKING-STRESS CURVES FOR SOFT BRASS OF VARIOUS VALUES OF PARAMETER B

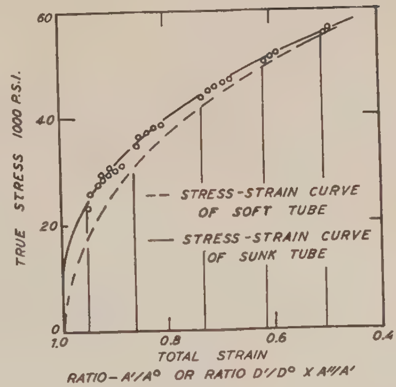

FIG. 7 COMPARISON OF STRESS-STRAIN CURVE OBTAINED FROM SOFT COPPER AND FROM COPPER SUNK THROUGH 14-DEG DIE

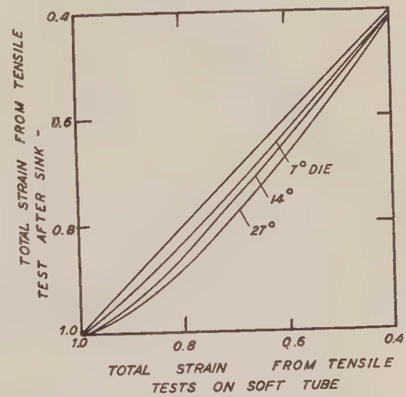

FIG. 8 CHART CORRELATING REDUCTION IN AREA FOR GIVEN STRESS ON STRESS-STRAIN DIAGRAMS OF SOFT TUBES AND SUNK TUBES

CONCLUSIONS

Any theory of a metal-forming process involves a number of simplifying assumptions in order to obtain sufficiently simple equations to allow numerical or graphical evaluation. Some discrepancies between theory and experimentation may therefore be expected. The analysis of tube-sinking presented here contains some minor refinements as compared with previous analyses of related processes in that no mathematical approximations are used other than that employed for the condition of plasticity. The experimentation indicates that this has yielded the gratifying result that the sinking stresses (i.e., the draw forces and the power consumption, excluding the losses in the equipment) can be calculated satisfactorily to within ± 2000 psi for soft copper drawn with a 27-deg die, and to within ± 9000 psi for soft tube brass drawn with a 7-deg die. This range of agreement is comparable to that reported in studies on wire-drawing, tube-drawing, etc.

It must be emphasized, however, that such an agreement cannot be claimed for either the distributions of the stresses or the strains over the cross section in drawing processes. So far, no theoretical analysis is available which promises to yield accurate results regarding the stress and strain distributions. This knowledge would be very desirable as the stress distribution is important in respect to various commercial problems, particularly those involving residual stresses.

Appendix

COMPARISON OF TUBE-SINKING WITH WIRE-DRAWING; EFFECTS OF TUBE THICKNESS

The experimental data on tube-sinking were obtained with thin-walled tubes, their ratio of wall thickness to diameter (relative thickness) being between $t_0/D^o = 0.01$ to 0.05. Tubes having a larger relative thickness were not investigated.

However, some indication regarding the effect of relative thickness can be obtained by comparing the stresses required to draw, respectively, thin-walled tube with those required to draw wire (or rod), other conditions (metal, dies, lubricant) being identical.

The wire represents the limiting condition of a tube, the relative wall thickness being $t_0/D^o = 0.5$. The wire-draw stress is given, according to Sachs (1) by the equation

$$s_1'' = \frac{B_1 k}{B_1 - 1}\left[1 - \left(\frac{D'}{D^o}\right)^{2(B_1-1)}\right] \qquad [11]$$

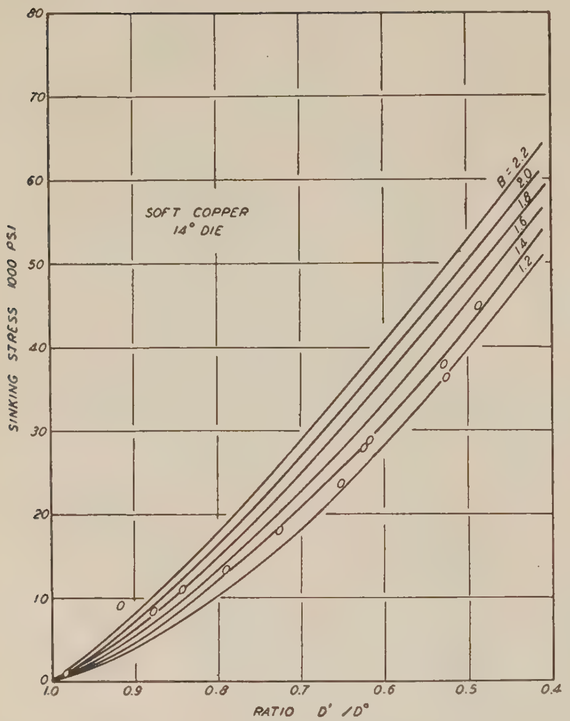

FIG. 9 THEORETICAL SINKING-STRESS CURVES CORRECTED FOR DIFFERENCES IN STRESS-STRAIN DIAGRAMS SHOWN IN FIG. 7

(Curves are for soft copper sunk through a 14-deg die. Experimental data are represented as circles.)

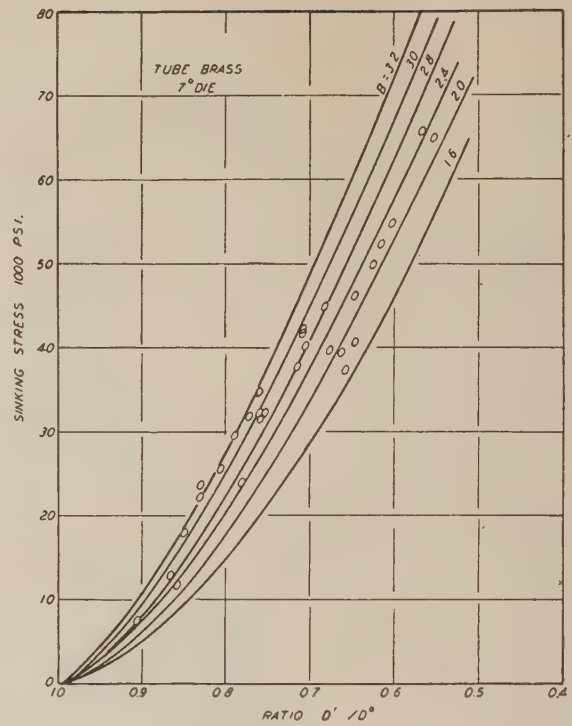

FIG. 11 THEORETICAL SINKING-STRESS CURVES CORRECTED FOR DIFFERENCES IN STRESS-STRAIN DIAGRAMS SHOWN IN FIG. 7

(Curves are for soft tube brass sunk through a 7-deg die. Experimental data are represented as circles.)

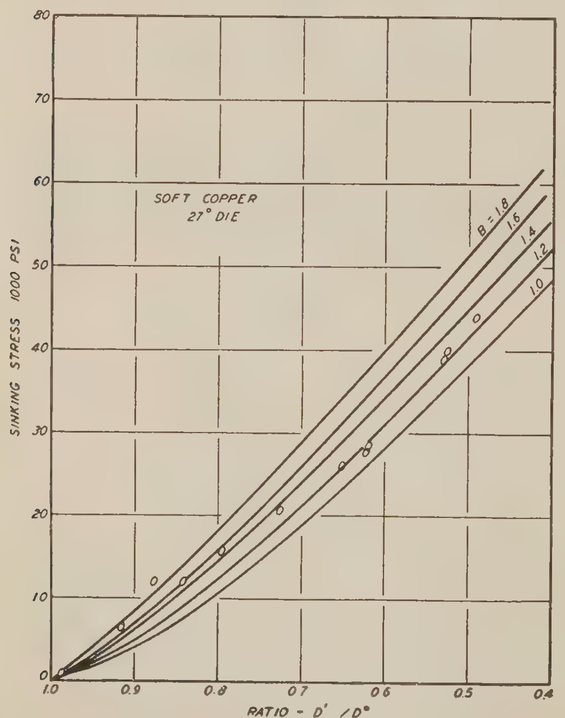

FIG. 10 THEORETICAL SINKING-STRESS CURVES CORRECTED FOR DIFFERENCES IN STRESS-STRAIN DIAGRAMS SHOWN IN FIG. 7

(Curves are for soft copper sunk through a 27-deg die. Experimental data are represented as circles.)

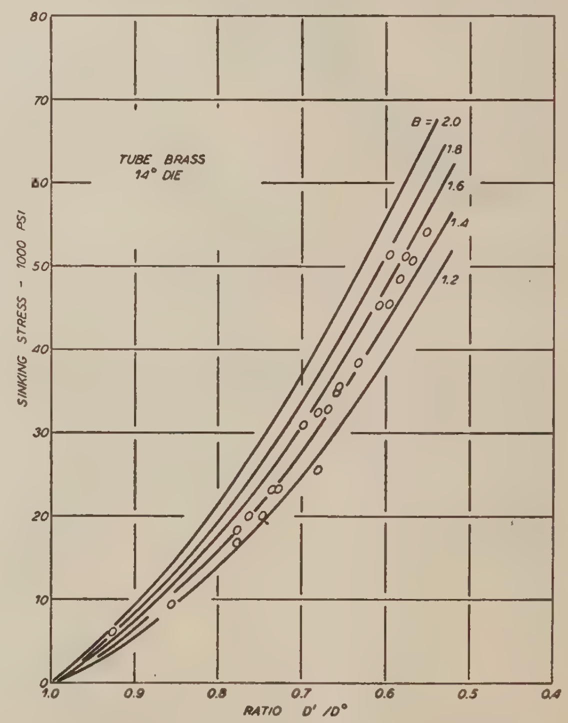

FIG. 12 THEORETICAL SINKING-STRESS CURVES CORRECTED FOR DIFFERENCES IN STRESS-STRAIN DIAGRAMS SHOWN IN FIG. 7

(Curves are soft tube brass sunk through a 14-deg die. Experimental data are represented as circles.)

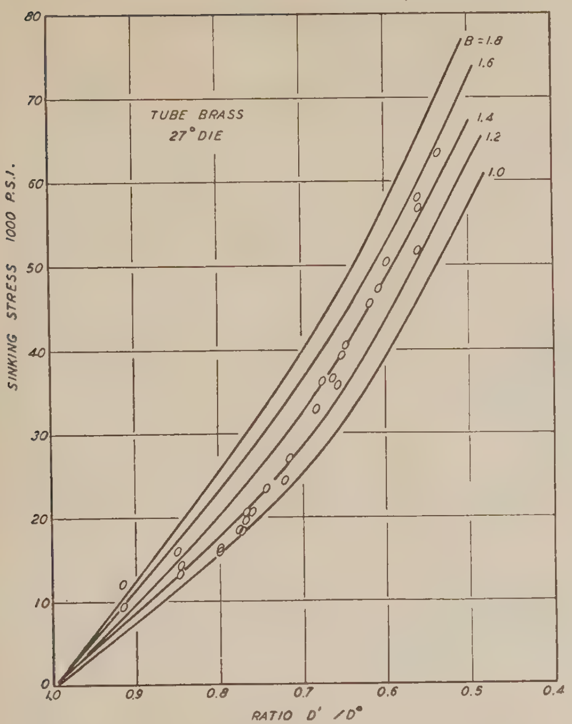

FIG. 13 THEORETICAL SINKING-STRESS CURVES CORRECTED FOR DIFFERENCES IN STRESS-STRAIN DIAGRAMS SHOWN IN FIG. 7
(Curves are for soft tube brass sunk through a 27-deg die. Experimental data are represented as circles.)

Neglecting the back pull s_1^o, the sinking stress given by Equation [10] becomes

$$s_1' = \frac{Bk_0}{B-1}\left[1-\left(\frac{D'}{D^o}\right)^{B-1}\right] \quad \text{[10a]}$$

The parameters, B_1 and B, in Equations [11] and [10a] are identical (see Equation [5])

$$B_1 = B = \frac{\tan \alpha + f}{\tan \alpha}$$

Considering that the cross-sectional area in wire-drawing is

$$A_1 = \pi D^2$$

consequently

$$\frac{A_1'}{A_1^o} = \left(\frac{D'}{D^o}\right)^2 \quad \text{[12]}$$

On the other hand, the cross-sectional area in tube-sinking is

$$A = \pi t D$$

consequently

$$\frac{A'}{A^o} = \frac{D'}{D^o} \quad \text{[13]}$$

Introducing Equations [12] and [13] into Equations [11] and [10a], respectively, yield the following

For wire-drawing

$$s_1'' = \frac{Bk}{B-1}\left[1-\left(\frac{A_1'}{A_1^o}\right)^{B-1}\right] \quad \text{[11a]}$$

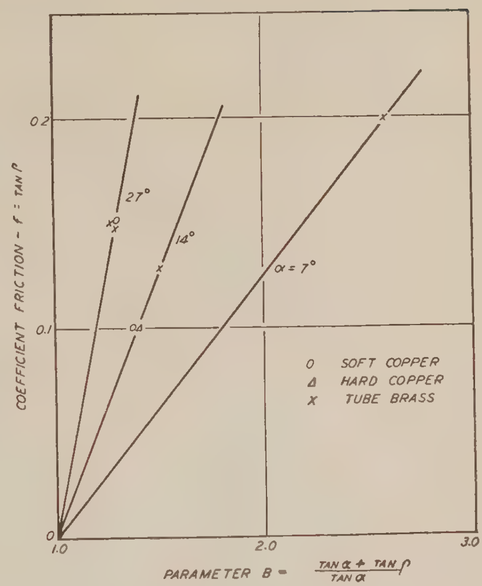

FIG. 14 PARAMETER B AS A FUNCTION OF COEFFICIENT OF FRICTION AND DIE ANGLE

and for tube sinking

$$s_1' = \frac{Bk_0}{B-1}\left[1-\left(\frac{A'}{A_1}\right)^{B-1}\right] \quad \text{[12a]}$$

Thus for a given ratio of cross sections at the entrance and exit, the ratio of the draw stresses for wire-drawing and tube-sinking should be

$$\frac{s_1''}{s_1'} = \frac{k}{k_0} = \frac{k}{1.1\,k} = 0.91$$

So far, the comparison of the two processes has not considered the effects of the shear stresses at the interface between die and metal, which increase the yield strength over that of a homogeneously deformed metal, as discussed previously. If these effects are larger in tube-sinking than in wire-drawing, the ratio k/k_0 would be larger than 0.9, and vice-versa.

To test these conceptions, two series of tests with a 14-deg and a 27-deg (half-angle) die, respectively, were carried out on annealed copper. The processing of both the copper tubes and the copper wire was such that identical stress-strain curves were obtained, represented in Fig. 3. All test conditions were also the same, as described previously for tube-sinking.

The results of the experimentation are shown in Figs. 15 and 16. For the smaller die angle, 14-deg, the ratio of the draw stresses for wire-drawing and tube-sinking respectively, is possibly slightly larger than 0.9. For the larger die angle this ratio is rather close to unity. Considering that the effects of shear increase with increasing die angle, the following conclusions can be drawn from these tests: (a) The draw stresses in tube-sinking and wire-drawing, respectively, correspond approximately to the fundamental difference in the flow stress derived from the difference in their stress states; (b) the shear stresses occurring under commercial processing conditions affect the metal flow in wire-drawing to a (slightly) greater extent than in sinking a thin-walled tube.

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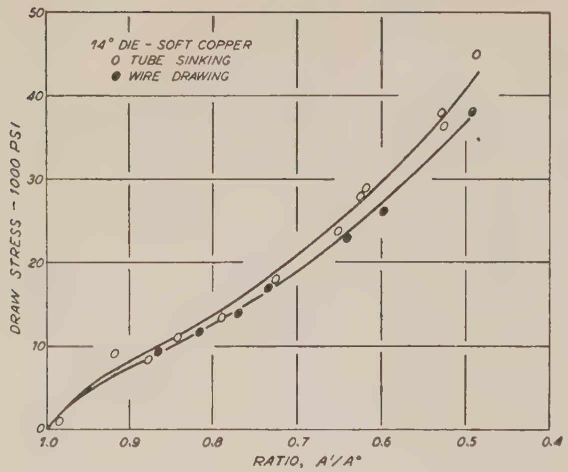

Fig. 15 Comparison of Draw Stresses for Sinking Annealed Copper Tube and Drawing Annealed Copper Wire Through a 0.50-In-Diam 14-Deg Die

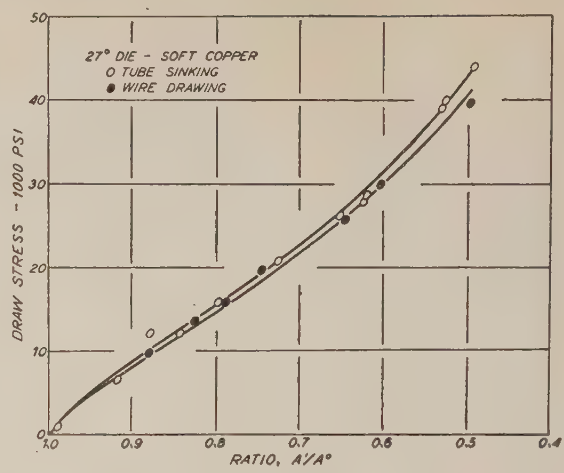

Fig. 16 Comparison of Draw Stresses for Sinking Annealed Copper Tube and Drawing Annealed Copper Wire Through a 0.50-In-Diam 27-Deg Die

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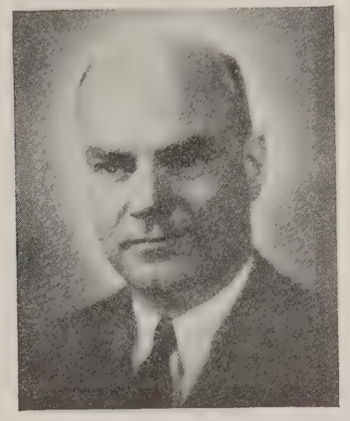

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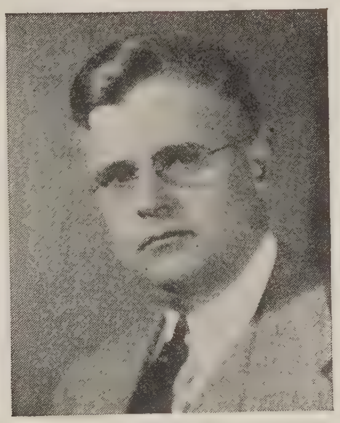

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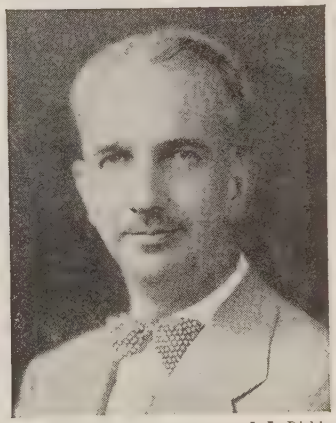

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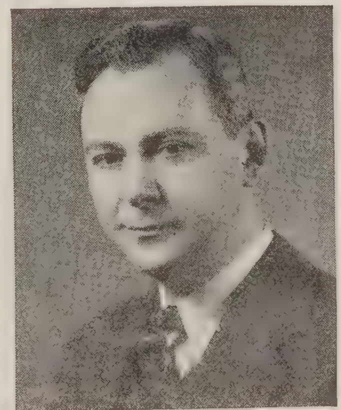

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FOREWORD

Both alphabetical and geographical lists of members of all grades except Student Members are included in this issue of the Membership List of The American Society of Mechanical Engineers. It is published as a Section Two of the A.S.M.E. Transactions, the third Section Two of Transactions in 1946. The Indexes to Publications appeared in January and the personnel of Council and committees and other general information in February. The colored page in the front of the Membership List is a guide to the latter pamphlet and to other sources of information about the Society.

Alphabetical List

Each entry in the Alphabetical List covers the member's grade and year of membership, symbols indicating registration in Professional Divisions, Society office held or award received, and his address.

The dates and letter symbols in parentheses immediately following the name of a member indicate his grade of membership and year of election or promotion to each grade. Where a single date is not accompanied by a letter symbol, the grade of Member is to be understood. Also where several dates are given, with no letter symbols, the final date indicates the year of election to full membership. The key to the symbols appears on the following page.

Professional Divisions Registration

One or more letter symbols appearing in a second parenthetical group designate Professional Divisions of the Society in which the member has signified his interest. The card sent to members asking for information for the 1946 Membership List gave them the opportunity to register in as many as six divisions, in order of interest. If a member failed to bring his Professional Division registration up to date, his latest previous registration has been retained. In cases where no preferred order has been indicated, the symbols are given in alphabetical order. The key to the symbols is given on the next page.

Addresses

The addresses given are, in most cases, those on record on March 1, 1946, the date set for the return of the information cards sent to members. However, the publication of the Membership List having been delayed, late returns were incorporated as long as was possible without causing further delay and an excessive increase in publication costs. Special attention was given to the listings of members being released from the armed forces, but up-to-date information regarding such members has not been received in many cases.

The extension of the actual closing date also made it possible to include all those who were in good standing on April 1, 1946.

Geographical List

The Geographical List gives the names of those in each locality. Facing the first page of that section of the book is an index giving the number of the page on which each country will be found.

In the United States and Canada, the name of each city which falls within a Section is followed by the name of that Section. Members residing in cities where no Section is mentioned may affiliate with any Section, without additional dues, upon request to the headquarters of the Society. Members are grouped according to the business addresses unless otherwise requested. Members of the armed forces who furnished an address in the United States in addition to an A. P. O. or F. P. O. service address, are listed under the former.

Restrictions on Use of Membership List

This Membership List is issued for the personal use of members of The American Society of Mechanical Engineers in connection with Society and professional affairs. Each member is expected to conserve it and not to permit his copy to be used for the basis of circularization. Such use is annoying to fellow members.

Key to Symbols

Grades of Membership

H—Honorary
F—Fellow
A—Associate
J—Junior

Professional Divisions

A—Aviation
B—Applied Mechanics
C—Management
D—Materials Handling
E—Oil & Gas Power
F—Fuels
G—Graphic Arts
H—Hydraulic
J—Metals Engineering
K—Heat Transfer
L—Process Industries
M—Production Engineering
N—Machine Design Group
O—Consulting Engineering Group
R—Railroad
S—Power
T—Textile
W—Wood Industries
Y—Rubber & Plastics
Z—Industrial Instruments

ALPHABETICAL LIST OF MEMBERS

A

Aarflot, Martin G. ('31) (BHK), Checker (Engrg.), Gibbs & Cox, Inc., 21 West St., New York, N.Y.; *for mail,* 180 N. 17th St., East Orange, N.J.

Aaron, Charles (J'37) (NOMB), Mch. Design Engr., Algren Mfg. Co., 325 Lafayette St., New York; *for mail,* 50 Westminster Rd., Brooklyn 18, N.Y.

Aaron, Hart Richard ('22; '26; '35) (CY), Lt. Comdr., U.S.N., 45 Rico Way (Marina Dist.), San Francisco 23, Calif.

Aaron, Robert Hart ('35; '45) (NMBJA), Design Engr., Harvey Mch. Co., 6200 Avalon Blvd., Los Angeles 3; *for mail,* 4186 Garthwaite Ave., Los Angeles 43, Calif.

Abadjieff, Ivan V. (J'29) (CHKFYM), Devel. & Pat. Engr., Leland-Gifford Co., 1001 Southbridge St.; *for mail,* Box 293, Worcester 1, Mass.

Abaya, Gonzalo T. (J'38) (CEM), 198 Espana St., Manila, P.I.

Abbey, Harold George (J'34) (ELZCS), Pres., Gen. Mgr., Ch. Engr., Indus. Installations Corp., 210 E. 40th St., New York 16, N.Y.

Abbondante, Cosmo (J'45) (MNCDBA), Tool Designer, Mech. Engr., Waltham Watch Co., 221 Crescent St., Waltham; *for mail,* 31 Rose St., Somerville 43, Mass.

Abbott, Douglas T. (J'42) (HCJMBA), Ch. Engr., Columbian Bronze Corp.; *for mail,* 188 Mt. Joy Ave., Freeport, N.Y.

Abbott, Ernest James ('37) (BCMNZ), Pres., Gen. Mgr., Physicists Research Co., 321 S. Main St., Ann Arbor, Mich.

Abbott, Horace Porter (J'42) (CLM), Supt., Lever Bros. Co., 5300 Holabird Ave., Baltimore 24; *for mail,* 4810 Keswick Rd., Baltimore 10, Md.

Abbott, James, Jr. (J'21), Mem. Tech. Staff, Bell Tel. Labs., Inc., New York, N.Y.; *for mail,* 334 Clinton Pl., Hackensack, N.J.

Abbott, Richard LeMoine (J'39) (SKMFJ), Sr. Mar. Draftsman, Fed. Shipbldg. & Dry Dock Co., Lincoln Highway, Kearny; *for mail,* 49c Garwood Trail, Lake Arrowhead, Denville, N.J.

Abbott, Ward D. (J'41) (MCSK), 33 Chauncy Lane, Orchard Park, N.Y.

Abbott, William George, Jr. ('19; '35) (TEALYN), Treas., Hillsborough Mills, Box 154, Wilton, N.H.

Abbott, William L. ('91; F'36; H'40) (BFS), *Manager,* '07-'10, *President,* '26; Treas., Pal-Verd, Inc.; *for mail,* 3500 Lake Shore Dr., Chicago, Ill.

Abderhalden, Robert Bruce (J'45), Design Engr., Westinghouse Elec. Corp., Wapak Rd.; *for mail,* 724 N. West St., Lima, Ohio.

Abdun-Nur, Edward Amin ('24; '39) (BHR), Engr., U.S. Bur. of Reclamation, Custom House, Denver 2, Colo.

Abel, Arthur William, Jr. (J'40) (ESA), Sales Engr., Cooper-Bessemer Corp., Box 1928, Odessa, Tex.

Abel, Stephen Taylor ('38) (MC), Ch. Engr., Engrg. Div., W. L. Coston & Sons, 1803 1st Natl. Bank Bldg., Birmingham 3; *for mail,* 3620—13th Ave. N., Birmingham 4, Ala.

Abel, Walter Lewis (J'41), Asst. Research Dept. Head, Research Div., United Shoe Mchy. Corp., Balch St., Beverly, Mass.

Abell, John Daniel (J'45) (CDJMN), Mech. Engr., Plant Maint., Internatl. Harvester Co. of Can., Ltd.; *for mail,* Apt. 1, 85 Proctor Blvd., Hamilton, Ont., Can.

Abelson, Richard Dodge (J'45) (CMO), Asst. Ch. Indus. Engr., Maremont Auto. Products Inc., 1625 S. Ashland Ave., Chicago 8; *for mail,* 5138 University Ave., Chicago 15, Ill.

Abendschein, Edward John (J'35) (CMNJ), Gen. Mgr., Chisholm-Ryder Co. of Pa., Hanover, Pa.

Aber, Joseph Stewart (J'43) (BHNSYZ), Mech. Engr., Goodall Rubber Co., 5 S. 36th St., Philadelphia; *for mail,* 1703 S. 58th St., Philadelphia 43, Pa.

Abercrombie, James H. ('01), Retired; "Rutland," West St., Reigate, Surrey, England.

Abercrombie, W. Taylor, Jr. ('21; '25; '35) (CHKMN), Sales Engr., U.S. Pipe & Fdy. Co., Burlington; *for mail,* Branch Pike, R.F.D. 2, Riverton, N.J.

Abernathy, George T. ('41) (HCS), Hyd. Engr., Newport News Shipbldg. & Dry Dock Co., Newport News; *for mail,* 126 James River Dr., Hilton Village, Va.

Aborn, Robert H. ('37) (JB), Supvr., Phys. Metal. Research, U.S. Steel Corp., Lincoln Highway, Kearny, N.J.

Abrahm, Michael Charles ('41) (KSE), Prin. Mech. Engr., New Orleans Pub. Serv., Inc., 317 Baronne St., New Orleans 9, La.

Abrams, Arnold (J'38), Project Dir., Jordanoff Aviation Corp., 595 Madison Ave.; *for mail,* 70 W. 95th St., New York, N.Y.

Acheson, Howard A. ('24; '34) (AFGY), Pres., Acheson Colloids Corp., Washington Ave., Port Huron, Mich.

Acheson, Louis Kruzan ('44) (CHBZTY), Lab. Co-ordinator, Hoover Co.; *for mail,* 816 Portage St., North Canton, Ohio.

Achilich, John Henry (J'42) (MAEHJY), Tool Engr., Plant 5, Grumman Aircraft Engrg. Corp., Bethpage, N.Y.

Ackart, Everett G. ('17), Ch. Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.

Acker, Alvin Wayne (J'42) (KAEHB), Army Air Forces, 1106th Base Unit, A.P.O. 845, c/o Postmaster, Miami, Fla.

Acker, George H. ('22; '28; '35) (BMN), V.P., Engrg. & Prod., Cleveland Worm & Gear Co., 3249 E. 80th St., Cleveland 4, Ohio.

Acker, Sidney H. ('35) (ERS), Mech. Engrg. Hall, Vanderbilt Univ., Nashville, Tenn.

Ackerman, Donald E. ('44) (CMDJLW), Mgmt. Consultant, McKinsey & Co., 240 Montgomery St., San Francisco 4, Calif.

Ackerman, George Ernest ('44), Cons. Engr., Mem. of Partnership, Holland, Ackerman & Holland, 20 N. Wacker Dr., Chicago; *for mail,* 1017 Wenonah Ave., Oak Park, Ill.

Ackerman, Robert Allen (J'44), 404 Yale Ave., Morton, Pa.

Ackerman, William Louis ('34; '46) (JMS), Mch. Designer, Keystone Steel & Wire Co., Peoria; *for mail,* 202 Carrie Ave., Peoria 5, Ill.

Ackermann, Frederick Andrew (J'39) (ABJMN), Designer, Van Tuyl Associates, 30 N. La Salle St., Chicago; *for mail,* 3107 S. Ridgeland Ave., Berwyn, Ill.

Ackert, George Ford (J'38) (BEM), Mech. Engr., Constr. Mgmt., Inc., 27 William St., New York; *for mail,* 29-16—166th St., Flushing, N.Y.

Ackley, Robt. A. ('40) (BLS), Design Engr., Republic Flow Meters Co., 2240 Diversey Pkwy., Chicago; *for mail,* 1535 Forest Ave., Wilmette, Ill.

Acomb, William Edward (J'39) (HWFSMC), Engr., Northwestern Mutual Fire Assn., 217 Pine St., Seattle 1; *for mail,* 3711 Tilden St., Seattle 99, Wash.

Acosta, Allan James (J'45) (HNMEJF), Ensign, U.S.N.R., U.S.S. *Rockingham,* APA-229, F.P.O., San Francisco; *for mail,* 1509 Artesia Blvd., Long Beach 5, Calif.

Acres, Harold Didsbury (J'43) (BYHASM), R.R. 1, Britannia Bay, Ont., Can.

Acurso, Louis A. (J'39), Asst. Engr., Pacific Gear Works, 2053 E. 38th St., Huntington Park, Calif.

Adair, John Gillis ('22; '30) (RSBEF), Mech. Engr., Interstate Commerce Comm., Washington 25, D.C.

Adams, Arlington Richard (J'39) (ABENS), Mech. Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 2181 Atkins Ave., Lakewood 7, Ohio.

Adams, Arthur Stanton ('41) (BACE), Office of Provost, Cornell Univ., Ithaca, N.Y.

Adams, Campbell V. ('19; '35), Mech. Engr., Asst. to Gen. Mgr., Vulcan Iron Works, 327 N. Irving Ave.; *for mail,* 2934 Wilson Ave., Chicago, Ill.

Adams, Charles Edward, Jr. (J'46), 318 Spring St., Portland, Me.

Adams, Clarence Harrison ('38) (BJ), Mech. Engr., Cone Automatic Mch. Co., Inc.; *for mail,* 29 N. Main St., Windsor, Vt.

Adams, Clarence Smith ('16; '25) (JLFDSM), Mech. Engr., Federated Metals Div., Am. Smelting & Refining Co.; *for mail,* 1820 Davis Ave., Whiting, Ind.

Adams, Comfort Avery ('17; F'39) (BEJKOR), Cons. Engr., Edward G. Budd Mfg. Co., 2450 Hunting Pk. Ave., Philadelphia 32, Pa.

Adams, Conrad Robert ('14; '35) (BCMNA), Pres., Gen. Mgr., Adams Engrg. Die & Tool Co., 1702 W. Washington St., South Bend 21, Ind.

Adams, E. Eugene ('35) (BGR), V.P., Trans. Research, Pullman Co., 79 E. Adams St., Chicago, Ill.

Adams, Earl Eugene ('44) (ECOY), Engr., C. L. Thompson & Co., 21 E. 40th St., New York 16; *for mail,* 969 Park Ave., New York 28, N.Y.

Adams, Franklin S. ('24; '33), Box 512, Stockbridge, Mass.

Adams, Frederick Clarke (J'45) (CMY), 67 Constance St., Toronto 3, Ont., Can.

Adams, Harold Eugene ('30) (HNEASO), Ch. Engr., Nash Engrg. Co., South Norwalk, Conn.

Adams, Harold Lamont ('30; '40) (AB), *Student Award,* '30; Unit Ch., Struc. Testing, Boeing Aircraft Co., 200 W. Michigan St., Seattle 6; *for mail,* 3222 Sierra Dr., Seattle 44, Wash.

Adams, Henry R. ('38), Treas., Gen. Mgr., William Bergfels & Co., Inc., 40 Walnut St., Newark; *for mail,* 53 Woodland Rd., Madison, N.J.

Adams, James Francis, Jr. (J'45) (OJKLSM), Engr., Tech. Serv., Solvay Process Co., 523 Allen St., Syracuse, N.Y.

Adams, James Walter ('44) (CELM), Adams Engrg. Co., 1618 N. Fitzhugh, Dallas 4; *for mail,* 3525 Milton Ave., Dallas 5, Tex.

Adams, John ('36; '45) (BCJLMO), Ch. Tool Designer, Detroit Gear & Mch. Co. (Div. of Borg-Warner Corp.), 670 E. Woodbridge, Detroit; *for mail,* 2031 Carpenter St., Detroit 12, Mich.

Adams, John James (J'44), Cpl., Army Serv. Forces, 3rd Platoon, 3189th Engr. Serv. Bn., Trng. Center, Ft. Lewis, Wash.

Adams, Lyman Daniel ('17; '24) (ABCM), V.P., Gen. Mgr., Barnes-Gibson-Raymond (Div. of Associated Spring Corp.), Miller Ave., Detroit 11, Mich.

Adams, Orlando Percival ('22; '31) (JSAHFC), Dir. of Devel., Natl. Tube Co., Frick Bldg., Pittsburgh 30, Pa.

Adams, Orville L., Sr. (A'43) (CEL), Editor, Tech. Indus. Dept., Prentice-Hall, Inc., 70—5th Ave.; *for mail,* 32 W. 40th St., New York, N.Y.

Adams, Paul ('42) (BCDHY), Supvr., Design & Drafting, Hood Rubber Co., 98 Nichols Ave., Watertown, Mass.

Adams, Raleigh John ('28; '35) (BJS), 644 Diversey Pkwy., Chicago, Ill.

Adams, Richard Donald ('37; '46) (EKSY), Pres., R. D. Adams Co., 555 Post St., San Francisco 2, Calif.

Adams, Robert Powell (J'40) (HBM), c/o Y.M.C.A., DeKalb & Airy Sts., Norristown, Pa.

Adams, Stafford Morley Sidney (J'31) (CNMDHY), Works Mgr., Truscon Steel Co. of Can., Ltd., 2275 Ottawa St., Walkerville; *for mail,* 2241 Moy St., Windsor, Ont., Can.

Adams, Thomas Burton, Jr. (J'42) (AJM), 1636 Maine St., Quincy, Ill.

Adams, Walter A. (J'42) (KMAW), Design Engr., Bur. of Ships, Navy Dept., Washington 25, D.C.; *for mail,* 5700—64th Ave., Riverdale, Md.

Adams, Walter Holbrook ('08; '12) (OSEHF), Cons. Mech. & Civ. Engr., 1633 Ard Eevin Ave., Glendale 2, Calif.

Adams, William Elvin (J'44) (SFEGNH), Asst. Ch. Engr., Am. Chicle Co., 30-30 Thomson Ave., Long Island City; *for mail,* 35 Harvard St., Westbury, N.Y.

Adamson, Arthur Paul (J'41) (ZAMO), Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* R.R. 3, Ballston Spa, N.Y.

Adamson, Arthur V. ('42) (FS), Dist. Mgr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.

Addams, Homer ('28), Pres., Kewanee Boiler Co., Inc. & Fitzgibbons Boiler Co., Inc.; *for mail,* 101 Park Ave., New York, N.Y.

Addicks, Lawrence ('11) (OJC), Cons. Engr., Bel Air, Md.

Addicks, Mentor C. (J'28) (BEFKLS), Ch. Engr., Internatl. Milling Co., 800 McKnight Bldg., Minneapolis 1; *for mail,* 5107 Arden Ave., Minneapolis 10, Minn.

Addington, Herbert B. ('37) (OSFLE), Partner, Myers, Fuller & Addington, 21 E. 40th St., New York 16, N.Y.

Addoms, John Fillmore ('45) (KP), Sr. Engr., C. F. Braun & Co., 1000 S. Fremont Ave., Alhambra, Calif.

Addy, John Louis, Jr. (J'39) (NMC), Sales Engr., Addy & Luby Mchy. Co., 8316 Woodward Ave., Detroit 2, Mich.

Adelman, Arthur ('16) (MNJBLZ), Ch. Engr., Am. Div., Indus. Serv., Ord. Dept., U.S.A., The Pentagon, Washington; *for mail,* 3709 Military Rd., Washington 15, D.C.

Adelson, Joseph S. ('34; '35) (SKACD), Ch. Metallurgist, Steel & Tubes Div., Republic Steel Corp., 224 E. 131st St., Cleveland, Ohio.

Adler, Abraham Charles (J'44) (ABE), 354 E. Eleanor St., Philadelphia 20, Pa.

Adler, Frank Read (J'42) (MACON), Supt. Mfg. Engrg., Adel Precision Products Inc., 10777 Van Owen St., Burbank; *for mail,* 2759 Westwood Blvd., Los Angeles 34, Calif.

Adler, George (J'38) (BKSN), Mar. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 2148 Barnes Ave., New York 60, N.Y.

Adler, Rubin (J'45) (BAHGNK), Jr. Test Engr., Arma Engrg. Corp., 254—36th St., Brooklyn; *for mail,* 45 Georgia Ave., Brooklyn 7, N.Y.

Adler, Rudolph C. (J'40) (ABMC), Owner, Firm of R. C. Adler, Mech. Engr., Rm. 419, Ocean Center Bldg., Long Beach 2, Calif.

Adolphson, Roy T. (J'34) (BJM), Capt., 16th Sep. Field Arty. Trng. Bn., Replacement Trng. Center, Ft. Bragg, N.C.

Adzima, George Rudolph T. (J'30) (NMJD), Asst. Ch. Engr., Westfield Mfg. Co., 1 Cycle St.; *for mail,* 65 E. Silver St., Westfield, Mass.

Africano, Alfred ('44) (ABE), Project Engr., Curtiss-Wright Corp.; *for mail,* 25 Forest Ave., Caldwell, N.J.

Agee, Charles Daniel (J'43) (WCMNBD), V. P., Prod. Mgr., Acme Screen Co., 915 S. Peak St., Dallas 1, Tex.

Agerstrand, Carl Leonard Folke ('45) (CMHJAN), Pres., Gen. Mgr., Agerstrand Corp., 1817-29 Commerce St.; *for mail* 160 E. Larch Ave., Muskegon, Mich.
Agner, Owen B. ('12; '19) (CFHKS), Application Engr., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4, Pa.
Agnew, Alexander Crawford (J'44), 16 Houghton Ave. N., Hamilton, Ont., Can.
Agnew, John Thomas (J'46), Asst. Prof., Mech. Engrg. Bldg., Purdue Univ.; *for mail,* 1009½ Hartford, Lafayette, Ind.
Agnew, Thomas Charles (J'30) (ZLFES), Mech. Engr., Minneapolis-Honeywell Regulator Co., Ltd., 117 Peter St., Toronto 2B; *for mail,* 52 Lyngrove Ave., Toronto 9, Ont., Can.
Agnew, William Griswold (J'43) (TSNB), Plant Engr., Chicopee Mfg. Corp., W. Main St., Chicopee Falls, Mass.
Agrell, Charles Fabian ('31) (DEFMOS), Dist. Engr., So. Cotton Oil Co., Montgomery, Ala.
Agronin, Tany ('31; '42) (HLWSNC), Ch. Engr., Shartle Bros. Mch. Co., Clark St., Middletown, Ohio.
Ahara, Edw. V. ('25; '36), Asst. Engr., Oxford Paper Co., Rumford, Me.
Ahlstromer, Magnus John, Jr. (J'41) (MJNLCA), Engr., Chicago Screw Co., 1026 S. Homan Ave., Chicago 24; *for mail,* 1625 Grace St., Chicago 13, Ill.
Ahrendsen, Lowell Kenneth (J'45) (LMN), Jr. Asst. Mech. Engr., Eastman Kodak Co., 343 State St., Rochester 13, N.Y.
Ahrens, Carl ('32; '35) (CYML), Plant Mgr., Plastics Div., Am. Cyanamid Co., Bound Brook; *for mail,* 1416 Park Ave., Plainfield, N.J.
Ahrens, Henry Richard, Jr. ('43; '44) (WYN), Engrg. Staff, Improved Paper Mchy. Corp., Box 515; *for mail,* Apt. 410, 184 Main St., Nashua, N.H.
Aidlin, Samuel S. (J'45) (HNBSEO), Project Engr., Mar. & Hyd. Devices, Greer Hydraulics, Inc., 454—18th St., Brooklyn 15; *for mail,* 2045 E. 22nd St., Brooklyn 29, N.Y.
Aikins, John Roberts ('39) (ENKHB), Asst. Design Engr., Gulf Research & Devel. Co., P.O. Drawer 2038, Pittsburgh 30, Pa.
Aikins, William Roy ('44) (MCOBNJ), Mech. Engr., Asst. to Ch. Engr., S. F. Bowser Co. Ltd., 183 George St., Toronto 2; *for mail,* 1196 Avenue Rd., Toronto 12, Ont., Can.
Ainslie, Henry Wilbur, Jr. (J'43) (ABHMJL), 276-A Beechwood St., Cohasset, Mass.
Airey, John ('15; '19) (CM), *Life Member for Distinguished Service,* '23; Pres., King-Seeley Corp.; *for mail,* 2009 Washtenaw Ave., Ann Arbor, Mich.
Airston, Alexander Jonathan ('21) (GCAMJZ), Tech. Writer, Instr. Books, RCA Victor Div., Radio Corp. of Am., Camden, N.J.; *for mail,* 2412 Hollis Rd., Merwood, Havertown, Upper Darby, Pa.
Aker, Donald Carr (J'42) (BCH), Ensign, 255 Hollywood Dr., Trenton 9, N.J.
Akerlow, Karl Gustave W. ('44) (FKE), Indus. Engr., West. United Gas & Elec. Co., 50 Fox St.; *for mail,* 75 S. Fordham Ave., Aurora, Ill.
Akerman, Nigel (J'38) (BJM), 606½ Cathedral St., Baltimore, Md.
Albano, Raymond James (J'44) (CJMN), Spec. Engr., Rolling Div., South Works, Carnegie-Ill. Steel Corp., 3426 E. 89th St.; *for mail,* 11239 Ave. J, Chicago 17, Ill.
Alberga, Glenn Harcourt ('34; '38) (OCTDMZ), Gen. Mgr., John J. Plocar Co., Singer Bldg., Stamford; *for mail,* 3 Woodside Dr., Greenwich, Conn.
Albert, Calvin Dodge ('11) (NBMJD), Prof. Mch. Design, Emeritus, Cornell Univ.; *for mail,* 205 Eddy St., Ithaca, N.Y.
Albert, Charles E. (J'42), Mgr., Bailey Meter Co., 618 W. 26th St., Kansas City 8, Mo.
Albert, Donald Joseph (J'46) (AHM), 18 Bardin St., Rochester 13, N.Y.
Albert, Joseph Adam (J'45) (BCE), Design Engr., Busch-Sulzer Bros.-Diesel Eng. Co., St. Louis; *for mail,* 2263 Indiana Ave., St. Louis 4, Mo.
Albert, Peter William (J'45) (SZCKOL), Test Engr., Gen. Elec. Co., Schenectady, N.Y.; *for mail,* 168 Fulton Park Ave.; Waterbury 70, Conn.
Albert, Robert C. (J'40) (NBKEMH), 101 Parker Pl., Trenton 9, N.J.
Albrecht, Don Kenneth (J'39) (OWCM), 5636 Fernwood Ave., Los Angeles 28, Calif.
Albrecht, Edwin George (J'41) (CMZBAN), Sr. Mech. Engr., Cornell Aero. Lab., Hangar 2A, Buffalo, N.Y.
Albrecht, George Frederic ('31) (CMT), 24 Pine St., Whitinsville, Mass.
Albrecht, Robert James (J'39) (ZYMC), Seaman, 1st Class, U.S.N.R., 1753 E. 7th St., Brooklyn 23, N.Y.
Albright, Albert Allen (J'43) (CSFMN), Sales Engr., Cooper-Bessemer Corp., N. Sandusky St., Mt. Vernon; *for mail,* R.F.D. 2, Fredericktown, Ohio.
Albright, Clayton Monroe, Jr. (J'40) (ZLTY), Devel. Engr., Exper. Sta., E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Alburger, Harry Allyne (J'27) (MNBAJY), Sr. Engr., Bur. of Ord., Navy Dept., 17th & Constitution Ave., Washington 25; *for mail,* 3624—10th St., N.W., Washington 10, D.C.
Alciati, Charles Joseph (J'40) (CMD), Asst. Mgr., Indus. Mchy. Div., Moore Dry Dock Co., Foot of Adeline St., Oakland; *for mail,* 434—5th Ave., San Francisco 18, Calif.
Alcock, George Walter ('40), Pres., Franklin Ry. Supply Co., Inc., 60 E. 42nd St., New York 17, N.Y.
Alcorn, Harry Jackson (J'40) (A), Designer, Joy Mfg. Co., Franklin; *for mail,* 208 Lincoln St., Oil City, Pa.
Alcorn, Robert L., Jr. (J'45), Engrg. Dept., Chambersburg Engrg. Co., Chambersburg, Pa.
Aldag, Robert, Jr. (J'45) (REFBCG), Mech. Inspr., Chicago, Burlington & Quincy R.R. Co., 547 W. Jackson Blvd., Chicago 6, Ill.
Alden, Carroll Russell ('27) (EHM), Research Engr., Patents, Ex-Cell-O Corp., 1200 Oakman Blvd., Detroit 6, Mich.
Alden, Clifford Edwin ('42) (CFHRSZ), Mgr., Utilities Sales Div., Foxboro Co., Foxboro; *for mail,* 20 Glendale Rd., Sharon, Mass.
Alden, Herbert Watson (F'42), Dir. Engrg., Timken-Detroit Axle Co., Detroit, Mich.
Alden, John Arthur (J'44) (MKCDAJ), Asst. Ch. Engr., Refrigeration Engrg. Inc., 6107 S. Central, Los Angeles 1; *for mail,* 3714½ E. 53rd St., Maywood, Calif.
Alden, John D. ('12) (FMOS), Apt. A3, Unit 2, 40 Lexington St., Newark, N.J.
Alden, John Leslie ('14; '21; '24) (CMNWTY), Mgr., Tel. Equip. Div., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 72 Durand Rd., Maplewood, N.J.
Alden, Vern E. ('20; '35) (FLS), Rm. 2350, 120 S. La Salle St., Chicago, Ill.
Aldinger, Heinrich Karl ('24) (CEHNOS), Plant Engr., Celotex Corp., Marrero; *for mail,* 1317 State St., New Orleans 15, La.
Aldrich, Benj. M. ('27; '34; '35) (EFJ), Asst. Prof. Mech. Engrg., Okla. Agric. & Mech. College, Stillwater, Okla.
Aldrich, Henry E. ('19; '25) (SFK), Mgr., Am. Boiler & Affiliated Industries, 16 Park Row, New York 7, N.Y.
Aldrich, Henry M. (J'40) (ES), Lt., U.S.N.R., 136-42—39th Ave., Flushing, N.Y.
Aldrich, Horace E. ('28), N.Y. State Gas & Elec. Corp., Liberty, N.Y.
Aldrich, John Glodding ('01) (CM), Pres., New England Butt Co., 304 Pearl St., Providence 7, R.I.
Aldrich, Richard Chanler ('33; '41) (FY), Code 341, Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* Barrytown, Dutchess Co., N.Y.
Aldrich, Wickham Hurd ('35) (SFC), Supt., Steam Dept., Cleveland Elec. Illum. Co., 75 Public Sq., Cleveland 1, Ohio.
Aldridge, C. Foster (J'45) (ZNL), Project Engr., Taylor Instrument Cos., 95 Ames St., Rochester 11, N.Y.
Aldridge, Eugene Froyard ('26; '35) (CEFHS), Engrg. Consultant, Lumbermen's Mutual Casualty Co., 100 Adelaide St. W., Toronto, Ont., Can.
Aldrin, Edwin Eugene ('19; '28) (AO), Aviation Mgr. & Consultant, Atlas Supply Co., 744 Broad St., Newark; *for mail,* 25 Princeton Pl., Montclair, N.J.
Aldworth, Edward H. ('42), 68 Brookby Rd., Scarsdale, N.Y.
Aleck, Benjamin Jacob (J'44) (BAHOWK), c/o Cohen, Apt. 20, 1270 E. 18th St., Brooklyn, N.Y.
Alexander, Alvin (J'37) (BJM), Ch. Draftsman Sinclair-Collins Valve Co., 454 Morgan Ave.; *for mail,* 442 Aqueduct St., Akron, Ohio.
Alexander, Mrs. Beatina Wilkinson (J'43) (ABJ), Process Engr., Radio Corp. of Am., Harrison; *for mail,* 28 Barnard Pl., Elizabeth, N.J.
Alexander, Charles Anderson ('99; '05) (CMS), 1260 Clover Rd., Rochester 10, N.Y.
Alexander, Charles Anton ('22; '35) (CEN), Ch. Engr., Woolson Spice Co., Summit & Sandusky Sts., Toledo; *for mail,* 1920 Erie St., Toledo 11, Ohio.
Alexander, Edward Emmet (A'08) (BHM), Engr., Taylor-Wharton Iron & Steel Co.; *for mail,* Nassau Rd., High Bridge, N.J.
Alexander, Joseph Bruce (J'43) (BAJ), 541 Spring St., New Kensington, Pa.
Alexander, Marvin Monroe, Jr. (J'43) (KABNCM), Stress Analyst, Consltd. Vultee Aircraft Corp., Ft. Worth; *for mail,* 3928 Locke St., Ft. Worth 7, Tex.
Alexander, Peter M. (J'44) (LKZ), Asst. Plant Engr., Colgate-Palmolive-Peet Co., 6th & Carlton, Berkeley 2; *for mail,* 1085 Folsom Ave., Hayward, Calif.
Alexander, Ralph Irwin ('19; '23; '35) (CEHKRS), Mech. Engr., Celanese Corp. of Am., Cumberland; *for mail,* 181 E. Main St., Frostburg, Md.
Alexander, Walter Crowell (J'43) (AFRWYH), Seaman, 1st Class, U.S.N., Co. 178, Naval Trng. Center, Great Lakes, Ill.; *for mail,* 653 St. John's Pl., Brooklyn 16, N.Y.
Alexander, William, Jr. (J'44), 444 S. Burlington Ave., Los Angeles, Calif.
Alexander, William Thurlow ('30; '46) (JMS), Prof. Indus. Engrg. & Dean, College of Engrg., Northeastern Univ., 360 Huntington Ave., Boston 15, Mass.
Alexander, William W. ('43) (DAHJM), Supt., Pacific Rock & Gravel Co., Box 30; *for mail,* 147 Lincoln Pl., Monrovia, Calif.
Alexandroff, Walter Anthony ('42) (TFMJ), Tech. Adviser, Pate Oil Co., 3460 W. Leeds Pl., Milwaukee; *for mail,* 2637 N. 69th St., Milwaukee 13, Wis.
Alexion, John (J'42) (CDM), Indus. Mgmt. Engr., Ornamental Plastics Mfg. Co., 329 Canal St., New York; *for mail,* 45-35—46th St., Woodside, L.I., N.Y.
Alford, Wayne Alexander (J'42), Erecting Engr., Ingersoll-Rand Co.; *for mail,* 302 W. Water, Painted Post, N.Y.
Alger, Harley C. (J'08), Mech. Engr., Methods Stands., W. F. Hall Printing Co., 4600 W. Diversey Ave., Chicago; *for mail,* 312 N. Grove Ave., Oak Park, Ill.
Alger, Philip Langdon ('23; '35) (CBJO), Staff Asst. to Mgr. of Engrg., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1758 Wendell Ave., Schenectady 8, N.Y.
Algino, Joseph Michael (J'46) (KMYBL), Plant Layout Engr., Electro-Motive Div., Gen. Motors Corp., La Grange; *for mail,* 2506 W. Flournoy St., Chicago 12, Ill.
Alico, John (J'42) (JBMNA), Dir. Research, Devel., Natl. Magnesium Corp., 74 Trinity Pl., New York 6; *for mail,* Box 71, Route 1, Tuttle Rd., Briarcliff Manor, N.Y.
Alker, Hayward R. (J'42) (COMD), Sr. Staff Engr., George S. Armstrong & Co., Inc., 52 Wall St., New York 5, N.Y.
Allan, Clifford Eligh (J'45) (NDSHLM), Mech. Engr., Engrg. Dept., Canada Paper Co.; *for mail,* Box 189, Windsor Mills, Que., Can.
Allan, George William ('36) (MNGCB), Asst. Ch. Engr., Intertype Corp., 360 Furman St., Brooklyn 2; *for mail,* 1134 E. 43rd St., Brooklyn 10, N.Y.
Allan, Walter Everett (J'40) (MJC), Warrant Officer (j.g.), Corps of Engrs., U.S.A., 91 W. 11th Ave., Columbus, Ohio.
Allardice, Thomas B. ('29), Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 59 Hanover Rd., Mountain Lakes, N.J.
Allardt, Ernst W. ('41), Ch. Engr., Welded Tube Div., Babcock & Wilcox Co.; *for mail,* 1635 Parkway Blvd., Alliance, Ohio.
Allcut, Edgar Alfred ('22) (SEAKCF), Prof., Head, Dept. of Mech. Engrg., Univ. of Toronto, Toronto, Ont., Can.
Allen, Alan R. ('38) (LS), Engr., Kennedy-Van Saun Mfg. & Engrg. Corp., 2 Park Ave., New York, N.Y.; *for mail,* Newark Athletic Club, Newark, N.J.
Allen, Bruce James (J'44), Technician, 4th Grade, U.S.A., Hdq. Co., 60th Inf., A.P.O. 9, c/o Postmaster, New York, N.Y.
Allen, Charles Lellan ('23) (AEFKRS), Retired; 711 N. Allen St., State College, Pa.
Allen, Charles Malcolm (J'44) (BAHMJ), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus, Ohio.
Allen, Charles Metcalf ('01; '03; F'36; H'44) (HS), *Manager,* '28-'31, *Vice-President,* '31-'33; *Worcester Reed Warner Medallist,* '36; Prof. Hyd. Engrg. & Dir., Alden Hyd. Lab., Worcester Poly. Inst., Boynton St., Worcester 8, Mass.
Allen, Chauncey Darrah ('31) (RY), Mgr., Sales & Serv., Baldwin Loco. Works, 2929—19th St., San Francisco 10, Calif.
Allen, Donald Parsons ('40; '43) (MBCDNH), Ch. Engr., Rensselaer Valve Co., Ontario St., Cohoes; *for mail,* 133—2nd Ave., Troy, N.Y.
Allen, Earle Francis ('44) (CONLR), Ch. Engr., Manning, Maxwell & Moore, Inc., 5 Watson St., Boston 18; *for mail,* 90 Fairbanks Ave., Wellesley Hills 82, Mass.
Allen, Edward ('43) (SFL), East. Div. Serv. Mgr., Republic Flow Meters Co., 420 Lexington Ave., New York 17, N.Y.; *for mail,* 1481 Gaylord Terrace, West Englewood, N.J.
Allen, Edward Knight, Jr. (J'37) (HMBC), Mech. Engr., Leland-Gifford Co., 1001 Southbridge St., Worcester 1; *for mail,* 18 Saxon Rd., Worcester 2, Mass.
Allen, Eldridge Edward ('45) (CMBNZ), Gp. Supvr., Research, Am. Can Co., 11th Ave. & St. Charles Rd., Maywood, Ill.
Allen, Ernest C. ('20; '24) (SEJW), Engr., Steam Turbines Dept., Allis-Chalmers Mfg. Co., 1126 S. 70th St., West Allis 14; *for mail,* 617 Glenview Ave., Wauwatosa 13, Wis.
Allen, Frank B. (J'35), 41 Afterglow Way, Montclair, N.J.
Allen, Henry A. ('00), Mech. Engr. in Charge, City of Chicago, 811 N. Michigan Ave.; *for mail,* 3138 Sheridan Rd., Chicago, Ill.
Allen, Herbert (J'36) (CNHD), V.P., Engrg. & Mfg., Cameron Iron Works, Inc., Box 1212, Houston 1, Tex.
Allen, Herrick K. (J'39) (MLDCN), 51 Newton St., Philadelphia 18, Pa.
Allen, Hugh M. ('35; '41) (AB), Asst. Mgr., Anchor Oil Co., Box 489; *for mail,* Box 775, Maricopa, Calif.
Allen, J. Matteson (J'43) (CMJSYZ), Mfg. Engr., Westinghouse Elec. Corp., Lester, Pa.
Allen, Jean March ('13) (HODSB), Cons. Engr., 3552 Lowry Rd., Los Angeles 27, Calif.
Allen, John Cochran ('22) (JMO), 301 E. Leasure Ave., New Castle, Pa.
Allen, John Dearing (J'34) (CKLAJ), Pres., Mgr., Process Equip. Engrs., 200 E. Reed; *for mail,* 115 E. Ann, Belding, Mich.

Allen, John Lee (J'42) (AKHOND), Designer, Douglas Aircraft Co., Inc., 300 Ocean Park Blvd., Santa Monica; *for mail*, 4650 Loleta Ave., Los Angeles 41, Calif.
Allen, John Warren (J'43) (AJCMDY), Asst. Tool Engr., Collins Radio Co., 855—35th St., N.E.; *for mail*, 1412—3rd Ave., S.E., Cedar Rapids, Iowa.
Allen, Joseph Fletcher ('45) (NMZBC), Project Engr., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3, Ohio.
Allen, Julian Howard (J'42) (AMB), 634 Otis Blvd., Spartanburg, S.C.
Allen, Laird Robert, Jr. (J'40) (ZSJBEF), Designer, Hagan Corp., 323—4th Ave., Pittsburgh; *for mail*, 836 Van Kirk St., Clairton, Pa.
Allen, Lucian Taylor (J'39) (ABHCJM), Lt., Submarine Serv., U.S.N.R.; 1 Lincoln Ave., Holden, Mass.
Allen, Lyman Hall, Jr. ('45), Corp. Engr., Am. Viscose Corp., 35 S. 9th St., Philadelphia 7; *for mail*, 14 Crest Lane, Swarthmore, Pa.
Allen, Maynard Coney ('34) (DEJR), Mech. Engr., Climax Molybdenum Co., Box 428; *for mail*, Box 295, Route 3, Golden, Colo.
Allen, Michael H. P. ('34) (KOFS), Dir., Gen. Mgr., Bell's Asbestos & Engrg. (Africa) Ltd., Box 4378, Johannesburg, S. Africa.
Allen, Millard Henry, Jr. (J'43) (HNO), Owner & Oper., Engrg. & Drafting Serv., 1201 Annapolis Dr., Corpus Christi, Tex.
Allen, Oliver Field ('17) (EROSWA), 117 Liberty St., New York 6; *for mail*, 59 W. 44th St., New York 18, N.Y.
Allen, Philip (J'35) (JEMZ), Mech. Engr., Test Lab., Bldg. 121, Naval Shipyard, U.S. Naval Base, Philadelphia 12, Pa.
Allen, Richard M. (J'36) (ES), Mech. Engr., Lawrance Aeronautical Corp., Linden, N.J.; *for mail*, 1775 E. 13th St., Brooklyn 29, N.Y.
Allen, Robert Chester ('41) (SAJENR), Mgr., Ch. Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1; *for mail*, 644 N. 68th St., Wauwatosa 13, Wis.
Allen, Robert Fay (J'45) (KSZY), Jr. Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1; *for mail*, 83 Willow Ave., Larchmont, N.Y.
Allen, Robert Kendrick (J'44) (FKS), Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York, N.Y.; *for mail*, 120 Lincoln Pl., Waldwick, N.J.
Allen, Robert Leon (J'44) (EKMNBR), Detail Engr., Allison Div., Gen. Motors Corp., Indianapolis; *for mail*, 5520 W. 16th St., Indianapolis 8, Ind.
Allen, Robert Lewis ('43), Asst. Prof., Mech. Engrg. Dept., Ga. Sch. of Tech.; *for mail*, 1446 Piedmont Ave., Atlanta Ga.
Allen, Robert Patrick (J'45) (CDM), Test Engr., Worthington Pump & Mchy. Corp., Harrison, N.J.; *for mail*, 676 Tinton Ave., Bronx 55, N.Y.
Allen, Robert Rubin (J'43) (ABC), Engrg. Dept., Carl L. Norden, Inc., 80 Lafayette St., New York; *for mail*, 3223 Metz Ave., Endwell, N.Y.
Allen, Samuel Gordon ('45) (CRS), Chmn. of Bd., Lima Loco. Works, Inc., 60 E. 42nd St., New York 17, N.Y.
Allen, Samuel Leeds ('42) (OMTBNC), Sales Engr., J. E. Rhoads & Sons, 35 N. 6th St., Philadelphia 6, Pa.
Allen, Miss Shirli Constance (J'45) (KHAN), Draftsman "B", Douglas Aircraft Co., Inc., Santa Monica; *for mail*, 1254 W. Bonita, Azusa, Calif.
Allen, Thos. H. ('12), Pres., Memphis Light, Gas & Water Div., Box 388, Memphis 1, Tenn.
Allen, William B. ('45) (CHMS), Asst. Ch. Design Engr., Tenn. Valley Authority, 304 Union Bldg., Knoxville, Tenn.
Allen, Wm. Gordon (J'45) (CES), 8306 Custer Rd., Bethesda, Md.
Allen, Wyeth ('29) (COM), Mgmt. Consultant, 161 W. Wisconsin Ave., Milwaukee 3, Wis.
Aller, Willis Fay ('42), Dir. of Research, Sheffield Corp., Dayton 1; *for mail*, 69 Pinehurst Ave., Dayton 5, Ohio.
Alley, Kenneth Gilbert (J'35) (NMACO), Research & Planning Engr., Vendo Co., 1907 Grand Ave., Kansas City; *for mail*, 528 Maple Blvd., Kansas City 1, Mo.
Allgaier, Joseph Michael (J'34) (MCDA), 701 Chestnut Hill Ave., Baltimore 18, Md.
Allgeyer, Guy H. (J'42) (ML), Mech. Engr., Swartzbaugh Mfg. Co., 1336 W. Bancroft St., Toledo 6; *for mail*, 2650 Gracewood Rd., Toledo 12, Ohio.
Alliger, William Thomas ('26) (EHJS), Pres., Alliger & Sears Co., 5646 Navigation Blvd., Houston 1, Tex.
Allingham, Henry William ('16), Wharf Road, Ponders End, Middlesex, London, England.
Allison, Carl O. ('28; '30; '35) (CEH), Assoc. Constr. Engr., Fed. Works Agency, Pub. Bldgs. Admin., Dist. 2, 731 Custom House, New York, N.Y.
Allison, Ray Dixon (J'39) (CJML), Research Engr., Cincinnati Planer Co., 3120 Forrer St., Cincinnati 9, Ohio.
Allman, William N. ('27) (RACYK), Asst. to V.P., Johns-Manville Sales Corp., 22 E. 40th St., New York 16; *for mail*, 36 Grandview Ave., Mt. Vernon, N.Y.
Alloway, Donald Miller (J'45) (GMCD), Plant & Prod. Engr., Times Publ. Co. of Oshawa, Ltd., 57 Simcoe St., South Oshawa, Ont., Can.
Allport, Hamilton ('23), 209 S. La Salle St., Chicago, Ill.
Allwein, A. Francis ('45) (BLZ), Sr. Devel. Engr., Brown Instrument Co., Wayne Junction, Philadelphia 44; *for mail*, 714 Carpenter Lane, Mt. Airy, Philadelphia 19, Pa.
Allwork, Ronald R. ('45) (CMW), Allwork Co. Inc., 30 Rockefeller Plaza, New York 20, N.Y.
Alman, Lawrence Cary ('34; '45) (CMYDL), Indus. Engr., Indus. Engrg. Dept., Kodak Park Works, Eastman Kodak Co., Rochester 4; *for mail*, 22 Bakerdale Rd., Rochester 12, N.Y.
Almirall, Juan A. ('92; '04) (COS), Chmn. of Bd., Almirall & Co., Inc., 58 Park Pl., New York 7, N.Y.
Alpaugh, John Clinton, Jr. (J'44) (AMS), Lt. (j.g.), U.S.N.R., 67 Bremond St., Belleville 9, N.J.
Alpern, Maxwell ('16) (COSFLK), Chmn. of Bd., Harmonic Reed Corp., 350 N. 16th St., Philadelphia; *for mail*, 6622 Greene St., Philadelphia 19, Pa.
Alpert, John L. (J'41) (MNCDLY), Ch. Engr., Irving Lampert Co., 236 Liberty Ave., Brooklyn 7; *for mail*, 104-21—68th Dr., Forest Hills, L.I., N.Y.
Alpert, Sumner (J'44) (AHBZSN), Mech. Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Brook Park Rd., Cleveland, Ohio.
Alsberg, Julius ('05; '19) (OYJBHS), Chief, Chemicals Unit, Tech. Indus. Intelligence Branch, Dept. of Commerce, Washington; *for mail*, 2120—16th St., N.W., Washington 9, D.C.
Alsop, Clinton E. (J'38) (NBSKJO), Mar. Engr., P-4, Naval Shipyard, San Francisco; *for mail*, 710-A Ashbury St., San Francisco 17, Calif.
Alstadt, Loring Ray (J'42) (SCLAWY), Control Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 1028 Holland Rd., Schnectady 3, N.Y.
Alt, Louis M. ('30; '41) (YMCJLD), Prod. Mgr., Rostone Corp., Earl Ave.; *for mail*, 922 S. 10th St., Lafayette, Ind.
Alteneder, Theodore George ('45) (BCJM), Pres., Gen. Mgr., Theo. Alteneder & Sons, 1217 Spring Garden St., Philadelphia 23, Pa.
Alter, Harry Albert (J'40) (NBDJYO), Mech. Engr., Belden Mfg. Co., Richmond, Ind.
Althouse, Edward George (J'34) (MJSN), Asst. Pur. Agt., Yarnall-Waring Co., 102 E. Mermaid Lane, Philadelphia 18; *for mail*, 8216 Bayard St., Philadelphia 19, Pa.
Althouse, William S., Jr. (J'38) (JMYCH), Ch. Prod. Engr., Baker Oil Tools, Inc., 6000 S. Boyle St., Los Angeles 11; *for mail*, 206 Westmont Dr., Alhambra, Calif.
Altman, Donald Charles (J'43), Field Engr., Allis-Chalmers Mfg. Co., Henry Bldg., Seattle; *for mail*, 2366 Boyer Ave., Seattle 2, Wash.
Altman, George Gustave, Jr. (J'42) (ASO), 221 E. Gaston, Savannah, Ga.
Alton, Darrel Dow ('24; '35) (REJSMH), Asst. Supt., M.P. & Equip., So. Pacific Lines in Tex. & La., Box 219, Houston 1, Tex.
Alton, David Elie (J'34) (KLHA), Pres., Horton Co., 68 Pratt St., Hartford 8; *for mail*, 727 Farmington Ave., West Hartford, Conn.
Altorfer, Hans A. (J'30) (BNEARK), Design Supvr., Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Altwein, Donald Walter (J'42) (BK), 758 Channing St., Palo Alto, Calif.
Alves, Alexander L. ('31; '36) (CHY), Asst. Gen. Mgr., Watertown Mfg. Co., 127 Echo Lake Rd.; *for mail*, 38 Sunset Ave., Watertown, Conn.
Alves, George Emmanuel (J'40) (KHL), Chem. Engr., Devel. Work, E. I. du Pont de Nemours & Co., Wilmington 98; *for mail*, Box 85, Wilmington 99, Del.
Alvung, Rolf ('41) (ERS), Mech. Engr., Chicago Great West. Ry. Co., Oelwein, Iowa.
Amadon, Frank Whitcombe ('45) (SKOLHE), Spec. Rep., Worthington Pump & Mchy. Corp., 522 Terminal Tower, Cleveland 13, Ohio.
Amato, Emanuel J. (J'41) (FMS), Lt., Co. C, 5th Engrs., Trng. Bn., Ft. Belvoir, Va.
Ambler, F. Marple (J'35) (BCM), Automotive Engr., Mack Mfg. Corp.; *for mail*, Star Route, Allentown, Pa.
Ambos, Jack F. (J'42), Apprentice Seaman, V7 (S), U.S.N.R., Midshipmen's Sch., Rm. 201, Sec. S-1, Badin Hall, Notre Dame, Ind.
Ambroff, Michel (J'38) (ONBAKH), Owner & Dir., Tech. Design Coordinator Co., 16218 Ventura Blvd., Encino; *for mail*, 1425 S. McBride Ave., Los Angeles 22, Calif.
Ambrose, Eugene R. (J'46), Am. Gas & Elec. Serv. Corp., 30 Church St., New York, N.Y.; *for mail*, 153 Lincoln Ave. E., Roselle Park, N.J.
Ambrose, Roy B. ('13; '21), Mgr. Bldgs. & Grounds, Carnegie Library & Inst., Schenley Park, Pittsburgh, Pa.
Ambrosius, Edgar Elmer ('37) (BHS), Prof. Mech. Engrg., Pa. State College, State College, Pa.
Amens, Harold C. (J'41), Surveyor, Am. Bur. of Shipping, 215 Market St., San Francisco 5; *for mail*, 886 Hilmar, Santa Clara, Calif.
Ames, Aubrey P. ('25; '30) (CEP), c/o Standard-Vacuum Oil Co., El Hogar-Filipino Bldg., Manila, P.I.
Ames, Charles Stuart, Jr. (J'42) (ACM), Process Engr., Gen. Motors Corp., 2700 Tweedy Blvd., South Gate; *for mail*, 23571 Maribel Ave., Avalon Village, Wilmington, Calif.
Ames, John Burton (J'29) (SEB), Steam Serv. Engr., Westinghouse Elec. Corp., 10 High St., Boston 10; *for mail*, 21 Kew Rd., Braintree 84, Mass.
Amey, Earl Bartley (J'43) (YTMNJW), Tech. Asst. to Officer-in-Charge Equip. Design Sec., Bur. of Ships, Navy Dept., Washington 25; *for mail*, 2716—30th St., S.E., Washington 20, D.C.
Amidon, Lee L. ('28; '34; '35) (SEJANC), Prof., Head, Dept. of Mech. Engrg., S.D. State College, Brookings, S.D.
Amneus, John Sigfrid (J'40) (BHJ), Ch., Research Design Sec., Jet Propulsion Lab., Guggenheim Aero. Lab., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4; *for mail*, 314 S. Michigan, Pasadena 5, Calif.
Amore, Jos. R. (J'36), Draftsman, Enterprise Engrg. Co., 782 Union St.; *for mail*, 158 Prospect Pk. W., Brooklyn, N.Y.
Amorosi, Alfred M. ('34; '45) (JKS), Mech. Engr., Bur. of Ships, Navy Dept., Washington; *for mail*, 1426—21st St., N.W., Washington 6, D.C.
Amos, Carl W. ('29; '35) (HBNJS), Engr., Corps of Engrs., U.S.A., Pittock Block, Portland 5; *for mail*, 1433 S.E. 30th Ave., Portland 15, Ore.
Amrein, Joseph ('29; '35), Assoc. Engr., Corps of Engrs., U.S.A., 120 Wall St., New York; *for mail*, 622 Ave. C, Brooklyn, N.Y.
Amsler, Rolf (J'44) (EAHBN), Mech. Engr., Boeing Aircraft Co., Seattle 14; *for mail*, 2224 E. 80th St., Seattle 5, Wash.
Amstead, Billy Howard (J'43) (CJM), Prod. Engr., Mission Mfg. Co., Houston; *for mail*, 4005 Ave. F, Austin, Tex.
Amstutz, James B. ('21; '27) (ABCN), Plant Supt., J. J. Tourek Mfg. Co., 4701 W. 16th St., Chicago; *for mail*, 726 S. Euclid Ave., Oak Park, Ill.
Amstuz, John O. ('31) (CDKLMT), Works Mgr., Ch. Engr., Behr-Manning Corp., Troy, N.Y.
Anastasi, Nunzio J. (J'38), 10 Monroe St., New York, N.Y.
Anbro, Gosta Anderson ('24; '35) (SCF), Power Engr., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City 2, N.J.
Ancell, Robert M. (J'46), Mech. Engr., Natl. Advis. Com. for Aero., Mun. Airport; *for mail*, 4477 W. 137th St., Cleveland, Ohio.
Ancona, John F. ('04; '12) (OBSHAL), Cons. Engr., Hiram Sibley Bldg., 311 Alexander St., Rochester 7, N.Y.
Andara, J. L. (J'43), Compania Venezelana de Cementos, Maracaibo, Venezuela, S.A.
Andeen, John Wallace (J'37) (CDS), Lt., Ord. Dept., U.S.A., Project Officer, Office of Chief of Ord., Union Guardian Bldg., Detroit 32; *for mail*, 15369 Steel St., Detroit 27, Mich.
Anderegg, Ruben H. ('20; '25; '35) (KLB), V.P., Ch. Engr., Trane Co.; *for mail*, 450 Losey Court, La Crosse, Wis.
Andersen, Arne Thorvald (J'42) (FKSH), Designer, Day & Zimmermann, Inc., Packard Bldg., Philadelphia; *for mail*, Apt. 4, 2312 Green St., Philadelphia 30, Pa.
Andersen, Edward Anthony (J'43) (FKS), Asst. Engr., Philadelphia Elec. Co., 28th & Christian Sts., Philadelphia; *for mail*, 1117 Yeadon Ave., Yeadon, Pa.
Andersen, Henry C. (J'36) (NBM), Mech. Engr., Gear Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail*, 71 Park Ave., Wakefield, Mass.
Andersen, James Russell (J'44) (KBA), Instr. Mech. Engrg., Univ. of Pa., Philadelphia 4, Pa.
Anderson, Albert William (J'37) (AFEZS), Univ. of Minn., 2013 University Ave., S.E.; *for mail*, 1100—23rd Ave., S.E., Minneapolis 14, Minn.
Anderson, Alex. (J'43), 110 Craiglea Dr., Edinburgh, Scotland.
Anderson, Alfred Andrew (J'38) (JMCWA), Tool Engr., Kennametal, Inc., Latrobe, Pa.; *for mail*, N. Rolling Rd. & Powers Lane, Baltimore 28, Md.
Anderson, Arnold Sigfrid (J'42) (ACM), Lt., Aircraft Lab., Engrg. Div., Wright Field, Dayton, Ohio.
Anderson, Arthur Walter ('32; '35) (BEHKSM), Ch. Engr., Army Trans. Corps, N.Y. Port of Embarkation, Brooklyn; *for mail*, 2 Grace Court, Brooklyn 2, N.Y.
Anderson, Arvid Edwin (J'41), 129 Bonner St., Hartford 6, Conn.
Anderson, Bror Gerhard ('27; '35) (ABECSN), Eng. Designer, Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail*, 291 Tryon St., South Glastonbury, Conn.
Anderson, C. Edward ('27) (AEFJKS), Prof. Mech. Engrg., Engrg. Bldg., Univ. of Wyo., Laramie, Wyo.
Anderson, C. Einar ('19) (HMB), 50 Milford Ave., Newark 8, N.J.
Anderson, Carl G. ('39) (BHCOZ), Ch. Engr., Chicago Metal Hose Corp., 840 N. State St., Elgin, Ill.
Anderson, Carl Merlin (J'42) (BSM), Research Engr., Elliott Co.; *for mail*, 415 S. Wittenberg Ave., Springfield, Ohio.
Anderson, Carl O. ('21; '27; '35), Designer, Internatl. Business Mchs. Corp., North St.; *for mail*, 1902 Monroe St., Endicott, N.Y.
Anderson, Carl Wallace (J'42) (ELS), K-4, Rm. 111, Richland, Wash.

Anderson, Charles Henning, Jr. (J'43) (NBS), Pvt., Ord. Translation Center, Ft. Eustis, Va.; *for mail,* 2912 Pyramid Ave., Pittsburgh 10, Pa.
Anderson, Clarence Russell (J'39) (NHYMSC), Drafting Rm. Supvr., Hydro-Dynamics Lab., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4; *for mail,* 4552 W. 17th St., Los Angeles 6, Calif.
Anderson, Clifford C. ('39) (CEFS), Engr., United Light & Power Serv. Co., United Light Bldg., Davenport, Iowa.
Anderson, David Leonard ('28) (MNGCY), Asst. Ch. Engr., Intertype Corp., 360 Furman St., Brooklyn 2; *for mail,* 154—73rd St., Brooklyn 9, N.Y.
Anderson, Donald Charles (J'44), 202 Colorado, Highland Park 3, Mich.
Anderson, Duane Fredrick (J'43) (BZAJ), Devel. Engr., Aero. & Mar. Eng. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1121 Parkwood Blvd., Schenectady 8, N.Y.
Anderson, E. E. ('20; '38) (CMAJ), Works Mgr., Goddard & Goddard Co., 12280 Burt Rd., Detroit 23; *for mail,* 351 Yarmouth St., Birmingham, Mich.
Anderson, Edward Alberg (J'42) (BZJDSH), Asst. to Supt., Blast Furnaces, South Works, Carnegie-Ill. Steel Corp.; *for mail,* 10712 Ave. F, Chicago 17, Ill.
Anderson, Edwin L. ('41; '44) (OACM), Asst. Constr. Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 63 Oakridge Rd., Verona, N.J.
Anderson, Einar F. ('30; '35), Cia. Argentina de Cemento Portland, Parana, Entre Rios, Argentina, S.A.
Anderson, George Philip ('35) (YCHMLO), Dir., Plastics Mchy. Div., Hyd. Press Mfg. Co., Marion Rd., Mt. Gilead; *for mail,* 573 Andover Rd. W., Mansfield, Ohio.
Anderson, Gotthard E. (J'24) (SKFECJ), Asst. Mgr., Steam Div., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Anderson, Gustave A. ('20; '35) (DMNT), Mech. Engr., Internatl. Handkerchief Co., 136th St. & Willow Ave., New York 54, N.Y.
Anderson, Gustave William (J'43) (CJM), Southampton, Bucks Co., Pa.
Anderson, Harry A. ('36) (CDM), Plant Engr., Footwear Plant, U.S. Rubber Co., Maple St.; *for mail,* 55 Dunn Ave., Naugatuck, Conn.
Anderson, Howard Milton (J'41) (ACNKSE), Custodian Engr., Bd. of Education, 133 St. Nicholas Ave., New York 26; *for mail,* 319 Pearl St., Brooklyn 1, N.Y.
Anderson, Hubert Cecil, Jr. (J'46), So. States Equip. Corp., Hampton, Ga.
Anderson, James A. ('23) (CDFMRS), Mem., Natl. R.R. Adjustment Bd., 220 S. State St., Chicago 4, Ill.
Anderson, James Hilbert ('45) (EBZAKH), Research Engr., Ingersoll-Rand Co., Phillipsburg, N.J.
Anderson, James Louis ('45) (JOS), Mech. Engr., 2421 Live Oak St., Huntington Park, Calif.
Anderson, James Walker, Jr. ('42) (CMDJ), Pres., Gen. Mgr., Monarch Governor Co., 1832 W. Bethune St.; *for mail,* 2526 Longfellow St., Detroit 6, Mich.
Anderson, Jasper Eugene ('30; '37) (BMJAD), Asst. Supt., Gear Plant 2, Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 76 Lincoln Ave., Saugus, Mass.
Anderson, John Arnold (J'39) (CEHKSY), Asst. Mgr., Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.
Anderson, John Norman ('14) (NDB), Mech. Engr., Otis Elev. Co., 260—11th Ave., New York 1, N.Y.; *for mail,* 905 Castle Point Terrace, Hoboken, N.J.
Anderson, John Wallace ('13; '19) (EBMNSK), Exec. Engr., Atlas Imperial Diesel Eng. Co., 1000—19th Ave., Oakland 6, Calif.
Anderson, Kenneth B. ('24; '36) (FZBCDE), Gas Supt., Coast Counties Gas & Elec. Co., 22 Pacific Ave., Santa Cruz, Calif.
Anderson, L. Douglass ('06; '13) (DLSO), Cons. Engr., Potash Co. of Am., Box 31, Carlsbad, New Mex.
Anderson, Marion Malone (J'44) (BCM), Box 145, Bowling Green, Ky.
Anderson, Melvin Travis (J'37) (KLMCG), Product Engr., Fansteel Metal. Corp., North Chicago, Ill.
Anderson, Niles Milton ('43) (CWBL), V.P., Mgr., Marathon Paper Mills of Can. Ltd., 320 Bay St., Toronto 1; *for mail,* 71 Old Forest Hill Rd., Toronto 12, Ont., Can.
Anderson, Nils H. ('32; '35) (MBDJLS), Plant Engr., Carboloy Co., Inc., Box 237, Roosevelt Park P.O., Detroit 32, Mich.
Anderson, Nils, Jr. (J'39) (YWA), 350 Madison Ave., New York 17, N.Y.; *for mail,* 1100 Pequot Rd., Southport, Conn.
Anderson, Norman Kenneth (J'40) (CMNH), Pvt., 1st Class, 2506th Serv. Comd. Unit, 17th Sig. Serv. Co., So. Post, Ft. Myer, Va.; *for mail,* 854 Gartner Court, Beloit, Wis.
Anderson, Oscar A. ('35) (CDS), Asst. Ch. Engr., Armour & Co., Union Stock Yards; *for mail,* 9830 S. Hamilton Ave., Chicago, Ill.
Anderson, Otto Holm ('23) (CMR), V.P., Ch. Engr., Natl. Steel Car Corp. Ltd., Hamilton, Ont., Can.
Anderson, Paul E. (J'37) (CMJ), Ch. Engr., Bradley Mfg. Co., 1221—2nd Ave., Rock Island; *for mail,* 4 N. Oak St., Hinsdale, Ill.
Anderson, Paul Washburn ('46), Cons. Engr., 140 Nassau St., New York 7; *for mail,* 378 Central Pk. W., New York 25, N.Y.
Anderson, Richard Terhune ('19; '20) (CDFMS), V.P., Gen. Mgr., Paterson Parchment Paper Co., Bristol, Pa.; *for mail,* Cold Soil Rd., Lawrenceville, N.J.
Anderson, Robert Caldwell (J'44) (LKMZC), Maint. Engr., Natl. Biscuit Co., 164 Forest Ave.; *for mail,* 164 Danforth St., Portland, Me.
Anderson, Robert Carl ('35; '45) (MCD), Prod. Engr., Sparrows Point Shipyard, Bethlehem Steel Corp., Sparrows Point; *for mail,* 2905 Dunran Rd., Baltimore 22, Md.
Anderson, Robert Charles (J'43) (AES), Ensign, U.S.N.R., 1321—11th Ave., Greeley, Colo.
Anderson, Robert Tyson ('23; '35) (SFCKZ), Engr., Charge Grounds & Bldgs., Girard College, Corinthian & Girard Aves., Philadelphia 21, Pa.
Anderson, Roger Scott (J'40) (ZMKSD), Sr. Draftsman, Carbide & Carbon Chemicals Corp., Box P, Oak Ridge; *for mail,* 110 Pine Rd., Norris, Tenn.
Anderson, Roy Frederick ('44) (EFCRSN), Sales Mgr. & Engr., Lorimer Diesel Eng. Co., Inc., 1530 Wood St., Oakland 7; *for mail,* 1071 Portola Dr., San Francisco 16, Calif.
Anderson, Thomas ('24; '35) (FNEKJB), Supt., Mech. Prod. Research Div., Cities Serv. Oil Co., 226 Highway 29, Hillside; *for mail,* 14 Clover St., Elizabeth 2, N.J.
Anderson, Vernal A. (J'42) (ABM), Jr. Engr., Brake Engrg. Dept., Bendix Products Div., Bendix Aviation Corp., 401 Bendix Dr.; *for mail,* 515 W. Marion St., South Bend, Ind.
Anderson, William Benton (J'41) (EAMCFJ), Maj., Army Air Forces, Postgraduate Student, Engrg. Sch., Wright Field, Dayton, Ohio; *for mail,* 629—12th Ave., Brookings, S.D.
Anderson, William Edward ('39) (HCKBM), Sales Mgr., Secy., Pa. Pump & Compressor Co.; *for mail,* 2004 Washington Blvd., Easton, Pa.
Anderton, Earl F. (J'38), Devel. Engr., Scott Paper Co., Chester; *for mail,* R.F.D. 1, Media, Pa.
Andes, Ammon S. (J'43) (AEBFJN), Test Engr., Consltd. Vultee Aircraft Corp., Grants Lane, Ft. Worth; *for mail,* Box 5476, Ft. Worth 8, Tex.
Andreini, Joseph I. (J'43) (MCAD), Prod. Mgr., Republic Aviation Corp., Conklin St., Farmingdale; *for mail,* 4 Locust St., Massapequa, L.I., N.Y.
Andres, Chas. S. ('39) (CDKLMS), Supt., Lone Star Cement Corp., Hudson, N.Y.
Andres, George Jerome (J'44) (YONBLH), Draftsman, Bakelite Corp., Bound Brook; *for mail,* 208 N. 3rd Ave., Highland Park, N.J.
Andres, Stanley Joseph (J'44) (AHJ), 4 Haight Ave., Poughkeepsie, N.Y.
Andres, William Albert (J'41) (MCHJNK), Products Engr., Gen. Mills, Inc., 1620 Central Ave., Minneapolis 13; *for mail,* Route 4, Minneapolis 9, Minn.
Andresen, Robert Louis (J'40) (CMDL), Installation Engr., George S. May Co., 235 Montgomery St., San Francisco; *for mail,* 3648—22nd St., San Francisco 14, Calif.
Andrew, James Dhu, Jr. ('43) (FKS), Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York, N.Y.; *for mail,* Old Chester Rd., Essex Fells, N.J.
Andrew, Lowell Ross ('25; '35) (FJMC), Mech. Engr., Medart Co., 3500 De Kalb St., St. Louis; *for mail,* 6220 Michigan St., St. Louis 11, Mo.
Andrew, Maurice Benedict ('32; '43) (BHJNO), Design Engr., U.S. Engr. Office, 1209—8th St.; *for mail,* 1300—9th St., Sacramento, Calif.
Andrew, Percy J. ('37), Engr., Dome Mines, South Porcupine, Ont., Can.
Andrew, William Garcia, Jr. (J'42), Ensign, *U.S.S. Philip,* c/o Postmaster, New York, N.Y.
Andrews, Albert Henry (J'42) (MCN), Planner, Mch. Shop 6, Bethlehem Steel Co., E. 3rd St.; *for mail,* 1158 W. Rosemont Dr., Bethlehem, Pa.
Andrews, Bernard Russell ('17; '35) (TNLC), Treas., Andrews & Goodrich, Inc., 336 Adams St., Dorchester, Boston, Mass.
Andrews, Carroll Adreon (J'46), 4022 Ridgecroft Rd., Baltimore 6, Md.
Andrews, Earle Edwin, Jr. (J'43) (WCYJL), Lt. (j.g.), U.S.N.R.; *for mail,* 196 Highland Ave., Winchester, Mass.
Andrews, Edward Vail ('20) (CSYE), Supvr., Parts Dept., Nordberg Mfg. Co., Oklahoma & Chase Aves.; *for mail,* 2853 S. Mabbett Ave., Milwaukee 7, Wis.
Andrews, Henry Ivan (J'29), Library, Research Dept., London, Midland & Scottish Ry. Co., London Rd., Derby, Derby, England.
Andrews, John Hampton (J'45) (CMRBSH), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.; *for mail,* Collingwood Rd., Bridgeport 4, Conn.
Andrews, John Taylor, Jr. (J'38), Capt., U.S.A., Hdq., 182nd Ord. Bn., A.P.O. 875, c/o Postmaster, N.Y.; *for mail,* 1403 Summit Ave., Catonsville 28, Md.
Andrews, Ralph John, Jr. (J'41) (ACFDKR), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., Main St., East Hartford 8; *for mail,* 26 Gillett St., Hartford 5, Conn.
Andrews, Ray Clark (J'41) (CJR), Asst. Roll Designer, Steelton Plant, Bethlehem Steel Co.; *for mail,* 363 Locust St., Steelton, Pa.
Andrews, Raynal Wride, Jr. ('44) (BJH), Div. Head, Cent. Mech. Engrg. Dept., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19, Pa.
Andrews, Roger W. (A'36) (CFKS), Gen. Sales Mgr., Riley Stoker Corp., 9 Neponset St., Worcester 1; *for mail,* 12 Kinnicutt Rd., Worcester 5, Mass.
Andrews, Samuel Warren ('26) (BEFHLS), Pres., H. G. Acres & Co., 2135 Culp St., Niagara Falls, Ont., Can.
Andrews, Zenas Ball (J'40) (NHECA), Lt. Comdr., U.S.N., Submarine Sqd. 6, Balboa, C.Z.
Andriola, Achilles D. (J'35) (BESOMO), Mar. Engr., Cramp Shipbldg. Co., Philadelphia; *for mail,* 1032 Drexel Ave., Drexel Hill, Pa.
Andrix, Earl Russell ('09; '16; '35) (BCJSME), Prof. Mech. Engrg., Tri-State Angola-Ind. College; *for mail,* 211 N. Wayne St., Angola, Ind.
Andromidas, Thomas Theodore (J'42) (SHKZE), Sr. Designer, United Engrs. & Constructors Inc., 1401 Arch St., Philadelphia; *for mail,* 4417 Fleming St., Philadelphia 28, Pa.
Andrus, Robert Emmett (J'44), Mech. Engr. in Research, Texas Co.; *for mail,* 134 Main St., Beacon, N.Y.
Angarano, Joseph A. (J'39), Asst. Engr., Crucible Steel Co. of Am., Yale Ave., Jersey City; *for mail,* 257 Grant Ave., Cliffside, N.J.
Angel, Theodore James (J'40) (ESCOMZ), Application Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7; *for mail,* Apt. 106, 5615 W. Martin Dr., Milwaukee 8, Wis.
Angell, Eugene N. (J'33) (BDL), Narrows, Va.
Angemeer, Charles William (J'46) (ABJ), Designer "C," Natl. Tube Co.; *for mail,* Box 43, Finleyville, Pa.
Angier, Edward Herbert (A'17) (CLNOY), Pres., Angier Corp., 50 Fountain St., Framingham, Mass.
Angstadt, J. Walton (J'38) (LBKHE), Field Engr., Buffalo Fdy. & Mch. Co., 1543 Filmore Ave., Buffalo; *for mail,* 50 Palmer Ave., Kenmore 17, N.Y.
Angstadt, Richard Rueben (J'42) (JLNKM), Lt. (j.g.), U.S.N.R., Naval Ord. Office, 1030 E. Green St., Pasadena 1, Calif.
Angus, Donald Lloyd (J'41) (OSKEF), Assoc., H. H. Angus, Cons. Engrs., 1221 Bay St., Toronto 5; *for mail,* 102 St. Hubert Ave., Toronto, Ont., Can.
Angus, Harry Holborn ('19) (OKSF), Cons. Engr., 1221 Bay St., Toronto 5; *for mail,* 34 Farnham Ave., Toronto, Ont., Can.
Angus, Robert William ('01, '08; F'36; H'40) (HSEBKN), *Vice-President,* '24-'26; Prof. Emeritus Mech. Engrg., **Mech. Bldg., Univ. of Toronto,** Toronto, Ont., Can.
Angus, Wm. J. ('46) (FKS), Mech. Supt., Consltd. Edison Co. of N.Y., Inc., 666—1st Ave., New York; *for mail,* 42-12—158th St., Flushing, L.I., N.Y.
Angus, William John, Jr. (J'45), 723 E. 42nd St., Brooklyn 3, N.Y.
Anheier, Arthur Loal (J'41), 533 W. 95th St., Los Angeles, Calif.
Anisman, George Frederick (J'44) (AKCH), Aerodynamicist, N. Am. Aviation, Inc., Imperial Highway, Inglewood; *for mail,* 547 N. Kings Rd., Los Angeles 36, Calif.
Ankstitus, John Peter (J'41) (CDJ), 19 Ward St., Worcester, Mass.
Annett, Edward B. ('14) (EFS), Sr. Engr., Pub. Utility Comm., 1060 Broad St., Newark 2; *for mail,* Box 56, 78 Greenwood Dr., Millburn, N.J.
Annin, Joseph Burt (J'43), Research Engr., Natl. Advis. Com. for Aero. Lab.; *for mail,* 1200 Elbur, Lakewood, Ohio.
Annis, Russell K. ('23; '31) (BHM), Cons. Engr., Pump Designer, 1978 Edgemont Rd., Columbus 8, Ohio.
Ansel, Harry Russell ('42) (CJL), Secy., Treas., Wm. J. Stange Co., 2536 W. Monroe St., Chicago 12, Ill.; Pres., Treas., Kor-Lok Co., Cleveland, Ohio; *for mail,* 300 S. Ridgeland Ave., Oak Park, Ill.
Antalec, Frank Andrew (J'43) (CMJB), Prod. Control, Bates Mfg. Co., 18 Central Ave., Orange; *for mail,* 47 Walnut Court, South Orange, N.J.
Anthony, Graham H. ('16; '25), Pres., Veeder-Root Inc., 20 Sargeant St., Hartford, Conn.
Anthony, James T. ('17) (FKR), V.P., Gen. Refractories Co., 1600 Real Estate Trust Bldg., Philadelphia, Pa.
Anthony, Richard L. ('27; '31; '35) (SC), Prof. Mech. Engrg., Rutgers Univ., New Brunswick, N.J.
Anthony, William Cornelison ('42) (CDHMNO), Pres., Anthony Co.; *for mail,* 110 W. Bluff St., Streator, Ill.
Anthony, William O. (J'41) (ACS), Ch. Engr., Reynolds Wire Co.; *for mail,* 310 N. Ottowa St., Dixon, Ill.
Antonsanti, Louis ('16; '35) (CHM), Pres., L. Antonsanti, Inc., Ponce, Ponce, P.R.
Antony, Charles, Jr. ('45; '45) (ABJYNM), Design Engr., Research Lab., Sperry Gyroscope Co., Clinton Rd., Garden City; *for mail,* 10 Wellesley Rd., Rockville Centre, N.Y.
Anzelon, George Joseph (J'41) (MCDNOA), Mfg. Engr., West. Elec. Co., Inc., Central Ave., South Kearny, N.J.; *for mail,* 580 E. 22nd St., Brooklyn 26, N.Y.

Apelgrain, Merl D. ('44), Pres., Apelgrain Engrg. Co.; *for mail*, Osakis, Minn.
Apgar, James William (J'40) (HBNEKF), Automotive Engr., Army Serv. Forces, Aberdeen Proving Ground; *for mail*, 260 Seneca Ave., Havre de Grace, Md.
Apitz, Russell Frank (J'44), Ensign, U.S.N.R., YMS-285, F.P.O., San Francisco, Calif.
Apolis, John Joseph (J'40) (NJHSBA), Radio Technician, U.S.N.R.; *for mail*, 225 Bowen St., South Boston, Mass.
Appe, Raymond Clifford (J'43) (EOSKBF), Lt., U.S.N., LSM-436, F.P.O., San Francisco, Calif.; *for mail*, 25 Hillcrest Rd., Maplewood, N.J.
Appel, William Joseph (J'41) (AKJFLG), Sales Engr., Bldg. Matls. Div., Armstrong Cork Co., W. Liberty St.; *for mail*, 1120 Pleasure Rd., Lancaster, Pa.
Appelt, Leonard (J'41), 4622 S. Paulina St., Chicago, Ill.
Appelt, Leslie L. (J'46) (GJK), Maj., Corps of Engrs., 1820 Branard St., Houston, Tex.
Appelton, Harry (J'45) (YZDCOK), Asst. Mech. Engr., Viceroy Mfg. Co., 345 Royce Ave.; *for mail*, 243 Beverly St., Toronto, Ont., Can.
Apperson, John Samuel ('21), Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Apperson, John Samuel, III (J'41), Gen. Elec. Co., 4966 Woodland Ave., Cleveland, Ohio.
Applebaum, Samuel Bernard ('21; '35) (STR KL), V.P., Permutit Co., 330 W. 42nd St., New York 18; *for mail*, 25 W. 54th St., New York 19, N.Y.
Applegate, Francis R. ('35; '41) (CFK), Plant Service Engr., Pratt & Whitney Aircraft, 95th & Holmes, Kansas City; *for mail*, 5129 Wornall Ave., Kansas City 3, Kan.
Applegate, Walter F. (J'36) (SE), Engr., Power Design, Hercules Powder Co., 909 Market St., Wilmington 99, Del.; *for mail*, 114 Rockland Rd., Merion Station, Pa.
Appleton, Glenn Ray (J'42), Draftsman, Babcock & Wilcox Co.; *for mail*, 668 Summit St., Barberton, Ohio.
Appley, Lawrence A. (A'41) (C), V.P., Personnel & Pub. Relations, Montgomery Ward & Co., 619 W. Chicago Ave., Chicago 10, Ill.
Appleyard, John S. (J'33) (CMB), Engrg. Dept., Marlin-Rockwell Corp.; *for mail*, 5 Hough St., Plainville, Conn.
Apt, Sanford R. ('26) (BDHOS), Mech. Engr., Parsons, Brinckerhoff, Hogan & Macdonald, 142 Maiden Lane, New York 7; *for mail*, 36-39—205th St., Bayside, N.Y.
Aptulsky, Walter (J'45) (NKSFEC), Jr. Engr., Furnace Design, Corning Glass Works, Corning, N.Y.
Arai, Tom (J'42), Pvt., Hdq. Btry., 522nd Field Arty. Bn., Camp Shelby, Miss.
Arashiro, Nicholas N. (J'41) (ABH), Stress Engr., A. O. Smith Corp., Milwaukee; *for mail*, Apt. 6, 2529 E. Webster Pl., Milwaukee 11, Wis.
Arata, George V. (J'40) (HKB), Asst. Design Engr., Mar. Prod., Pelton Water Wheel Co., 2929—19th St., San Francisco 10; *for mail*, 159 Bella Vista, San Francisco 16, Calif.
Arbuckle, Lawrence Lee ('44) (CKO), Bell & Gossett Co., Morton Grove; *for mail*, 1631 Estes Ave., Chicago 26, Ill.
Arbuckle, Tom Edward, Jr. (J'41), Maint. Engr., Southwest Grain Co., Box 302; *for mail*, 2110 N. Adams St., Hutchinson, Kan.
Archea, Walter David ('19; '35) (BCJM), Design Supvr., Cincinnati Milling Mch. Co., Marburg & Disney St., Cincinnati 9; *for mail*, 6621 Iris Ave., Cincinnati 13, Ohio.
Archer, Arthur Albert ('42) (FKLDY), Ch. Engr., Pittsburgh Coal Carbonization Co., 1243 Oliver Bldg., Pittsburgh 30; *for mail*, 208 Mohawk Dr., Pittsburgh 16, Pa.
Archer, Edward Gibson ('31) (CDLNTY), Research & Design Engr., Tubize Div., Celanese Corp. of Am., Chatillon Rd.; *for mail*, 124 Chatillon Rd., Rome, Ga.
Archer, W. Harry ('36) (DLS), 4516 Modoc Rd., Santa Barbara, Calif.
Archibald, Francis Littlefield ('30; '43) (SFK HAB), Engr., Boston Edison Co., 39 Boylston St., Boston 12; *for mail*, 26 Bradlee Rd., Medford 55, Mass.
Aréchiga, Leopoldo Eugenio ('39) (EAHKS), Ch. Engr., C. Madero Refinery, Petroleos Mexicanos, Box 161, Tampico, Tamaulipas, Mex.
Arents, Chester Abbo ('44), Asst. Prof., Mech. Engrg. Dept., Ore. State College; *for mail*, 809 S. 6th St., Corvallis, Ore.
Arentzen, Einar Martin ('27; '35) (CDM), Pres., Gen. Mgr., Lee-Norse Co., Charleroi, Pa.
Arenz, Robert James (J'45) (AJN), Stress Analyst, Douglas Aircraft Co., Inc.; *for mail*, 958—4th St., Santa Monica, Calif.
Arey, Homer Reynor (J'42) (OAHK), Sales Engr., Gen. Elec. Co., 5 Lawrence St., Bloomfield; *for mail*, 227 N. Walnut St., East Orange, N.J.
Argabrite, Adelph Wayne (J'41) (SYJMDZ), 730 Jackson St., Denver 6, Colo.
Argersinger, John Irvin (J'38) (SFKEJZ), Calculating Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 54 Carthage Rd., Scarsdale, N.Y.
Argust, Frank Funk (J'43) (CMODAB), Mfg. Engr., Lockheed Aircraft Corp., Burbank, Calif.
Arias, Enrique Ricardo (J'40) (LKFOZ), Asst. Tech. Dir., Jose Arechabala, S.A., Edificio Arechabala; *for mail*, Apartado 2856, Cardenas, Matanzas, Cuba.
Arkoosh, Fred George (J'38) (CME), Capt., Ord. Dept., Officer in Charge, Omaha Regional Office, St. Louis Ord. Dist., 109 S. 10th St.; *for mail*, 3532 N. 47th Ave., Omaha, Neb.
Arledge, W. F., Jr. (J'41) (CDP), Indus. Salesman, Texas Co., Box 6171, Dallas 2; *for mail*, Box 234, Grapevine, Tex.
Arlt, Waldemar Paul (J'39) (NOBHLA), Engr., Bridge Design, Bridge Dept., Utah State Rd. Comm., State Capitol, Salt Lake City; *for mail*, 3352 S. 1940 East St., Salt Lake City 5, Utah.
Arm, David Lehr ('40) (BCN), Prof., Head, Dept. of Mech. Engrg., Iowa State College, Ames, Iowa.
Armacost, Wilbur Hering ('17; '21; '35) (SFK WJB), V.P., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 26, N.Y.
Armitage, Joseph Bradley ('19) (MBHJZ), V.P., Charge Engrg., Kearney & Trecker Corp., 6784 W. National Ave., Milwaukee 14, Wis.
Armor, Merritt K. (J'39) (CYS), Assoc. Elec. Engr., War Dept., Office of Ch. Sig. Officer, The Pentagon, Washington 25, D.C.; *for mail*, Marshallton, Del.
Armour, James Wallace ('20; '25; '30) (SFKE), Dist. Mgr., Riley Stoker Corp., 101 Walker St., Detroit 7; *for mail*, 335 Rivard Blvd., Grosse Pointe 30, Mich.
Arms, John H. R. ('23; '31) (ABC), Secy., Gen. Mgr., United Engrg. Trustees, Inc., 29 W. 39th St., New York 18, N.Y.
Arms, Merton H. ('31) (NMCJW), Ch. Engr., Bryant Chucking Grinder Co.; *for mail*, 190 Summer St., Springfield, Vt.
Arms, Richard Philip (J'39) (ABNSKF), Engr., Aircraft Gas Turbine Engrg. Div., River Works, Gen. Elec. Co., West Lynn, Mass.
Armstrong, Cecil E. (J'35) (ABY), Gen. Mgr., Armstrong Plastics Co., Box 428, Warsaw; *for mail*, Burket, Ind.
Armstrong, Charles A. Newton ('43) (SJFKO), Supvg. Engr. Inspr., Ocean Accident & Guarantee Corp., 620 Brandeis Theater, Omaha 2, Neb.
Armstrong, Clarence E. ('22; '35), 38 Morton Pl., East Orange, N.J.
Armstrong, Edward (J'42) (MDC) U.S. Navy; 526 Thatcher Ave., River Forest, Ill.
Armstrong, Edward R. ('16), 6023 Drexel Rd., Overbrook, Philadelphia 31, Pa.
Armstrong, Edward T. (J'38) (BAJ), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus, Ohio.
Armstrong, Edwin H. (Non-Member), *Holley Medallist*, '40; Prof. Elec. Engrg., Columbia Univ.; *for mail*, River House, 435 E. 52nd St., New York, N.Y.
Armstrong, George Simpson ('13; '21) (CEJL OY), Pres., George S. Armstrong & Co., Inc., 52 Wall St., New York 5, N.Y.
Armstrong, Gordon ('42) (CL), Owner, Gordon Armstrong Co., 315 Bulkley Bldg., Cleveland 15, Ohio.
Armstrong, Homer Thomas (J'42) (AEFHCO), Design Engr., Link Aviation Devices, Inc.; *for mail*, 1269 Chenango, Binghamton, N.Y.
Armstrong, John T. (J'42), 104 Farm St., Providence, R.I.
Armstrong, Paul L. ('37) (HSEKLB), Union Steam Pump Sales Co., 1709 W. 8th St., Los Angeles 14, Calif.
Armstrong, Robert E. (J'35) (HLO), Asst. Engr., Metro. Water Dist. of So. Calif., 306 W. 3rd St., Los Angeles 13, Calif.
Armstrong, William Marvin ('94; '16) (BRAC), Treas., Merckens Chocolate Co., Inc., 506-20—7th St., Buffalo 1, N.Y.
Arndt, Gilbert Edward (J'42), Ensign, E-V(S), U.S.N.R.; *for mail*, 5 Hanns Ave., Vallejo, Calif.
Arnerich, Paul F. (J'32) (ACH), Engr., Douglas Aircraft Co., Inc., Santa Monica; *for mail*, 25 Annandale Rd., Pasadena, Calif.
Arnett, Robert Reiff (J'27) (CJMADL), Secy., Am. Art Alloys, Inc., 1142 S. Main St.; *for mail*, 115 N. Indiana St., Kokomo, Ind.
Arney, Thomas Robert (J'45) (EOSNAB), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 984 Vischer Ave., Schenectady 6, N.Y.
Arnold, Alvin William ('44) (BCHKMN), Mech. Engr., Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove; *for mail*, 8243 N. Tripp Ave., Skokie, Ill.
Arnold, Arthur A. ('41) (CDJ), 56 Greenway St., Hamden, Conn.
Arnold, Brigham Alicen (J'44), 1441 E. 5th St., Tucson, Ariz.
Arnold, Charles Becker ('26; '36) (SF), Asst. Div. Engr., Mech. Engrg. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York; *for mail*, 15 Witley Court, Hempstead, N.Y.
Arnold, DeWitt Russell (A'42), V.P., Stand. Ry. Equip. Co.; *for mail*, 247 Park Ave., New York, N.Y.
Arnold, Edwin E. ('00; '06) (BER), University Club, 123 University Pl., Pittsburgh, Pa.
Arnold, George ('04) (JNHRD), Mech. Engr., Cleveland Frog & Crossing Co., 6917 Bessemer Ave., Cleveland 4; *for mail*, 3565 Lytle Rd., Shaker Heights, Cleveland 22, Ohio.
Arnold, Grayson Bert (J'45) (BHKNFJ), Engrg. Asst., Los Angeles Dept. of Water & Power, 202 S. Broadway; *for mail*, 100 S. Edinburgh Ave., Los Angeles 36, Calif.
Arnold, Harry Monroe (J'36) (SFE), Jr. Mech. Engr., Bur. of Mines, Box 547, Golden; *for mail*, Apt. 8, 1837 Grant St., Denver, Colo.
Arnold, John Wendell (J'37) (ADCJNE), Mech. Design Engr., Plant A-1, Lockheed Aircraft Corp.; *for mail*, 314 E. Alameda Ave., Burbank, Calif.
Arnold, Jonah S. (J'45) (KOBZHG), Estimating Engr., Stuart Cooling Corp., 153 E. 26th St., New York 10; *for mail*, 219 S. 9th St., Brooklyn 11, N.Y.
Arnold, Phillip Justin ('44) (CGJLMZ), Test Co-ordinator, Otis Elev. Co., 44 Wells Ave., Yonkers 1; *for mail*, 138 McLean Ave., Yonkers 5, N.Y.
Arnold, Richard Michael ('44) (FOSBK), Dir., Smoke Abatement Comm., City of Louisville, 302 City Hall; *for mail*, 1135 S. 2nd St., Louisville, Ky.
Arnold, William (J'38) (CGM), Statistical Dept., Kodak Pk. Works, Eastman Kodak Co., Rochester; *for mail*, 20 Upton Park, Rochester 7, N.Y.
Arnoldi, Robert Alfred (J'42) (BAS), Faculty Instr., Physics Dept., Stevens Inst. of Tech., 5th & River Sts.; *for mail*, 604 River St., Hoboken, N.J.
Arnow, Saml. M. ('38) (HKS), Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia, Pa.
Arnstein, Karl ('25; F'38) (ABJCFR), V.P., Charge Engrg., Goodyear Aircraft Corp., 1210 Massillon Rd., Akron 1, Ohio.
Arnstein, Leonard A. ('15; '21) (C), Mech. Draftsman, Dept. of Pub. Works, Bur. of Arch., Div. of Engrg., Rm. 1400, Municipal Bldg., New York, N.Y.
Aronoff, Daniel Joseph (J'43) (JHBL), Apt. 2B, 330 S. 4th St., Richmond, Calif.
Aronoff, Leonard (J'42), 1258—40th St., Brooklyn, N.Y.
Aronofsky, Julius Solomon (J'45) (ABK), 26 E. 80th St., New York, N.Y.
Aronson, Carl Arthur (J'41) (MCDNOZ), Sec. Leader, Product Quality Sec., Transmitter Div., Gen. Elec. Co., Thompson Rd., Syracuse, N.Y.
Arrigo, Vincent Paul (J'40) (ACMGLD), Planning Engr., Airplane Div., Curtiss-Wright Corp., Columbus; *for mail*, 2311 Sherwood Rd., Columbus 9, Ohio.
Arthur, Paul David (J'44), Pvt., Hdq., 4th A.D., A.G. Sec., Classification, A.P.O. 254, c/o Postmaster, New York, N.Y.
Arthur, Wilson E. (J'38) (KJFY), Design & Constr. Engr., Ehrhart & Arthur, 417 S. Hill St., Los Angeles, Calif.
Artley, Will H. (A'29), Pres., Artley Co., 504 E. Bay St., Savannah, Ga.
Artner, Wilbur Joseph (J'44) (BEMN), Engrg. Officer, Naval Air Corps, VR-2, Alameda, Calif.; *for mail*, 1772 Lee Rd., Cleveland Heights 18, Ohio.
Arvidson, Donald Nels (J'42) (CMS), 732—13th Ave., Prospect Park, Pa.
Asbury, Abner D. ('37) (TDLMOY), Head, Textile Dept., J. E. Sirrine & Co., S. Main St., Greenville, S.C.
Asch, Abraham Bernard ('30; '45) (NMLABC), Ch. Engr., Equip. Div., Taller & Cooper, 75 Front St., Brooklyn 1; *for mail*, 98-11 Queens Blvd., Forest Hills, N.Y.
Aseltine, Arthur Warwick ('25; '35) (BLGC), Pres., Gen. Mgr., Sterling Appraisal Co. Ltd., 18 Toronto St., Toronto, Ont., Can.
Ashbaugh, Bernard David (J'40) (HYCBLN), Sales Engr., Hyd. Press Mfg. Co. (Mt. Gilead, Ohio), 201 N. Wells St., Chicago, Ill.
Ashby, Carroll Charles, Jr. (J'41) (AMY), Lt., Army Air Forces, 407th Serv. Gp., Pratt, Kan.
Ashby, Thomas F. (J'38) (ACS), Engr., Kan. Elec. Power Co., 101 W. 6th Ave., Emporia, Kan.
Ashby, William Bacon (J'40) (ZMLDCA), Am. Meter Co., Inc., 1513 Race St., Philadelphia, Pa.
Ashcroft, Alfred Griffin ('33) (TC), Dir. Research, Alexander Smith & Sons Carpet Co., Yonkers 1, N.Y.
Ashcroft, Herbert, Jr. (J'43) (CMJE), Engr., Charge of Prod., N.J. Optical Co., 56 S. 20th St.; *for mail*, 653 Stuyvesant Ave., Irvington 11, N.J.
Ashe, William Oliver ('42) (RBLNS), Sales Engr., Gen. Steel Castings Corp., Eddystone; *for mail*, 907 Burmont Rd., Drexel Hill, Pa.
Ashenden, Ernest Wickham ('21) (SKD), Ch. Engr., Wm. Bros. Boiler & Mfg. Co., 1057—10th Ave., S.E., Minneapolis 14, Minn.
Ashkinazy, Samuel Bernard ('34; '41) (JYLMA), Stands. Engr., Matls. & Process, Sperry Gyroscope Co., Inc., Great Neck, L.I.; *for mail*, 8210—19th Ave., Brooklyn 14, N.Y.
Ashleman, Russell H. (J'37) (ABJ), Engr., Boeing Aircraft Co.; *for mail*, 10623—20th Ave. S., Seattle, Wash.
Ashley, Edward E. ('10; '14; '16) (BEFKOS), Sr. Partner, Firm of Edw. E. Ashley, Cons. Engr., 10 E. 40th St., New York 16, N.Y.
Ashley, Henry Coomes ('34; '44) (CJKM), Metal. Engr., Chase Brass & Copper Co., Inc., 236 Grand St., Waterbury 91; *for mail*, 12 Hillcrest Ave., Watertown, Conn.
Ashley, Jesse E., Jr. (J'41) (LJSM), Design Engr., Vulcan Copper & Supply Co., 120 Sycamore St., Cincinnati 2, Ohio; *for mail*, 69 W. Southgate Ave., Ft. Thomas, Ky.

Ashmead, Albert Sidney (J'42) (MCNJBD), Trng. Program, Torrington Co., 59 Field St., Torrington; *for mail,* 3 Circle Dr., Bantam, Conn.

Ashmead, Richard Robert (J'44) (BJKANC), Sgt., Co. H, 4th Bn., Separation Center, Ft. George G. Meade, Md.; *for mail,* 1668 E. Cheltenham Ave., Philadelphia 24, Pa.

Ashmun, Louis Henry ('17; '26; '35) (OLCY), 1st Lt., Chem. Warfare Serv., Ret., U.S.A.; *for mail,* Box 927, Battle Creek, Mich.

Ashton, James W. (J'41), 10356 Commerce, Tujunga, Calif.

Ashton, Randolph ('35; '35) (ABH), Test & Research Engr., Exper. Towing Tank, Stevens Inst. of Tech., Hoboken, N.J.; *for mail,* 800 Crown St., Morrisville, Pa.

Ashton, Thomas P. (A'41) (MNC), Field Rep., State of Conn. Devel. Comm., State Office Bldg., Hartford; *for mail,* 290 Monroe Ave., Waterbury 72, Conn.

Ashwood, Loren Frisk (J'45) (DNCMKS), Mech. Engr., Ft. Wayne Corrugated Paper Co., E. Water St., Hartford City, Ind.

Ashworth, J. Vernon (J'44) (CM), Prod. Engr., Fed. Shipbldg. & Dry Dock Co., Lincoln Highway, Kearny; *for mail,* 32 Coeyman Ave., Nutley 10, N.J.

Askew, Miles A. ('21; '35) (RS), Lub. Engr., Texas Co., 135 E. 42nd St., New York 17, N.Y.; *for mail,* 13308—6th Ave., East Cleveland 12, Ohio.

Askren, John Arthur ('45) (CLOMD), Mech. & Struc. Engr., Engrg. Dept., A. E. Staley Mfg. Co., Decatur; *for mail,* 1310 W. Macon, Decatur 32, Ill.

Asmussen, Jes ('44) (CBJNSL), Ch. Engr., Menasha Div., Marathon Corp., Menasha; *for mail,* 1112 W. Prospect Ave., Appleton, Wis.

Asnin, James I. (J'43) (MCHZD), Ord. Engr., Bur. of Ord., Navy Dept., Washington, D.C.; *for mail,* 2109 N. Glebe Rd., Arlington, Va.

Aspinall, Arthur Rutherford (J'45), Mgr., Hulton Screw Bolt Works, F. Aspinall & Co., Over Hulton; *for mail,* "The Cottage," Bottom-O'th-Moor, Bolton, Lancs, England.

Assaykeen, Ivan V. ('28) (HSBJ), Mech. Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 3 Fremont Rd., Summit, N.J.

Ast, John Herman (J'45) (MCNJBH), Jr. Mech. Engr., Joshua Hendy Iron Works, Sunnyvale; *for mail,* 401 Webster St., Palo Alto, Calif.

Astley, Wayne Casselberry (J'44) (COS), Asst. Test Engr., Phila. Elec. Co., 10th & Chestnut St.; *for mail,* 812 N. 63rd St., Philadelphia, Pa.

Aston, George F. ('44) (EOS), Consultant Engr., 214 Columbia Bldg., Portland 5, Ore.

Atchison, William Rogers (J'46) (CJP), Sales Engr., Key Co.; *for mail,* 2665 Grand Ave., Granite City, Ill.

Atherholt, Gordon M. ('21; '25) (CDEFKL), Lawyer & Tech. Expert, Gulf Oil Corp.; *for mail,* 4645 Garfield St., N.W., Washington 7, D.C.

Atherton, Henry Burwell ('42) (SCJFBY), Asst. Ch. Opera. Engr., Kansas City Power & Light Co., 1330 Baltimore St., Kansas City 1; *for mail,* 714 E. 71st Terrace, Kansas City 5, Mo.

Atkins, David Fowler ('07) (KEFS), Cons. Engr., 144-47—37th Ave., Flushing, N.Y.

Atkins, Harold B. ('03) (C), Cons. Engr., 1 E. 42nd St., New York 17, N.Y.

Atkinson, David William (J'40) (ARSCY), Capt., 1205 N. Madison St., Peoria, Ill.

Atkinson, Embry Swearingen ('45) (MSACEF), Pres., E. S. Atkinson Co., 301 W. 12th St.; *for mail,* 509 Gordon Lane, Erie, Pa.

Atkinson, Francis Whitmer (J'40) (OCGSND), Draftsman & Bldg. Inspr. with Gordon Ferguson, Arch. & Engr., 111 S. Amherst Ave.; *for mail,* 502 S. Dartmouth Ave., Albuquerque, New Mex.

Atkinson, Herbert S. ('15) (D), Mgr., Clam Shell Bucket Dept., Hayward Co., 50 Church St., New York 7, N.Y.

Atkinson, John H. (J'40) (FKS), U.S.N., 210 S. Washington, Moorestown, N.J.

Atkinson, Kerr ('30) (SEFKHB), Project Engr., Jackson & Moreland, Engrs., 31 St. James Ave., Boston 16, Mass.

Atkinson, Rha Linwood ('28) (BCJMO), Head Mech. Engr., Am. Div., Indus. Serv., Ord. Dept., U.S.A., The Pentagon, Washington, D.C.; *for mail,* 10 W. Mason Ave., Alexandria, Va.

Atkinson, Vernon Lee (J'27) (HKC), Maint. Engr., Bayonne Refinery, Stand. Oil Co. of N.J., Bayonne; *for mail,* 32 Lafayette Ave., East Orange, N.J.

Atlas, Reynold A. (J'36) (SFKW), Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York; *for mail,* 104 Manor Rd., Staten Island 10, N.Y.

Attendu, A. C. ('44), Turcot Aircraft Plant, Canadian Car & Fdy. Co. Ltd., 621 W. Craig St., Montreal, Que., Can.

Atterbury, George R. (J'35) (AEHMR), Comdr., U.S.N.R., Ithan, Pa.

Attinello, John Salvatore (J'43), 363 Grant St., Phillipsburg, N.J.

Attschuler, William Howard ('44) (CJLMYD), Factory Mgr., Shure Bros., 225 W. Huron St., Chicago 10, Ill.

Attwood, John G. ('41) (BJTHZ), Test Engr., Union Spec. Mch. Co., 400 N. Franklin St., Chicago; *for mail,* 616 Harrison St., Oak Park, Ill.

Atwater, Harry Arthur ('14; '25) (FKS), Ch. Engr., Combustion Equip. Co., 1820 Cherry St., Kansas City 8; *for mail,* 641 W. 67th St., Kansas City 5, Mo.

Atwell, Charles Simpson ('39), Retired; 3349—6th St., Port Arthur, Tex.

Atwood, Hugh Moore (J'39) (CAHM), Engr., Quality Maint., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 607 Riverside Ave., Scotia 2, N.Y.

Aude, Theodor Rendtorff ('44) (CEH), Ch. Engr., Stanolind Pipe Line Co., Box 591, Tulsa 2, Okla.

Audet, Clement Richard (J'42) (ACMSOH), Ensign, U.S.N.R., U.S.S. *Deft,* AM-216, F.P.O., San Francisco, Calif.; *for mail,* 27 Harbor St., Salem, Mass.

Auer, Gustavus ('27; '35) (EFKL), Sanderson & Porter, 52 William St., New York, N.Y.

Aufrichtig, Jess Irving (J'42), Pvt., B Btry., 31st Bn., 8th Regt., Field Arty. Replacement Trng. Center, Ft. Sill, Okla.

Aug, William F. (J'34) (BAE), Asst. Dir. Research, Mack Mfg. Corp., 1120 S. 2nd St., Plainfield, N.J.

Aughenbaugh, Elmer Eugene ('36; '43) (CHM SRN), Asst. Supt., Charge of Prod., S. Morgan Smith Co., Lincoln & Hartley Sts.; *for mail,* 594 Linden Ave., York, Pa.

August, Irving Edward (J'37) (HCR), Supt., Vertical Turbine Pump Div., Worthington Pump & Mchy. Corp., Box 6864, Stockyard Sta., Denver 16, Colo.

Augustine, Alfred (J'36) (EFKZ), Combustion Engr., Loftus Engrg. Corp., 1620 Oliver Bldg., Pittsburgh 22; *for mail,* 406 S. Highland Ave., Pittsburgh 6, Pa.

Auld, Robert Archibald (J'43), Draftsman with George G. Sharp, Naval Arch., 30 Church St., New York, N.Y.; *for mail,* 67 Girard Ave., East Orange, N.J.

Ault, E. Stanley ('21; '28; '35) (BML), Prof. Mch. Design, Purdue Univ.; *for mail,* W. Stadium Rd., West Lafayette, Ind.

Aumenta, Joseph Edward (J'45) (CKSBE), Co. 1082, Naval Trng. Center, Great Lakes, Ill.

Aurien, Ray Gerard ('44) (CJR), Asst. Ch. Mech. Engr., Am. Steel Foundries, 410 N. Michigan Ave., Chicago 11, Ill.

Aust, Leo Gustave R. (J'42), Asst. Ch. Engr., Richard Dudgeon, Inc., 24-26 Columbia St.; *for mail,* 141 E. 26th St., New York, N.Y.

Austin, Calvin T. ('43), 2175 Mendocino Lane, Altadena, Calif.

Austin, Clem C. ('29) (CA), Gen. Mgr., Mancha Div., Goodman Mfg. Co., 4836 S. Halsted St., Chicago 9, Ill.

Austin, Elden (J'43), Lt., U.S.N.R., 2609 Wheeling St., El Paso, Tex.

Austin, Harold Raymond ('16) (C), Exec. V.P., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.

Austin, Henry Warner ('44) (SREAKO), Mar. Specifications Engr., U.S. Maritime Comm., Commerce Bldg., Washington 25; *for mail,* 909 Ridge Rd., S.E., Washington 19, D.C.

Austin, Hoyt (J'37) (BE), Petroleum Engr., Mene Grande Oil Co., Apartado 45, Barcelona, Venezuela, S.A.

Austin, Richard Smith ('16; '21; '31) (ACMS), Mgr., Mech. Div., Calco Chem. Div., Am. Cyanamid Co., Bound Brook; *for mail,* Beaver Brook Farm, Annandale, N.J.

Austin, Sidney Fairbanks (J'44) (ABNEKS), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 3760 Bosworth Rd., Cleveland 11, Ohio.

Austin, William Sumner ('02; '06) (ORKDM), Mgr., Firm of W. S. Austin, 802 Maryland Trust Bldg., Baltimore 2, Md.

Autenrieth, George C. ('08; '14) (EGM), Prof. Drafting, Supvr. Mech. Plant, College of City of N.Y., 139th St. & Convent Ave., New York, N.Y.

Auth, George Herbert (J'35) (BEKNOS), Asst. Prof. Mech. Engrg., Villanova College, Villanova, Pa.

Avedovech Myer ('44) (CDL), Gen. Mgr., Quaker Oats Co., 141 W. Jackson Blvd., Chicago 4, Ill.

Avent, John Sheppard (J'46) (JMY), Box 596, Broadway, N.C.

Averill, Earl A. ('40) (RMJY), 65 Church St., Adams, N.Y.

Avery, Clarence L. ('30; '35) (EHS), Engr., Woodward Governor Co.; *for mail,* 1735 Cumberland St., Rockford, Ill.

Avery, George Raymond ('26; '33; '35) (SFE DLM), Supt., Steam & Power Dept., So. Alkali Corp.; *for mail,* 922 Oak Park Ave., Corpus Christi, Tex.

Avery, Harold Terry ('14; '35), Ch. Engr., Potter & Dugan, Inc., 203 Herald Bldg.; *for mail,* 816 Euclid Ave., Syracuse, N.Y.

Avery, Harold Tolman ('35) (ANMCJB), Ch. Engr., Merchant Calculating Mch. Co., 1475 Powell St., Oakland 8, Calif.

Avery, Jasper William ('41; '45) (HKS), Mech. Engr., Havens & Emerson, 1140 Leader Bldg., Cleveland 14, Ohio.

Avery, John Richard (J'32) (RMJ), Mech. Engr., Indus. Test Lab., Bldg. 121, Naval Base Sta., Naval Shipyard; *for mail,* 247 W. Abbotsford Ave., Philadelphia 12, Pa.

Avery, True M. ('31) (WTYJNG), V.P., Union Bag & Paper Corp., Hudson Falls, N.Y.

Avery, William Murdoch ('44) (DCES), Field Engrg. Editor, Complete Serv. Publ. Co., 538 S. Clark St., Chicago 5, Ill.

Avey, Harry T. ('18) (ABM), Yaphank, L.I., N.Y.

Avigdor, Robert (J'43) (CJLM), Sr. Process Engr., RCA Victor Div., Radio Corp. of Am., 415 S. 5th St., Harrison, N.J.; *for mail,* 3951-7 Gouverneur Ave., New York 63, N.Y.

Avilez, Ignacio ('45), Cons. Engr., Compania de Construcciones, S.A.; *for mail,* Cuautla 158, Mexico, D.F., Mex.

Aviza, John Joseph (J'41), Pvt., 95th Air Base Sqd., Army Air Base, Ft. Dix, N.J.

Avner, Sidney Howard (J'44) (JKB), U.S.N., 161 Buffalo Ave., Brooklyn 13, N.Y.

Avnsoe, Thorkild ('13; '16; '22) (CDL), V.P., Lone Star Cement Corp., 342 Madison Ave., New York, N.Y.

Avram, Mois Herban ('37) (ACMO), Pres., Indus. Electronic Corp., 505 Court St., Brooklyn 32; *for mail,* 25—5th Ave., New York, N.Y.

Axelson, John Allan (J'43) (ABK), Mech. Engr., Natl. Advisory Comm. for Aero., Moffett Field; *for mail,* 551 Salvatierra St., Stanford University, Calif.

Axon, Albert E. ('28; '35), Bank of Australasia Chambers, Queen St., Brisbane, Queensland, Aust.

Axtell, Frank Foy ('45) (CPS), Asst. Power Engr., Neches Butane Products Co., Port Neches, Tex.

Ayars, William Stewart ('13), Retired; 269 Leonia Ave., Leonia, N.J.

Ayasun, Nurettin (J'44) (CMAZND), Erbaa, Turkey.

Aydelott, Max Merton (J'45) (BJKH), Seaman, 1st Class, U.S.N., Commissary Store, Great Lakes, Ill.

Ayer, Luther S. ('14), Mgr., Internatl. Motor Co., S. 2nd St., Plainfield, N.J.

Ayer, William Travis ('16) (BHM), Retired; 801 W. 32nd St., Wilmington 276, Del.

Ayers, Donald Jean ('43) (BJN), Ch. Engr., Quality Spring Products, Inc., Coldwater, Mich.

Ayers, Roscoe Grove ('30) (BCELNY), Gen. Sales Mgr., Bethlehem Supply Co., 805 E. Archer St., Tulsa 2; *for mail,* 1742 E. 30th St., Tulsa 5, Okla.

Ayres, James Marx (J'45) (KAYFMD), 11147 S. Berendo Ave., Los Angeles 44, Calif.

Ayres, Richard (J'43) (AJP), Mech. Engr., Magnolia Petroleum Co., Box 900, Dallas 1, Tex.

Ayres, Russell William ('37) (CKM), Ch. Engr., Coolerator Co., Box 399, Duluth 1, Minn.

Azpell, Edward William (J'43) (NEFS), Instr. Mech. Engrg., Purdue Univ., West Lafayette, Ind.

Azua, Ramon Virgilio, Jr. (J'41) (CHSTOE), Commercial Engr., Internatl. Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 2431 Jaffrey St., Schenectady 8, N.Y.

B

Baas, Philip B. R., Jr. (J'46), Rep., Bur. of Aero., Navy Dept., 1003-05 Customs Bldg., 2nd & Chestnut Sts.; *for mail,* 5969 Springfield Ave., Philadelphia, Pa.

Baase, Fred Carl (J'39) (HRS), 31 Avon St., Reading, Mass.

Babaycon, M. A. ('33), Ch. Engr., Asoka Mills, Ltd.; *for mail,* Asoka Banglow, Naroda Rd., Ahmadabad, Bombay, India.

Babb, Robert Masters (J'39) (DHLOSW), Engrg. Dept., Brunswick Pulp & Paper Co., Brunswick, Ga.

Babcock, C. L. ('41), Babcock Mchy. Co., Inc., 1475 Broadway, New York, N.Y.

Babcock, Hugh Howard ('45), Asst. Supt., Natl. Carbon Co., Charlotte, N.C.

Babcock, Lawrence R. ('30; '36) (EFS), Mgr., Field Engrg. Div., Petroleum Heat & Paper Co., Box 1457, Stamford; *for mail,* Box 22, Noroton Heights, Conn.

Babcock, Stanley Earle (J'43) (MCD), R.D. 1, Binghamton, N.Y.

Babcock, William W. ('38) (S), Supt. Power, Cent. Ill. Light Co., 316 S. Jefferson St., Peoria 2; *for mail,* 320 Margaret St., Peoria 5, Ill.

Babeor, Joseph A. (J'36) (ACM), Mgr., Rice's Bakery, 709—12th St., S.E., Washington, D.C.

Babikian, Hrant Melkon ('38; '45) (SHBOLT), Cons. Engr., 38 Blvd. Saad Zaghloul, Alexandria, Egypt.

Babin, Alexander (J'41) (MCB), 619 W. 140th St., New York 31, N.Y.

Babington, Neil G. (J'45), Lt. Comdr., U.S.N.R.; *for mail,* 129 W. Adeline St., Buffalo 11, N.Y.

Bach, George William ('18; '19; '34), Pres., Gen. Mgr., Am. Sterilizer Co., 1230 Plum St., Erie, Pa.

Bacha, Charles Paul (J'35) (BAGJC), *Student Award,* '35; Project Engr., Tech. Writer, Hyatt Bearings Div., Gen. Motors Corp., 4th St., Harrison; *for mail,* 91 Amboy Ave., Metuchen, N.J.

Bache, William Hugh (J'45) (SKB), Mech. Engr., Crawford Sprinkler Supply Co., Box 28, 374-78 Plant St., East Point, Ga.

Bacheller, Allan Walker (J'43) (BCDJMN), Engr., Olson Mfg. Co., Box 1487; *for mail,* 917 E. Jefferson, Boise, Idaho.

Bachman, Benjamin Byron ('14; '18) (ABEH JK), V.P., Autocar Co., Ardmore, Pa.

Bachman, Charles Amandus (J'43) (MCD), Technician, 5th Grade, Corps of Engrs., U.S.A. (assigned to Carbide & Carbon Chemicals Corp.); *for mail,* Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.

Bachman, Gerald G. (J'39) (CKS), Neb. Power Co., 4th & Jones St., Omaha, Neb.

Bachman, Joseph L. (J'35) (ABMN), Project Engr., Naval Air Matls. Center, Naval Base Sta., Philadelphia; *for mail,* 3327 Walnut St., Philadelphia 4, Pa.
Bachman, Walter Crawford (J'40) (SBCA), Mar. Engr., Gibbs & Cox, Inc., 21 West St., New York 6, N.Y.; *for mail,* Brantwood, Summit, N.J.
Bachman, William A. ('22) (ACDJR), Supt. Fdys., N.Y. Cent. R.R. Co.; *for mail,* 217 N. 2nd, Elkhart, Ind.
Bachus, Benson Floyd (J'42) (MOLKJC), Apt. 21, Exposition Hotel, Aurora, Ill.
Backity, Stephen M. (J'37) (CDJMN), Exec. V.P., Treas., Am. Prod. & Grinding Corp., 213 Washington Sq., Syracuse 8, N.Y.
Backus, Richard Allison ('21) (JBW), Ch. Struc. Engr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York 17, N.Y.
Bacon, David Leonard (J'17) (CMO), V.P., Greist Mfg. Co., 446 Blake St., New Haven 15, Conn.
Bacon, George Wood ('98; '99; F'41), Chmn. of Bd., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Bacon, Howard E. ('16; '26), Spec. Rep., Stoker Dept., Westinghouse Elec. Corp., 30th & Walnut St., Philadelphia; *for mail,* 913—10th Ave., Prospect Park, Pa.
Bacon, John L. ('99; '09) (CJM), 2833 Nutmeg St., San Diego, Calif.
Bacon, Lewis Howard, III (J'44), Lt. (j.g.), U.S.N.R., 1478 Main St., South Weymouth, Mass.
Bacon, Malcolm ('33) (EOACWY), Appraisal Engr., Assessor of Los Angeles County, Hall of Justice, Los Angeles 12; *for mail,* 5316 Lockhaven Ave., Los Angeles 41, Calif.
Bacon, Rinaldo A. (J'39) (BJM), Lt., U.S.N.R.; Mar. Engrg. Dept., U.S. Naval Acad., Annapolis, Md.
Bacon, Robert Hamilton ('23; '35) (CGNELO), Owner, R. H. Bacon & Co., 221 N. La Salle St., Chicago 1, Ill.
Badeau, Carroll A. ('44) (JMNC), Ch. Engr., Thomas & Betts Co., Inc., 36 Butler St., Elizabeth 1, N.J.
Badenhausen, John Phillips ('22) (FS), Mgr., Firm of John Phillips Badenhausen, 1801 Packard Bldg., Philadelphia 2, Pa.
Baecher, Bernard Joseph (J'35) (BZE), Matl. Engr., Matl. Lab., N.Y. Naval Shipyard, Brooklyn; *for mail,* 425—101st St., Brooklyn 9, N.Y.
Baechlin, Alfred C., Jr. ('43) (CO), Engr., West. Elec. Co., Inc., 717 Ave. A, Bayonne; *for mail,* 26 Whitney Rd., Short Hills, N.J.
Baeckler, Walter ('45) (BCDHKL), Works Engr., Natl. Carbon Co., Inc., Buffalo Ave.; *for mail,* 764 Van Rensselaer, Niagara Falls, N.Y.
Baender, Frederick Gottlieb ('27) (BEFO), Prof. Mech. Engrg., Univ. of Va., Charlottesville, Va.
Baer, Carl Adams ('43) (YCMJT), Admin. Asst., Rubber Reserve Co.; *for mail,* 1717 G St., N.W., Washington, D.C.
Baer, Roy ('23; '35), Harris Hotel, Rogers, Ark.
Baer, Simon ('45) (ACFMOR), Cons. Engr., 76-36—113th St., Forest Hills, N.Y.
Baeyertz, Francis Phelps ('44) (SEF), Supvr. of Indus. & Mar. Sales Trng., Gen. Petroleum Corp., 108 W. 2nd, Los Angeles 12; *for mail,* 101 E. Hermosa Dr., San Gabriel, Calif.
Bafaro, Michael E. (J'45) (NBKJYS), Jr. Design Engr., Gen. Elec. X-Ray Corp., 2012 W. Jackson Blvd., Chicago 12; *for mail,* 2806 S. 50th Ave., Cicero 50, Ill.
Bagby, Frederick L., Jr. (J'42) (AMCJ), Prod. Design Engr., Airplane Div., Curtiss-Wright Corp., 4300 E. 5th Ave., Columbus 16; *for mail,* Apt. A, 774 Chesterfield Rd., Columbus 9, Ohio.
Baggs, Arthur Eugene, Jr. (J'39) (BNMJY), Design & Devel. Engr., Polaroid Corp., 109 Brookline Ave., Boston; *for mail,* 8 Redstone Lane, Marblehead, Mass.
Bagley, Glen D. ('18; '35), Research Engr., Union Carbide & Carbon Research Labs., Royal Ave., Niagara Falls, N. Y.
Bahr, John L. ('46), Mech. Engr., Natl. Vulcanized Fibre Co.; *for mail,* Marshall Heights, Yorklyn, Del.
Baier, Edwin W. (J'41), Lt. (j.g.), U.S.N.R.; 54 Welton St., New Brunswick, N.J.
Bailey, Albert ('28) (MDC), Mech. Engr., United Am. Bosch Corp., Springfield; *for mail,* 859 Chestnut St., Springfield 7, Mass.
Bailey, Alex Davison ('10; '16; F'36) (SFC), *Manager,* '32-35, *Vice-President,* '35-37, *President,* '45; V.P., Commonwealth Edison Co., 72 W. Adams St., Chicago 90, Ill.
Bailey, Bruce L. ('34; '41), 573—82nd St., Niagara Falls, N.Y.
Bailey, Charles Alonzo ('36: '42) (CDJZ), Spec. Engr. (Oper. Methods), Carnegie-Ill. Steel Corp., 1 Broadway; *for mail,* 628 Vermont St., Gary, Ind.
Bailey, Charles Henry (J'44) (EHK), Jr. Engr., Shell Devel. Co., Emeryville; *for mail,* 26 Avon Rd., Berkeley 8, Calif.
Bailey, Chas. J. ('21; '23; '35), Sales Engr., Bridgeport Brass Co.; *for mail,* 55 Laurel Ave., Bridgeport, Conn.
Bailey, Daniel Robert (J'43) (SEKC), Supvr., Ship Repair, East Coast Shipyards, Inc., E. 21st St., Bayonne; *for mail,* 450 Highland Ave., Wood-Ridge, N.J.
Bailey, E. G. ('03; '12; F'43) (FSRZ), *A.S.M.E. Medallist,* '42; V.P., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Bailey, Miss Ethel H. ('26; '35) (CGM), Assoc. Dir., Tech. Publications Div., Buck Ptg. Co., 145 Ipswich St., Boston, Mass.
Bailey, Eugene Cary (J'39) (SFK), Capt., Ord. Dept., U.S.A., Office of Chief of Ord., Guardian Bldg., Detroit 32; *for mail,* 11508 Balfour Ave., Detroit 24, Mich.
Bailey, Eugene G. ('27) (CTW), 250 Bronxville Rd., Bronxville, N.Y.
Bailey, F. E. ('31; '35) (FLS), Plant Engr., Crocker Burbank & Co. Assn.; *for mail,* 40 Haskell St., Fitchburg, Mass.
Bailey, George Bowen ('44) (HFZBC), Chem. Engr., Hays Corp.; *for mail,* Mayfield Way, Long Beach, Michigan City, Ind.
Bailey, James ('19; '33) (BHJKY), V.P., Dir. Research, Plax Corp., 133 Walnut St., Hartford 5; *for mail,* 95 Steel Rd., West Hartford 7, Conn.
Bailey, Joel F. (J'39) (KSB), Asst. Prof. Mech. Engrg., Northwestern Tech. Inst., Evanston, Ill.
Bailey, John A. (J'39) (BCJ), Capt., Army Serv. Forces, Hdq., Intermediate Sec., Distribution Div., Ord. Branch, A.P.O. 503, c/o Postmaster, San Francisco, Calif.
Bailey, Joseph Tindell (J'43) (HSW), Ensign, U.S.N.R., U.S.S. *Reform,* AM-286, F.P.O., San Francisco, Calif.
Bailey, Neil Phillips ('24; '30; '35) (KABEN), Head, Mech. Engrg. Dept., Rensselaer Poly. Inst., Troy, N.Y.
Bailey, Norborne Foster ('45) (BEJKSY), Ch. Inspr., Hartford Steam Boiler Inspc. & Ins. Co., 1325 C. & S. Natl. Bank Bldg.; *for mail,* 1791 Beecher St., S.W., Atlanta, Ga.
Bailey, Ralph Reginald (J'45) (SCEFKO), Asst. Mgr., Turbine Engrg. Corp., 201 Harrison St., Hoboken; *for mail,* 152 Sunset Lane, Tenafly, N.J.
Bailey, Robert Dedan (J'41) (JEH), Maj., 322nd Engr. Bn., A.P.O. 445, Ft. Leonard Wood, Mo; *for mail,* 1820 S. 50th St., Lincoln 6, Neb.
Bailey, Robert Leroy (J'45) (KWNBHY), Graduate Engr. in Trng., Universal Cooler Co. of Can., Ltd.; *for mail,* 52 Murray St., Brantford, Ont., Can.
Bailey, Robert Oliver ('39) (FS), Sales Engr., Am. Blower Corp., 50 W. 40th St., New York 18, N.Y.
Bailey, Roland J. (J'43) (SKL), Engrg. Draftsman, Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 399 Malborough St., Boston 15, Mass.
Bailey, Wayland S. ('30; '30; '35) (BKS), Asst. Prof. Mech. Engrg., Northeastern Univ., 360 Huntington Ave., Boston; *for mail,* Box 37, Norwell, Mass.
Bailey, William H. (J'41), Lt. (j.g.), U.S.N.R., Assembly & Repair Office, Naval Air Sta., Lakehurst, N.J.
Bailey, William J. ('10; '17) (FSEKHJ), Owner, William J. Bailey Coal & Coke, 1617 Pennsylvania Blvd., Philadelphia 3; *for mail,* 228 Essex Ave., Narberth, Pa.
Bailie, Bertram Lawrence ('39) (CLKM), Plant Engr., Hiram Walker & Sons, Inc., Edmund St.; *for mail,* 1518 Starr Terrace, Peoria, Ill.
Baillie, Robert Randall ('19; '35) (SJB), Leader Inspr. of Engrg. Matls., U.S.N., 1600 Arch St., Philadelphia 3; *for mail,* 4001 Walnut St., Philadelphia 4, Pa.
Bainbridge, Harold (J'46) (BCD), Sales Engr., Mech. Handling Systems, Inc., 1065 Commercial Trust, Philadelphia, Pa.; *for mail,* 5 Pingree Court, Trenton 8, N.J.
Bainbridge, Thomas Warren (J'44), 208 Yeakel Ave., Philadelphia 18, Pa.
Bainton, Arthur H. ('25), Works Mgr., Brown & Sharpe Mfg. Co., Providence 1, R.I.
Bainton, Wallace Boyd ('43) (CJM), Asst. Works Mgr., Brown & Sharpe Mfg. Co., Providence 1, R.I.
Baird, Charles Augustus (J'43) (KS), Test Engr., C. H. Wheeler Mfg. Co., Lehigh & Sedgley Aves., Box 6910, North Philadelphia 32, Pa.
Baird, Eugene Frederick (J'45) (ABO), Struc. Research Engr., Plant 5, Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* 318 W. 108th St., New York 25, N.Y.
Baird, Hobart Brown ('21) (EF), V.P., East. Gas & Fuel Associates, 250 Stuart St., Boston, Mass.
Baird, Walter G. ('44) (BCMNJZ), Research Engr., Stow Mfg. Co., 445 State St.; *for mail,* 5 Hotchkiss Ave., Hillcrest R. D. 3, Binghamton, N.Y.
Baites, James Robert (J'40) (BDEHS), 911 Pratt Ave., Huntsville, Ala.
Baits, Stuart G. ('18; '30) (CEBFJ), 1st V.P., Asst. Gen. Mgr., Hudson Motor Car Co., Jefferson Ave. E., Detroit 14, Mich.
Baity, Geo. W. ('39) (JLT), Gen. Engr., Proximity Mfg. Co., 4th & Maple Sts.; *for mail,* Box 17, Denim Sta., Greensboro, N.C.
Bak, Anders Kristian ('20; '25) (EFS), Vognmagerade 8, Copenhagen K, Denmark.
Baker, Albert Lundy ('36) (EFL), Gen. Mgr., Kellex Corp., 233 Broadway, New York 8, N.Y.
Baker, Aquila Woodfin, Jr. (J'45) (AEFKRS), Graduate Student, Ga. Sch. of Tech., Box 209, Atlanta, Ga.
Baker, Arthur Lee (J'39) (ONKBW), Col., Corps of Engrs., U.S.A., Engr. Sec., Hdq., Army Ground Forces, Washington 25, D.C.
Baker, Bernard L. (J'40) (ABS), 32 Whitin Ave., Point of Pines, Mass.
Baker, Clarence LeRoy (J'39) (FSYM), Power Engr., B. F. Goodrich Co., Box 1858, Louisville 1, Ky.
Baker, Dickerson G. ('05), Ch. Engr., Mchy. Devel. Dept., Am. Thread Co.; *for mail,* 1 Yale St., Holyoke, Mass.
Baker, Douglas Brooks ('28) (CDMLAW), V.P., Internatl. Telecommunication Labs., 67 Broad St., New York 4, N.Y.
Baker, Edward George ('45) (BNAJES), Mem. Tech. Staff, Am. Bur. of Shipping, 47 Beaver St., New York 4, N.Y.
Baker, Ellis Crain ('29) (AMNS), Head, Mech. Engr. Dept., Okla. Agric. & Mech. College, Stillwater, Okla.
Baker, Frank (J'39) (AJM), Design Draftsman, Diesel Devel. Div., Am. Loco. Co.; *for mail,* 1152 Baker Ave., Schenectady, N.Y.
Baker, H. Raymond, Jr. (J'41) (DKS), Mech. Engrg. Div., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5; *for mail,* 315 Chandler St., Philadelphia 11, Pa.
Baker, Harry Damon ('43) (CJMN), Asst. Plant Engr., Dennison Mfg. Co., Howard St., Framingham; *for mail,* 207 Bacon St., Natick, Mass.
Baker, Henry Ashley ('44), V.P., W. B. Connor Engrg. Corp., 114 E. 32nd St., New York; *for mail,* 145-53—17th Rd., Whitestone, N.Y.
Baker, Henry Dean ('45), Instr., Columbia Univ., Pupin Bldg., New York 27, N.Y.; *for mail,* 28 Delaware Ave., Dumont, N.J.
Baker, Hiram M. ('29) (CES), Ch. Engr., Compressor Dept., United Fuel Gas Co., Box 1256, Charleston, W. Va.
Baker, James Oliver (J'45) (BES), Capt., Trans. Corps, Co. F, War Dept., Personnel Center, Ft. Lewis, Wash.
Baker, John B. ('27; '32; '35) (BHK), Asst. Prof. Mech. Engrg., Drexel Inst. of Tech., 32nd & Chestnut Sts., Philadelphia, Pa.
Baker, John R. (J'38), 540 N. Los Robles Ave., Pasadena 4, Calif.
Baker, John Rea ('23; '35) (SHCF), Supt. Power, Pa. Water & Power Co., 1611 Lexington Bldg., Baltimore 3, Md.
Baker, Linnaeus E. ('20), Mech. Engr., Shell Oil Co., Inc.; *for mail,* 803 Cedar Ave., Long Beach, Calif.
Baker, Louis Ralph ('40) (RJFCSM), Ch. Engr., Charge Design, Pilliod Co., 302 Church St., Swanton, Ohio.
Baker, Neal Allen (J'43) (ANBEKO), Mar. Engr., Supvr. of Shipbldg., U.S.N., Bethlehem Steel Co., Quincy 69; *for mail,* 14 Prospect Hill Dr., North Weymouth 91, Mass.
Baker, Paul H. (J'43) (BFS), Power Supvr., Ecusta Paper Corp., Pisgah Forest; *for mail,* 325 Probart St., Brevard, N.C.
Baker, Ralph D. (J'38) (ABKNHY), Prof. Mech. Engrg., Univ. of Utah, Salt Lake City, Utah.
Baker, Robert Emmett ('15) (CJEOLD), Secy., Treas., Dir., Arthur G. McKee & Co., 2300 Chester Ave., Cleveland 1, Ohio.
Baker, Robert L. (J'41) (NYABMC), Project Engr., Army Air Forces, Air Tech. Serv. Comd., Wright Field, Dayton, Ohio.
Baker, Robert Macy (J'34) (OM), Cpl., Hdq., 5th Army, A.P.O. 464, c/o Postmaster, New York, N.Y.
Baker, Roland Henry ('18; '26; '32; '43) (CO MSA), Pres., R. H. Baker Co., Elkins, N.H.
Baker, Stanley Veach ('43) (SFK), Ch. Engr., Gould St. Sta., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington & Liberty Sts., Baltimore 1; *for mail,* 823 Kingston Rd., Baltimore 12, Md.
Baker, Thomas A. ('42) (ABFHK), V.P., Baker Specialty & Supply Co., Inc.; *for mail,* 2205 Broadway, Logansport, Ind.
Baker, Walter Charles ('19) (AE), 1822 N.B.C. Bldg., Cleveland 14, Ohio.
Bakerjian, Benjamin Harry (J'46), Ensign, U.S.N.R.; 90 Boylston St., Watertown, Mass.
Bakesef, Samuel ('23) (BHO), Cons. Engr., 4461 Crocker St., Los Angeles 11, Calif.
Bakhmeteff, Boris Alexander ('31) (HBA), Chmn. of Bd., Lion Match Co., 250 W. 57th St., New York 19, N.Y.
Bakule, Carl V. ('39) (MCNEA), Wyoming, Minn.
Balavitch, John Michael (J'40) (EASKNH), Weight Estimator, Engrg. Dept., Glenn L. Martin Co., Baltimore 3, Md.; *for mail,* 9 Hamilton St., Lawrence, Mass.
Balch, William (J'28) (AZBE), Exper. Test Engr., Pratt & Whitney Aircraft, Main St., East Hartford 7; *for mail,* 127 Main St., Manchester, Conn.
Balcom, John A. ('24; '35) (BJKSYZ), 173 Broad St., Eatontown, N.J.
Baldo, Louie (J'39) (CAMDOB), 723 W. Goshen Ave., Visalia, Calif.
Baldwin, Ben A. (J'39) (EHJ), Converted Rice Co., 5612 Clinton Dr., Houston, Tex.
Baldwin, Edward Post (J'40) (ABS), Design Engr., Lockheed Aircraft Corp., 1705 Victory Pl., Burbank; *for mail,* 10908 Riverside Dr., North Hollywood, Calif.
Baldwin, Harrison P. (J'36) (WLKDOH), Engr., Hammermill Paper Co.; *for mail,* 832 W. 5th St., Erie, Pa.
Baldwin, Neil Allen (J'43) (S), Serv. Engr., Westinghouse Elec. Corp., 20 N. Wacker Dr., Chicago 6; *for mail,* 1400 E. 53rd St., Chicago 15, Ill.

Balint, Albert Bernard (J'41) (SMNCBZ), Sr. Serv. Engr., Arma Corp., 254—36th St., Brooklyn; *for mail,* 108 Hawthorne Ave., Yonkers 2, N.Y.
Balka, William H. ('40) (CELK), Ch. Engr., J. A. Jones Constr. Co., Inc., 506 Scanlan Bldg., Houston 2; *for mail,* 1824 Portsmouth, Houston 6, Tex.
Ball, C. Winthrop ('19; '25) (EN), Designing Engr., Steel Products Engrg. Co., 1205 W. Columbia St.; *for mail,* 2150 Broadway, Springfield, Ohio.
Ball, Eustace T. ('41) (LNWY), Mem. Tech. Staff, Bell Tel. Labs., 463 West St., New York, N.Y.
Ball, Frederick Stephen ('46), Ry. Engr., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia; *for mail,* 815 Glendalough Rd., Chestnut Hill, Pa.
Ball, Herbert James ('27) (TBC), Prof. Textile Engrg., Lowell Textile Inst., Lowell, Mass.
Ball, Herman F. ('13) (CRS), Pres., Franklin Ry. Supply Co., 60 E. 42nd St., New York, N.Y.
Ball, Lawrence Rutherford ('28; '35) (SFK), Ch. Power Engr., Whitin Mch. Works; *for mail,* 41 Prospect St., Whitinsville, Mass.
Ball, Robert Victor (J'39) (LKBNS), Jr. Engr., Calif. & Hawaiian Sugar Refining Corp., Ltd.; *for mail,* 45 Baldwin Ave., Crockett, Calif.
Ball, William, Jr. (J'44), Devel. Engr., Walter Kidde & Co., 675 Main St., Belleville; *for mail,* 112 N. 6th St., Newark 7, N.J.
Ball, William Spencer ('32) (BCDFKM), Maint. Engr., Canadian Arsenals, Ltd., Box 6112, Montreal, Que.; *for mail,* 14 Munro Park Ave., Toronto 8, Ont., Can.
Ballantine, John H. ('17; '27; '35), V.P., Charge Sales, Neptune Meter Co., 50 W. 50th St., New York, N.Y.
Ballard, Levi ('24; '31; '35) (EM), Tide Water Associated Oil Co., 17 Battery Pl., New York, N.Y.
Ballauer, Alb C. (J'39) (ACBK), Research Engr., McDonnell Aircraft Corp., Lambert Field, St. Louis 21; *for mail,* 213 Moundale, Ferguson, Mo.
Balleisen, Charles Edward (J'37) (BNJK), Maj., Ord. Dept., U.S.A., Research & Devel. Serv., 4D 432, The Pentagon, Washington 25, D.C.; *for mail,* 615 S. Irving St., Arlington, Va.
Ballenger, James Morley (J'38) (CMDLJ), Lt. Comdr., U.S.N.R., Officer in Charge, Internal Combustion Eng. Salvage Center, 4701 Grand Ave., Maspeth; *for mail,* 109-20—71st Rd., Forest Hills, L.I., N.Y.
Ballenger, Robert Daniel (J'44) (ONMDCB), Struc. Engr., U.S. Engrs., Tivoli; *for mail,* Box 938, Curundu, C.Z.
Baller, Philip W. G. ('23; '25; '35) (FS), Turbine Room Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York; *for mail,* 548 Bement Hill, Staten Island 10, N.Y.
Ballester, Rodolfo Emiliano ('44) (HBS), *Calvin W. Rice Lecturer;* Prof. Hyds., Univ. of Buenos Aires, Peru 222; *for mail,* 3 de Febrero 2318, Buenos Aires, Argentina, S.A.
Ballin, Alfred Edward ('06; '38) (CEMO), Owner Balco Engrg. Co., 508 Petroleum Bldg., Tulsa 3, Okla.
Ballinger, Philip Fazio ('45), Ch. Civilian Engr., Office of Post Engr., War Dept., Camp Stewart; *for mail,* 126 E. Taylor St., Savannah, Ga.
Ballman, Harry Charles ('35; '45) (FSKDE), Mgr. Personnel Trng., U.S. Mch. Corp., Lebanon; *for mail,* 1941 Medford Ave., Indianapolis 8, Ind.
Ballou, Frederick Hubbard ('15; '19) (LDKSFJ), Ch. Engr., B. C. Sugar Refining Co. Ltd., Foot of Rogers St., Vancouver, B.C., Can.
Ballou, Frederick Hubbard, Jr. (J'36) (LKS), Engr., Spreckels Sugar Co., 2 Pine St., San Francisco 11; *for mail,* Apt. 11, 1435 Washington St., San Francisco 9, Calif.
Ballou, John McK. ('16; '21; '25) (DJO), Cons. Engr., 219 W. 7th St., Los Angeles 14; *for mail,* 3475 San Pasqual St., Pasadena 8, Calif.
Ballway, Robert Carl ('44) (CEKMNY), Asst. to Ch. Exper. & Design Engr., Vacuum Gas Div., Uninfork & Hoe Co.; *for mail,* 603 Riverside Dr., Rome, N.Y.
Balmanno, William Crosby (J'39) (BENRZS), Design & Test Engr., Joshua Hendy Iron Works; *for mail,* 432 S. Mathilda Ave., Sunnyvale, Calif.
Balogh, Stephen I. ('27) (SEFBKJ), Elec. Engr., Dept. Head, Solar Mfg. Corp., 23rd St. & Ave. A, Bayonne, N.J.; *for mail,* 456 Riverside Dr., New York 27, N.Y.
Balough, Chas. ('15), V.P., Gen. Mgr., Hercules Motors Corp., Halliwell Pl., S.E., Canton, Ohio.
Balough, Harold Oliver (J'44) (BKS), Ensign, U.S.N.R.; Bellevue Hotel, Miami, Fla.
Balter, Harvey Maxwell (J'43) (BCMN), 54 Henry St., Passaic, N.J.
Balthasar, Frank L. ('25; '27; '35) (SBLYJC), Dist. Mgr., Foster Wheeler Corp., 1239 Leader Bldg., Cleveland, Ohio.
Balthrop, William Plummer (J'44) (ACS), Gallatin Rd., Goodlettsville, Tenn.
Balton, Herman (J'44) (ABFMND), Prod. Mgr., Gen. Oil Heating Corp., 324—55th St., West New York, N.J.
Baltzly, Clifford C. ('21; '30) (CF), Gen. Supt., Sta. Oper. Dept., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Balun, John J. (J'41) (BES), *Charles T. Main Award,* '41; 7032 Arcola St., Detroit 5, Mich.
Bame, Morris (J'45) (LCZ), Supt., Plant Engr., Toledo Soybean Products Co., 215 Pontiac St., Toledo 11; *for mail,* 4145 Kingsbury, Toledo 12, Ohio.
Banach, Sidney David (J'42) (AHL), Pvt., QM. Corps, QM. Bd., Camp Lee, Va.; *for mail,* 3965 Sedgwick Ave., New York 63, N.Y.
Banash, James Ira ('17) (AEJLOY), Cons. Engr., 230 N. Michigan Ave., Chicago 1, Ill.
Bance, Edlow Shipman ('34; '45) (FLKZDM), Lt., U.S.N.R., Ship Supt., Naval Shipyard, Boston, Mass.; *for mail,* Roseville Rd., Lake Lackawanna, Stanhope, N.J.
Banchero, Julius Thomas (J'45) (KLZ), Instr., Dept. of Chem. & Metal. Engrg., Univ. of Mich., Ann Arbor, Mich.
Banck, Hans J. E. ('21) (FKS), V.P., Ch. Engr., Springfield Boiler Co., 1901 E. Capitol Ave., Springfield, Ill.
Bancroft, Chas. F. ('11), Retired; Ferris Hill Rd., New Canaan, Conn.
Bancroft, Edward Kenneth (J'37) (SEF), Engr., Babcock & Wilcox Co., 85 Liberty St., New York, N.Y.; *for mail,* 45 Warren St., Brookline 46, Mass.
Bancroft, Gilbert Howard ('45) (DHMNOW), Ch. Engr., Secy., Treas., Engrg. & Mchy. Ltd., 786 Granville St., Vancouver, B.C., Can.
Bancroft, John ('37) (LMS), Asst. Ch. Engr., Cia. Azucarera Punta Alegre, S.A., Punta San Juan, Camagüey, Cuba.
Bancroft, Wilfred ('12) (CGT), Controller (semi-retired), Lanston Monotype Mch. Co., 24th & Locust Sts., Philadelphia 3; *for mail,* 761 Millbrook Lane, Haverford, Pa.
Bandelier, George Emil ('29; '35) (SEC), Gen. Mgr., Water & Light Dept., City of Stillwater; *for mail,* 105 S. Duck St., Stillwater, Okla.
Banerian, Gordon (J'44) (ABEHK), Analytical Engr., Continental Aviation & Engrg., 1280 Jefferson, Detroit 14, Mich.
Banfield, Frederic Eugene, Jr. ('09; '18) (CDJMNT), V.P., Works Mgr., Whitin Mch. Works, Whitinsville, Mass.
Bange, George Edmond ('45) (AHY), Mech. Engr., Mech. Design Gp., Remington Arms Div., E. I. du Pont de Nemours & Co., Barnum Ave., Bridgeport, Conn.
Banghart, Lee Edward ('21; '25; '35) (HNBJEL), Ch. Engr., Union Steam Pump Co., Capital Ave., S.W.; *for mail,* 76 Clinton Dr., Battle Creek, Mich.
Bangs, John R., Jr. ('38) (CM), Gen. Mgr., Personnel & Indus. Relations, Edw. G. Budd Mfg. Co., Philadelphia, Pa.
Bangser, Wm. ('31) (JMN), Secy., H. Maimin Co., 571—8th Ave., New York 18, N.Y.
Banister, Guy ('45) (DMNOKH), Ch. Engr., Barber Greene Co., Aurora, Ill.
Banks, Joseph Turney ('32; '35) (OS), Dist. Mgr., Gullett Gin Co., 1199 Spring St., N.W., Atlanta, Ga.
Banks, S. J. Egerton ('45), Regional Dir., Ministry of Works & Consultant, Vigilia, St. Thomas' Hill, Canterbury, Kent, England.
Banks, T. K. ('18; '35) (EMS), 4300 Jefferson St., Hyattsville, Md.
Bankson, E. Edwin (J'42), Ensign, U.S.N.R., 6562 Bartlett St., Pittsburgh, Pa.
Banner, Wilber Keith (J'44) (SEKFAB), Lt. (j.g.), U.S.N., Engrg. & Gunnery Officer, U.S.S. *Bronstein,* DE-189, c/o Postmaster, Green Cove Springs, Fla.
Banner, William (J'42) (MBHRA), Lt. (j.g.), 2nd Asst. Engr., U.S. Maritime Serv., United Fruit Co., Pier 9, New York; *for mail,* 87-72—97th St., Woodhaven 21, N.Y.
Bannerman, Charles Ranald, Jr. (J'35) (AHC), Test & Serv. Engr., Ford Instrument Co., Inc., Long Island City; *for mail,* 39-08—210th St., Bayside, N.Y.
Bannerman, Donald K. (J'43) (BA), Lt. (E), Royal Canadian Naval Volunteer Reserve, R.C.N. Barracks, c/o F.M.O., St. John's, Newfoundland; *for mail,* 4633 W. 3rd Ave., Vancouver, B.C., Can.
Bansbach, Herbert Leo (J'44), 104 Copley Rd., Upper Darby, Pa.
Banthin, John Frederick ('44) (CDGJMN), Sr. Partner, Mgr., Banthin Engrg. Co., 1849-57 Main St., Bridgeport 4, Conn.
Banton, Madison Whitten ('39) (CLMDS), Dir. Engrg. & Research, Congoleum-Nairn, Inc., 195 Belgrove Dr., Kearny; *for mail,* 79 Blackburn Rd., Summit, N.J.
Baratta, Henry E. ('45; '46) (SCOAWY), Plant Constr. Engr., Bell Tel. Labs., Murray Hill, N.J.
Barba, Charles Elmer ('44) (JLMNR), Designer, Franklin Inst., Philadelphia; *for mail,* 307 Chester Pike, Norwood, Pa.
Barbalich, Robert Peter (J'41) (AJM), Maj., Army Air Forces, Hdq., 42nd Wing, A.P.O. 650, c/o Postmaster, New York, N.Y.; *for mail,* 8123 Oleander St., New Orleans 18, La.
Barber, Dan Henry (J'43) (FMACKZ), Mech. Engr., Stand. Oil Co. of So. Calif., El Segundo Ave.; *for mail,* 940 Eucalyptus Dr., El Segundo, Calif.
Barber, Edmund Amaral, Jr. (J'38) (BNMJYZ), Lab. Technician, Internatl. Business Mchs. Corp., 1807 North St., Endicott; *for mail,* 24 Brewster St., Johnson City, N.Y.
Barber, K. B. (J'38) (AFS), Technician, 3rd Grade, Field Serv. Maint. Dept., Frankford Arsenal, Philadelphia 27, Pa.
Barber, Norman H. (J'45) (BCJ), Engr., Mead Co., 208 Fremont, San Francisco 5; *for mail,* Box 365, Route 1, Walnut Creek, Calif.
Barber, William Jackson ('25; '32; '35) (NBGS), 152 Milford St., East Lansing, Mich.
Barbieri, Cesare ('08), Cons. Engr., 340 Park Ave., New York, N.Y.
Barbieri, John David (J'38) (OGHKM), Mar. Engr., N.Y. Naval Shipyard, Brooklyn 1; *for mail,* 292 Baltic St., Brooklyn 2, N.Y.
Barbour, Dana Little ('29) (SNCK), Mgr. Prod. Engrg., Elliott Co., Jeannette; *for mail,* 515 Guthrie St., Greensburg, Pa.
Barbour, Robert ('16; '24) (SREKT), Retired; Box 374, Manhasset, N.Y.
Barbour, Russell C. (J'43) (EAKSB), Crescent Bluff Ave., Pine Orchard, Branford, Conn.
Barclay, Elbert Hubbard (J'38) (LBKFZY), Engr., Maint. Engrg. Dept., Minn. & Ont. Paper Co.; *for mail,* 328—3rd St., International Falls, Minn.
Bardoff, Louis F. (J'37) (RSHCEO), 1st Lt.; *for mail,* 250 Magellan Ave., San Francisco 16, Calif.
Bardwell, Arthur Glenn, Jr. (J'41) (KABJNO), Capt., Army Air Forces, Project Engr., Air Tech. Serv. Comd., Wright Field, Dayton; *for mail,* 200 Lexington Ave., Dayton 7, Ohio.
Bareis, Felix ('23; '35) (RS), Asst. Engr., N.Y. Cent. R.R., 1824 W. 3rd St., Cleveland 13, Ohio.
Barfus, Jerome (J'41) (ANOBJY), Tool Engr., Grumman Aircraft Engrg. Corp., Bethpage, L.I.; *for mail,* 2163 Morris Ave., New York, N.Y.
Barger, Lorin W. ('13) (DJR), Works Engr., Symington-Gould Corp., Rochester; also Gould Coupler Corp., Depew; *for mail,* 32 Court St., Lancaster, N.Y.
Barickman, Harold Gilbert (J'40) (ABH), Sr. Contact Engr., Lockheed Aircraft Corp., Box 551; *for mail,* 827 E. Olive Ave., Burbank, Calif.
Bariffi, Herbert Frank ('30; '45) (AB), Wallace Metal Products Co., Inc., 50 Greene St., New Haven; *for mail,* 75 Park Ave., Hamden 14, Conn.
Barish, Herman Bernard (J'44) (CDLMN), Time Study Engr., Century Metal Craft Corp., 6000 Avalon Blvd., Los Angeles; *for mail,* 157 S. Berendo, Los Angeles 4, Calif.
Barish, Norman N. (J'44) (CDT), 41 Sellers Ave., Millbourne (Upper Darby), Pa.
Bark, Melvin Elmer ('13) (JLSAEZ), Mech. Engr., Naval Shipyard, Philadelphia; *for mail,* Grove Ave., Southampton, Pa.
Barkan, Harold (J'38) (AE), Admin. Procurement Inspr., War Dept., Army Air Forces, Army Office, c/o Wright Aero. Corp., Wood-Ridge; *for mail,* 2-15 Bellair Ave., Fair Lawn, N.J.
Barker, Curtis Herbert, Jr. ('43) (DWYARC), Dir. Research & Serv., Pallet Sales Co., Rm. 3208, 122 E. 42nd St., New York 17, N.Y.
Barker, George Sands ('16; '39) (ACJMOS), Lt. Comdr., U.S.N., Res. Inspr. of Naval Matl., 291 Kossuth St., Bridgeport 8; *for mail,* R.F.D. 1, Bridgeport, Pa.
Barker, Harry ('22) (CEFHKM), Cons. Engr., Barker & Wheeler, Engrs., 11 Park Pl., New York 7, N.Y.
Barker, Herbert ('34; '35) (TW), c/o Cia. Chilena de Tejidos, Casilla 2-D, Santiago, Chile, S.A.
Barker, John Laurance, Jr. (J'37) (JLNB), Sr. Design Engr., Carbide & Carbon Chemicals Corp., 437 McCorkle Ave.; *for mail,* 210 Joseph St., South Charleston 3, W. Va.
Barker, Joseph Warren ('30) (BKLCSN), Pres., Chmn. of Exec. Com., Research Corp., 405 Lexington Ave., New York 17, N.Y.
Barker, Richard H. (J'39) (SHR), Engr., Mech. Engrg. Div., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5; *for mail,* Box 16, Moylan, Pa.
Barker, Virgil D. ('24; '33) (JYBMHA), Pres., Barker Engrg. Corp., Fairfield Ave., Kenilworth, N.J.
Barkley, John F. ('18; '26) (FSE), Ch., Div. of Solid Fuels Utilization for War, Bur. of Mines, Interior Bldg., Washington 25, D.C.
Barkley, Kenneth Lester ('44) (KNOBLS), Designing Draftsman, Ecusta Paper Corp., Box 200, Pisgah Forest, N.C.
Barkley, Walter Russell ('24; '25; '35) (EJYMA), Design Engr., Gen. Motors Corp., Gen. Motors Bldg., Detroit 2; *for mail,* 883 Henrietta St., Birmingham, Mich.
Barkovitz, William Ernest (J'46) (JLS), Application Engr., Am. Blower Corp., 8111 Tireman, Detroit; *for mail,* 1021 Harrison Ave., Lincoln Park 25, Mich.
Barkstrom, Edward Carl ('23; '35) (DCMG), Ch. Engr., Stephens-Adamson Mfg. Co., 2227 E. 37th St., Los Angeles; *for mail,* 2105 Sherwood Rd., San Marino 9, Calif.
Barley, Louis Jean (J'40) (SK), Cadet Engr., Indianapolis Power & Light Co., 17 N. Meridian St.; *for mail,* 133 N. 18th St., Beech Grove, Ind.
Barlow, DeWitt Dukes, Jr. (J'41) (HBSK), *Student Award,* '36; 1015 Prospect Ave., Plainfield, N.J.
Barlow, Edwin H. ('13; '23) (SEOCLM), Cons. Engr., Stand. Oil Devel. Co., 26 Broadway, New York 4, N.Y.
Barmington, Raymond Dedrith ('42) (YRC), Asst. Research Engr., Colo. Agric. & Mech. College, Ft. Collins, Colo.
Barnard, John A. ('22; '35) (FS), Supt., Coal Bur. & Steam Htg., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.

Barnard, Niles Hutton ('30; '37) (CMJRL), Assoc. Prof. Mech. Engrg., Univ. of Neb., Lincoln 8, Neb.
Barnard, Norris C. ('20; '25; '35) (CFAERS), Dist. Sales Mgr., Colonial Beacon Oil Co., Box 1239, Syracuse 1; *for mail,* 400 Rugby Rd., Syracuse 6, N. Y.
Barnard, W. Grover (A'22) (CFKNS), Bldg. Supt., Hartford Elec. Light Co., Hartford P.O. 15, Hartford, Conn.
Barnard, William Nichols ('00; '05) (SAEFBN), Dir., Sibley Sch. of Mech. Engrg., Cornell Univ.; *for mail,* 201 Bryant Ave., Ithaca, N.Y.
Barnes, Fuller F. (A'19) (ACJ), Pres., Associated Spring Corp., 18 Main St., Bristol, Conn.
Barnes, George Eric ('37) (HO), Prof. Hyd. & Sanitary Engrg., Head, Dept. of Civ. Engrg., Case Sch. of Applied Sci., Cleveland, Ohio.
Barnes, Gladeon Marcus ('44), Maj. Gen., Hunting Park Ave. Plant, Edw. G. Budd Mfg. Co., Philadelphia, Pa.
Barnes, Horace Bricknell ('22; '35) (BH), Supvr. Mech. Drafting, Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Barnes, Howel H., Jr. ('10), Retired; 20 E. 76th St., New York 21, N.Y.
Barnes, James Francis (J'45) (MJNLBC), Process & Methods Engr., Wilmington Shops, E. I. du Pont de Nemours & Co., Maryland & South Sts., Wilmington 5, Del.; *for mail,* 3420 Decatur St., Philadelphia 36, Pa.
Barnes, Joseph Mendon ('21; '28) (SFCK), Engr. in Charge, Drafting Div., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Barnes, Ralph M. ('25; '29; '38) (CMLDWJ), Prof. Indus. Engrg., Dir. Personnel, College of Engrg., Univ. of Iowa, 122 Engrg. Bldg., Iowa City, Iowa.
Barnes, Wilbur James Almon ('43) (RSWEF), Mech. Engr., Bur. of Loco. Inspc., Interstate Commerce Comm., 12th & Constitution Ave., Washington 25, D.C.
Barnes, William J. (J'38), Mech. Engr., Mch. Devel., E. I. du Pont de Nemours & Co., 1007 Market St., Wilmington 98, Del.; *for mail,* 300 Yale Ave., Swarthmore, Pa.
Barnett, Eric Thornhill ('45; A'45), Asst. Gen. Mgr. Sales, Columbia Steel Co., 1410 Russ Bldg., San Francisco, Calif.
Barnett, James M. (J'30) (BEM), Lane Wells Co., 5610 S. Soto St., Los Angeles, Calif.
Barnett, Sydney A. ('38) (MACZ), Pres., Ray Proof Corp., 330 E. 26th St., New York 10, N.Y.
Barnett, Roger Marshall (J'46) (ABO), Aerodynamic Research & Design (Supersonics) Dept., Calif. Inst. of Tech., Pasadena; *for mail,* 998 San Pasqual, Pasadena 5, Calif.
Barnett, Sydney Webb (J'45), V.P., Charge of Engrg., All Fla. Automatic Sprinkler Corp., Box 730, Route 5, Barclay Rd., Tampa 4, Fla.
Barnett, William Robert (J'44), W.V. 43, Rm. 249, Oak Ridge, Tenn.
Barney, Jerome ('43) (TCDWSO), Supt., R. K. Laros Silk Co., Broad & Wood Sts., Bethlehem, Pa.
Barnhart, Charles Edgar (J'44), Box 151, Santa Rosa, New Mex.
Barnhart, Harry J. ('26), Engr., Charge Design, Osgood Co.; *for mail,* Box 310, Marion, Ohio.
Barningham, Charles Strawson ('30) (CJMT), Sales Mgr., New England Butt Co., 304 Pearl St., Providence 7; *for mail,* 83 Bluff Ave., Edgewood 5, R.I.
Barnsley, Herbert J. ('19; '35), V.P., Sales Div., Jenkins Bros., 510 Main St., Bridgeport 9, Conn.
Barnum, Starr Hickok ('12; '16; '35) (SRCM), Pres., Bigelow Co., Lloyd & River Sts., New Haven 3, Conn.
Baron, Burnett (J'44) (ANJBYC), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 1493 Cordova Ave., Lakewood 7, Ohio.
Baron, Jerome John (J'43), 1582 W. 3rd St., Brooklyn 4, N.Y.
Baron, Max Edward (J'45) (NAB), Design Engr., M. L. Bayard & Co., 20th & Indiana, Philadelphia; *for mail,* 4852 N. Broad St., Philadelphia 41, Pa.
Barone, Joseph John (J'43), U.S.N., 339 Willow St., Waterbury 41, Conn.
Barr, Clarence Dale ('15; '35) (CDEH), V.P., Secy., Am. Cast Iron Pipe Co., Box 2603, Birmingham, Ala.
Barr, David Matthew, Jr. (J'42) (BEK), Lt. (j.g.), U.S.N.R., LCI (L)-470, F.P.O., San Francisco, Calif.
Barr, Howard Verner ('43) (CSFZ), Opera. Supvr., Philo Plant, Ohio Power Co., Philo, Ohio.
Barr, Samuel Rothrock (J'41) (AJBL), 21 Haddon St., Bridgeport 5, Conn.
Barr, Victor Leo (J'42) (CMJNA), Dir. Engrg., Roller Bearing Co. of Am., Whitehead Rd., Trenton 3, N.J.; *for mail,* 6372 Woodbine Ave., Philadelphia 31, Pa.
Barrance, James Alfred ('29; '35) (CEFJSY), Designer, Sun Shipbldg. Co., Chester; *for mail,* 609 Prospect Ave., Prospect Park, Pa.
Barraza, Mario von Chrismar (J'45) (SRAKNO), Pvt., 9138th Tech. Serv. Unit, QM. Bd., Camp Lee, Va.; *for mail,* c/o Rensselaer Alumni Assn., 10 E. 43rd St., New York, N.Y.
Barre, Wade Edward, Jr. (J'42) (ACMBJN), Contract Administrator, Airplane Div., Curtiss-Wright Corp., Columbus; *for mail,* Apt. C, 899 Kenwick Rd., Columbus 9, Ohio.
Barrett, Andrew E. ('31; '35) (BCM), Cons. Tech. Designing & Erecting Mech. Engr., Bussman Mfg. Co., 2536w University St.; *for mail,* 4353 Forest Park Blvd., St. Louis, Mo.
Barrett, Dwight Orion ('12; '26) (EHN), Ch. Mech. Engr., Gulf Oil Corp., Box 661, Tulsa 2; *for mail,* 5531 S. Peoria Ave., Tulsa 15, Okla.
Barrett, Jack William (J'43), 2135 E. 13th St., Brooklyn 29, N.Y.
Barrett, LeRoy (J'40) (NCHJ), Ch. Engr., Farm Equip. Div., Ellinwood Industries, 150 W. Slauson Ave., Los Angeles 3; *for mail,* 2240 Northside Dr., Montebello, Calif.
Barrett, Robert David (J'41) (BCM), Design Analyst, Internatl. Harvester Co., 2626 W. 31st St., Chicago 8; *for mail,* 617 N. Cicero St., Chicago 44, Ill.
Barrett, Robert Edward, Jr. ('45) (CSFH), Pres., Treas., Holyoke Water Power Co., 1 Canal St., Holyoke; *for mail,* 92 Woodbridge St., South Hadley, Mass.
Barrett, William Felton ('15) (C), V.P., Dir., Union Carbide & Carbon Corp., 30 E. 42nd St., New York 17, N.Y.
Barretta, Val Shield ('45), Boiler Rm. Engr., Detroit Edison Co., Drawer E, Trenton, Mich.
Barrie, John Gregg (J'39) (SEF), Asst. Mech. Engr., Gibbs & Hill, Inc., Pennsylvania Sta., New York 1; *for mail,* 266 Lee Ave., Yonkers 5, N.Y.
Barriger, John Walker, III ('45) (R), Mgr., Diesel Loco. Div., Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5; *for mail,* 255 Sheridan Rd., Glencoe, Ill.
Barron, Claude Mitchell ('10; '14; '35) (R), Gen. Pur. Agt., Consltd. Railroads of Cuba, 70 E. 45th St., New York 17, N.Y.
Barron, Donald B. (J'35), Plant Engr., Crawford Steel Fdy., Bucyrus, Ohio.
Barron, Hugh J. (J'42), Tech. Supvr., Design & Layout Work, Fdy. Div., Waterman Steamship Corp., Mobile; *for mail,* 307½—7th St., Chickasaw, Ala.
Barron, John I. (J'41) (A), Student Engr., Bell Tel. Co. of Pa., 416—7th Ave.; *for mail,* 3137 Pioneer Ave., Pittsburgh 26, Pa.
Barron, John T. (J'44) (CJDN), Ch. Engr., Paragon Gear Works, Inc., Cushman St., Taunton, Mass.
Barrow, Chas. J. ('27) (SLFEO), Engr., 50 State St., Albany 7, N.Y.
Barrows, Donald Stuart ('21) (CJMR), V.P., Symington-Gould Corp., 25 Lincoln Ave., Box 993, Rochester 3, N.Y.
Barrows, Walter Irving ('30) (FHB), Owner, W. I. Barrows & Associates, 1169 Reibold Bldg., Dayton 2, Ohio.
Barry, Edward H. ('39) (SYO), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Barry, Frank William (J'45) (SEK), Graduate House, Mass. Inst. of Tech., Cambridge 39, Mass.
Barry, James Milne ('23) (SCFEDB), V.P., Gen. Mgr., Ala. Power Co., 600 N. 18th St., Birmingham 2, Ala.
Barry, John S. C. ('46) (BMT), Product Engr., Atwood Div., Farrel-Birmingham Co., Inc., Stonington; *for mail,* 8 Burdick Lane, Pawcatuck, Conn.
Barry, R. E. ('37), Retired; 320 Pequot Ave., New London, Conn.
Barry, Richard W. (J'42) (CJM), Lt., 359th Sqd., Army Air Base, Alamogordo, New Mex.
Barry, T. J. ('17; '21; '35) (CDKFSJ), Owner, Firm of T. J. Barry, Park Bldg., Pittsburgh 22, Pa.
Barry, William Burris ('34; '35) (BHJKNY), Chief Engrg. Design Sec., Jet Propulsion Lab., Calif. Inst. of Tech., Pasadena; *for mail,* 279 Naomi Ave., Arcadia, Calif.
Barsell, Birger Edwin (J'40) (AZBCJM), Seaman, 1st Class, Project Engr., Exper. Dept., Naval Ord. Test. Sta., Inyokern, Calif.
Barta, George L., Jr. (J'43), 41-72 Glenwood St., Little Neck, N.Y.
Barta, James E. (J'40) (AESC), Asst. to Ch. Exper. Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 130 Market St., Passaic, N.J.
Bartak, A. M. ('43) (TBJ), Tenn. Eastman Corp., Kingsport; *for mail,* 1670 Woodside Ave., Kingsport, Tenn.
Barten, Edward A. ('21; '45) (BHLSM), Ch. Engr., Hunter Mfg. Corp., Croydon; *for mail,* 921 Ashbourne Rd., Cheltenham, Pa.
Barten, Henry John (J'44) (BNAJZL), Asst. Engr., Armour Research Foundation, 35 W. 33rd St., Chicago 16, Ill.
Barth, Robert Charles (J'41) (ABE), Asst. Project Engr., Ranger Aircraft Engs. Div., Fairchild Eng. & Aircraft Corp., Farmingdale; *for mail,* Box 444, Huntington, L.I., N.Y.
Barthel, Oliver Edward ('21) (BNOEAY), Cons. Mech. & Automotive Engr., Firm of Oliver E. Barthel, 2323 Dime Bldg., Detroit 26, Mich.
Barthelemy, Carl Rudolph ('45) (SJFC), Plant Results Engr., No. States Power Co., High Bridge Sta., St. Paul 2, Minn.
Bartholomaei, Herman Alamo ('44) (CMDOAY), Budget Analyst, Bendix Radio, Towson 4; *for mail,* 3750 Old York Rd., Baltimore 18, Md.
Bartholomew, Earl ('30; '35) (FEAC), Gen. Mgr., Research Labs., Ethyl Corp., 1600 W. Eight Mile Rd., Detroit 20, Mich.
Bartle, George Richard (J'41) (KSCJN), Design Engr., Thomas C. Wilson, Inc., 21-11—44th Ave., Long Island City 1; *for mail,* 51-55—45th St., Woodside, L.I., N.Y.
Bartlett, Bradley Strough (J'42), Engr., Foote Bros. Gear & Mch. Co., 5301 S. Western Blvd., Chicago 9; *for mail,* 4461 Johnson Ave., Western Springs, Ill.
Bartlett, Henry ('97) (RS), 989 Memorial Dr., Cambridge 38, Mass.
Bartlett, John Minot, Jr. (J'42) (JMNC), Supt., Washburn Shops, Worcester Poly. Inst., Worcester 2; *for mail,* 143 W. Boylston St., Worcester 5, Mass.
Bartlett, Luis Hamilton ('43) (JKL), Assoc. Prof. Mech. Engrg., Univ. of Tex., Austin 12 & McEvoy Co., 600 Milby St., Houston, Tex.
Bartlett, Parker M. (J'46), Tech. Advis. Bd., J. P. Riddle Co., Box 4632, Coral Gables, Fla.; *for mail,* 1427 Beecher St., S.W., Atlanta, Ga.
Bartlett, Robert Danforth (J'45) (KY), U.S. Navy; 204 May St., Worcester 2, Mass.
Bartlett, Robert Lawrence (J'45) (HESONJ), Ensign, U.S.N., Engr. Officer, PC-1077, F.P.O., San Francisco, Calif.; *for mail,* 128 James River Dr., Hilton Village, Va.
Bartlett, Robert White (J'42) (MCJ), Box 342, R.D. 1, Bound Brook, N.J.
Bartley, Charles Owen (J'42), Ensign, Dept. of Mar. Engrg., U.S. Naval Acad., Annapolis, Md.
Bartling, Homer L. (J'38) (CKL), Maj., Ord. Dept., Ord. Sec. Serv. Comd., A.P.O. 708, c/o Postmaster, San Francisco, Calif.
Bartmess, John Edward (J'41) (AS), Indus. Sales, Cincinnati Dist. Office, Westinghouse Elec. Corp., 207 W. 3rd St., Cincinnati 2; *for mail,* 531 Terrace Ave., Cincinnati 20, Ohio.
Barto, Morris J. (J'44), 92—3rd St., Carneys Point, N.J.
Bartolero, Carlo (J'30) (BFH), Ch. Engr., United Iron Works, 580—2nd St.; *for mail,* 6116 Contra Costa Rd., Oakland, Calif.
Barton, Albert Radford ('36; '41) (CMLD), Asst. Supt., E. J. Brach & Sons, 4656 W. Kinzie St., Chicago 44, Ill.
Barton, Edward Allen ('42) (LHECBK), 31 Park Hill Ave., Norwalk, Conn.
Barton, Millard Vernon ('42) (AB), Prof. Aero. Engrg., Research Engr., Defense Research Lab., Univ. of Tex., Austin 12, Tex.
Barton, Richard B. (J'36) (BSCN), Mech. Engr., Turbine Div., Gen Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 19 Knickerbocker Rd., Scotia 2, N.Y.
Bartow, Paul Lockwood ('44) (SOKDJF), Plant Constr. Engr., Fla. Power Corp.; *for mail,* 977—8th Ave. S., St. Petersburg, Fla.
Bartram, Paul Robert ('34; '44) (CMOALW), Asst. Factory Controller, Rudolph Wurlitzer Co., Inc., North Tonawanda; *for mail,* 84 Orchard Dr., Kenmore 17, N.Y.
Bartsch, Arthur G. ('28; '35), Propr., Enterprise Tool & Mfg. Co., Newark; *for mail,* Forest Rd., Essex Fells, N.J.
Bartusek, Robt. J. (J'40) (JMS), Research Engr., Armour Research Foundation, 35 W. 33rd St.; *for mail,* 2537 S. Drake Ave., Chicago, Ill.
Baruch, Milton ('14; '21; '35) (DCLBEH), V.P., Baruch Corp., 625 S. Olive St., Los Angeles 14, Calif.
Barzen, Richard Gregory ('24; '30; '35) (CDHM), Pres., Sterling Mchy. Corp., 411 Southwest Blvd., Kansas City 10, Mo.
Basche, Harold Sheldon (J'44) (CMLJZG), Quality Control Engr., RCA Victor Div., Radio Corp. of Am., Harrison, N.J.; *for mail,* 199 E. 2nd St., Brooklyn, N.Y.
Basci, Joseph (J'42) (BS), Asst. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York; *for mail,* 1859—72nd St., Brooklyn, N.Y.
Bascom, Jos. H. (J'37) (CDJ), Engr., Broderick & Bascom Rope Co., 4203 N. Union Blvd., St. Louis, Mo.
Bascombe, Frank J. ('39) (R), Apt. 701, Palace Hotel, Long Beach 2, Calif.
Bascome, George Lightbourn ('19) (SEFBHC), Valuation Engr., State Corp. Comm., State Office Bldg., Richmond 19, Va.
Baselt, Walter H. ('43) (RNAB), Mech. Asst. to V.P., Am. Steel Fdys., 410 N. Michigan Ave., Chicago; *for mail,* 1448 Braeburn Rd., Flossmoor, Ill.
Baser, Fred Richard (J'46), Research Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 288 W. Scott Ave., Rahway, N.J.
Baserman, Kenneth James (J'44), c/o Sperry Gyroscope Co., Inc., 55 Johnson St., Brooklyn 1, N.Y.
Bashen, George B. ('39) (FS), Ch. Engr., Fisk Sta. Commonwealth Edison Co., 1111 W. Cermak Rd., Chicago 8, Ill.
Baskerville, Ralph James ('41) (BJNAMS), Mech. Engr., A. B. Chance Co., Centralia, Mo.
Bass, George Ebert ('43) (ACDJNR), Pres., Ferracute Mch. Co., E. Commerce St., Bridgeton, N.J.
Bass, Russel Belknap ('36; '39) (CMDJ), Plant Engr., RCA Victor Div., Radio Corp. of Am., 5750 W. 51st St., Chicago 38; *for mail,* 233 N. Harvey, Oak Park 1, Ill.

Bassett, Burdett E. ('21; '31) (JMNC), Research & Devel. Dept., Olin Industries, Inc., East Alton; *for mail,* 1212 Callahan Dr., Alton, Ill.
Bassett, George Barclay ('14) (CHJYZ), Pres., Buffalo Meter Co., 2917 Main St., Buffalo; *for mail,* 270 Depew Ave., Buffalo 14, N.Y.
Bassett, Royal Marion ('14; '35) (AEFKOS), Dist. Mgr., Riley Stoker Corp., 12 S. 12th St., Philadelphia 7, Pa.
Bassett, Walter G. R. ('29) (LADSZ), Field Engr., E. R. Squibb & Sons, Georges St., New Brunswick, N.J.
Bassett, William Varick (J'41) (SBR), Devel. & Research Branch, Cent. Tech. Dept., Shipbldg. Div., Bethlehem Steel Co., E. Howard St.; *for mail,* 29 Edison Pk., Quincy 69, Mass.
Bastante, Alejandro M. ('43) (BCKM), Mgr., Arsenal Naval Del Callao, Peruvian Navy; *for mail,* Apartado 154, Callao, Peru, S.A.
Bastian, Alvin Lewis (J'44) (ABDMNX), Staff Engr., Firestone Tire & Rubber Co., Firestone Pkwy., Akron; *for mail,* 1010 Packard Dr., Akron 2, Ohio.
Bastian, Earl Louis Henry ('44) (EMNJCH), Indus. Lubricants Engr., Shell Oil Co., Inc., 50 W. 50th St., New York 20; *for mail,* Scarsdale Chateaux, Scarsdale, N.Y.
Basuino, Francis Michael, Jr. (J'44) (SKFEB), Engrg. Specialist, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 305—95th St., Brooklyn 9, N.Y.
Bataille, Eugene Jerome (J'39) (AHBCKM), Jr. Design Engr., Peerless Pump Div., Food Mchy. Corp., 301 W. Ave. 26, Los Angeles; *for mail,* 448 N. Las Palmas, Los Angeles 4, Calif.
Batchelder, Lew Alan (J'40) (ZMC), Engr., Gen. Elec. Co., 40 Federal St., West Lynn; *for mail,* Box 80, Lake St., Middleton, Mass.
Batchelder, Nelson A. ('20), 180 Division St., Welland, Ont., Can.
Batchelor, Robert Barlow (J'44) (ANOB), Devel. Design Engr., Eng. Div., Chrysler Corp., Highland Park; *for mail,* 202 Colorado Ave., Highland Park 3, Mich.
Batcock, Charles Edward ('44) (BMNDJC), Mech. Supt., Canadian Gen. Elec. Co., 940 Lansdowne Ave.; *for mail,* 61 Yonge Blvd., Toronto, Ont., Can.
Bateman, Edward Latrobe (A'97) (DEJWZ), Managing Dir., Edward L. Bateman (Pty.) Ltd., 90 Fox St., Johannesburg, Transvaal, S. Africa.
Bateman, Geo. F. ('21) (EFS), Dean, Sch. of Engrg., Prof. Mech. Engrg., Cooper Union, Cooper Sq., New York, N.Y.
Bateman, Thomas Peter ('30; '35) (FKS), West. Serv. Engr., Consolidation Coal Co., 1832 Buhl Bldg., Detroit 26; *for mail,* 1347 W. Ann Arbor St., Plymouth, Mich.
Bates, Albert Edward ('38) (CDMZY), Prod. Planning Mgr., Roller-Smith Div., Realty & Indus. Corp., 1756 Market St., Bethlehem, Pa.
Bates, Albert H. ('90; '12), Partner, Bates, Teare & McBean, Pat. Lawyers, 1125 Terminal Tower Bldg., Cleveland, Ohio.
Bates, Arthur C. ('36; '42), Roller-Smith Co.; *for mail,* 709 Prospect Ave., Bethlehem, Pa.
Bates, Charles Jarvis ('13), Retired; 473 Tenafly Rd., Englewood, N.J.
Bates, Daniel Moore ('12) (CTMDSZ), Pres., Cold Spring Bleachery, Yardley, Pa.
Bates, Douglas Petit (J'40) (YCRE), Engr., Process Control Dept., Stand. Oil Co. of N.J., Bayway Refinery, Elizabeth; *for mail,* Box 724, Short Hills, N.J.
Bates, Erastus Newton ('17) (NGDW), Equip. Supvr., Pacific Coast Hdq., Grain Products Div., Prod. & Marketing Admin., U.S. Dept. of Agric., 345 U.S. Courthouse, Portland 5, Ore.
Bates, George Harold ('14) (CDM), Gen. Mgr., N.Y. Shipyards, Bethlehem Steel Co., 25 Broadway, New York 4, N.Y.; *for mail,* 18 Hampton St., Cranford, N.J.
Bates, Harvey C. (J'35) (NJMC), Mech. Maint. Foreman, Socony-Vacuum Oil Co., Inc., Box 546; *for mail,* 337 Main St., Augusta, Kan.
Bates, James Hervey Simpson ('14) (O), Cloverfields Farm, Olympia, Wash.
Bates, Joseph Clark, Jr. (J'44), Lt. (j.g.), U.S.S. *Sealion,* F.P.O., New York, N.Y.
Bates, Nathan W. (J'35) (EHS), Plant Oper., Cohasset Water Co., South St., Hingham; *for mail,* 683 Beechwood St., Cohasset, Mass.
Bates, Ralph E. (J'40) (JZSC), Engr., Procter & Gamble Mfg. Co., 1232 W. North Ave., Chicago 22; *for mail,* 8112 Clyde Ave., Chicago 17, Ill.
Bates, Robert Edgar (J'39) (JMG), Mch. Shop Foreman, Bethlehem Steel Co., E. 3rd St.; *for mail,* 1231 Monocacy St., Bethelehem, Pa.
Bates, Rodman H. ('44) (ARS), 368 Noah Ave., Akron, Ohio.
Bates, Thomas M. (J'42), 24 Davis Ave., Auburn, Me.
Batesole, Dwight E. ('20; '29) (CNMZBA), V.P., Ch. Engr., Norma-Hoffmann Bearings Corp., Stamford, Conn.
Bateson, Harvey Wayland, Jr. (J'43) (AJM), c/o Carl L. Norden, Inc., 80 Lafayette St., New York 13, N.Y.
Batie, Joseph E. ('28), M.M., Kelsey-Hayes Wheel Corp., 3600 Military Ave., Detroit, Mich.
Batiuk, Martin (J'39) (NYBD), Devel. Engr., B.F. Goodrich Co., 500 S. Main St., Akron 11; *for mail,* 1020 Brown St., Akron 1, Ohio.
Bato, Andrew A. ('24; 27) (SOFEHK), Cons. Engr., 227 Park Ave., East Orange, N.J.
Batson, Jack E. D. ('46), Prod. Mgr., Kittinger Co., Inc., 1893 Elmwood Ave., Buffalo 7; *for mail,* 301 North Dr., Buffalo, N.Y.
Batt, William Loren ('11; '26; F'38; H'42) (CR), *Manager,* '30-'33, *Vice-President,* '33-'35, *President,* '36; Pres., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34, Pa.
Battey, Paul Leon ('22) (ORLSFK), Partner, Battey & Childs, 231 S. La Salle St., Chicago 4, Ill.
Battey, William Aldrich ('19; '35) (CDL), V.P., Pa. Crusher Co., Liberty Trust Bldg., Philadelphia 7, Pa.
Battle, John Rome ('18; '21) (EFOSA), Mech. Engr., J. R. Battle & Co., T-M Bldg., Ardmore, Pa.
Battley, Edwin Roy ('43) (RSJFEK), Chief of M.P. & Car Equip., Canadian Natl. Rys., 360 McGill St., Montreal 1, Que., Can.
Batty, Stanley C. (J'38) (BGM), Ch. Test Engr., Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90; *for mail,* 1118 N. 11th Ave., Melrose Park, Ill.
Batu, Ahmet Muhtar (J'41) (AJS), Asst. Test Engr., Aircraft Eng. Div., Packard Motor Car Co., 1580 E. Grand Blvd., Detroit 32; *for mail,* 138 Gladstone, Detroit 2, Mich.
Bauder, Burton Eberman (J'45) (MCEFJK), Engr., West. Elec. Co., Inc., 100 Central Ave., Kearney; *for mail,* 52 Lexington St., Newark 5, N.J.
Baudry, René André ('35; '35) (BNZAHK), Mgr., Mech. Devel. Sec., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 1200 Graham Blvd., Pittsburgh 21, Pa.
Bauer, Charles August ('25; 46) (YBGJ), V.P., Charge of Devel., Hoosier Cardinal Corp., Eichel Ave., Evansville 7, Ind.
Bauer, Charles Louis ('00; '01) (BCLNWY), Pres., Bauer Bros. Co., Sheridan Ave., Springfield, Ohio.
Bauer, Ernest Kurt ('31; '34; '35) (TNB), Devel. Engr., Am. Viscose Corp., Meadville, Pa.
Bauer, Frank Stanley ('20; '35) (NBGESM), Head of Dept., Prof., Engrg. Drawing & Mch. Design, Univ. of Colo.; *for mail,* 944 Lincoln Pl., Boulder, Colo.
Bauer, Harry John ('23; '35) (CFMS), V.P., N.Y. Steam Corp., 4 Irving Pl., New York, N.Y.; *for mail,* 212 Knickerbocker Rd., Tenafly, N.J.
Bauer, Jacob Louis (J'37) (CJ), Asst. Regional Engr., Stand. Oil Co. of N.J., Bayway Refinery, Elizabeth; *for mail,* 315 Mercer Ave., Roselle, N.J.
Bauer, Jacob R. (J'28), Mech. Engr., Procter & Gamble Co., Ivorydale; *for mail,* R.R. 9, Box 432, College Hill Sta., Cincinnati, Ohio.
Bauer, Peter W. ('21; '27; '35) (GNBJLA), Asst. Dir., E. C. Goodwin Tech. Sch., New Britain; *for mail,* 100 Slater Rd., New Britain, Conn.
Baugh, Everett L. (J'39) (BEH), Moraine Products Div., Gen. Motors Corp., 1420 Wisconsin Blvd., Dayton; *for mail,* 451 E. Linden Ave., Miamisburg, Ohio.
Baugh, Herbert Hill (J'38) (AEBKSN), Analytical Engr., Lycoming Div., Aviation Corp., Williamsport 38; *for mail,* 638—3rd Ave., Williamsport, Pa.
Bauhan, Alexander E. ('14; '17; '35) (SCOERW), Engrg. Asst. to Gen. Auditor, Pub. Serv. Elec. & Gas Co., 80 Park Place, Newark 1; *for mail,* 27 Virginia Rd., Maplewood, N.J.
Baum, Edwin Preston ('37) (AEHNOS), Ch. Erecting Engr., Hardie-Tynes Mfg. Co., 800 N. 28th St., Birmingham; *for mail,* 3031 N. 11th Ave., Birmingham 4, Ala.
Baum, Richard Theodore (J'42) (KBHZAC), Engr., Jaros, Baum & Bolles, 415 Lexington Ave., New York 17; *for mail,* 600 W. 111th St., New York 25, N.Y.
Bauman, Jacob Arthur ('36; '45) (SHEZK), Mar. Engr., N.Y. Naval Shipyard, Brooklyn 1; *for mail,* 201 Clinton Ave., Brooklyn 5, N.Y.
Bauman, Wallace Harris (J'42) (SA), Engr., Westinghouse Elec. Corp., c/o Hawaiian Elec. Co.; *for mail,* 2723-B Aolani Pl., Honolulu, T.H.
Bauman, Willard Edward (J'41) (AMECF), Box 661, Tulsa, Okla.
Baumann, George William ('21; '27; '35) (SOFEC), Pur. Engr., Long Island Ltg. Co., 250 Old Country Rd., Mineola, L.I.; *for mail,* 36 Curtis Pl., Lynbrook, L.I., N.Y.
Baumeister, Paul A. ('18; '35) (CMOS), 33-57—162nd St., Flushing, N.Y.
Baumeister, Theodore ('23; '30) (SEFRBO), Prof. & Exec. Officer, Mech. Engrg. Dept., Columbia Univ., 120th St. & Broadway, New York 27, N.Y.
Baumgartner, Charles Greene ('13) (DBEFHK), Ch. Mech. Engr., Am. Bridge Co., Park Rd. & 4th St., Ambridge; *for mail,* 233 Hilands Ave., Ben Avon, Pittsburgh 2, Pa.
Baumis, Frank Joseph ('43) (BJOMH), Pres., Baumis Mfg. Corp., Main St., Townsend, Mass.
Bausch, Carl L. ('11; '13; '21; F'41) (CNJZ), *Manager,* '37-40; V.P., Charge Research & Engrg., Bausch & Lomb Optical Co., 635 St. Paul St., Rochester 2, N.Y.
Bausch, William George (J'41) (MJCZYA), Engr., Trainee, Instrument Div., Bausch & Lomb Optical Co., 635 St. Paul St., Rochester 2; *for mail,* 153 Landing Rd. N., Rochester, N.Y.
Bavetta, Vincent Edward (J'45) (ABCN), Design Engr., Norden Labs. Corp., 34 Varick St., New York 13; *for mail,* 87 Crooke Ave., Brooklyn 26, N.Y.
Baxa, James F. (J'40) (BJL), c/o Frank F. Le Van, 8519—108th St., Richmond Hill, L.I., N.Y.
Baxley, C. Herbert ('22; '27; '30) (AE), V.P., Tech. Mgr., Intava, Inc., 25 Broad St., New York 4, N.Y.
Baxter, Allan H. ('27), Designing & Cons. Engr., A. E. Baxter Engrg. Co., 344 Delaware Ave., Buffalo 2, N.Y.
Baxter, Edward Dows (J'36) (OBDCM), A. E. Baxter Engrg. Co., 344 Delaware Ave., Buffalo 2, N.Y.
Baxter, James W. ('27) (JMNB), Mech. Engr., Natl. Tube Co.; *for mail,* 513 Pershing St., Ellwood City, Pa.
Baxter, Meriwether Lewis (J'35) (NB), Mech. Engr., Gleason Works, 1000 University Ave., Rochester; *for mail,* 73 Washington Dr., Rochester 10, N.Y.
Baybutt, John William ('42) (MNKZYJ), Mech. Engr., Eastman Kodak Co., Kodak Park Works, Rochester 4, N.Y.
Bayer, Anthony Ralph (J'43) (CYSJM), Maj., QM. Corps, Memorial Div., Office of QM. Gen.; *for mail,* 1412 Montague St., Washington, D.C.
Bayer, Lloyd Felch ('15; '22; '35) (CEAFLK), V.P., Tide Water Associated Oil Co., 79 New Montgomery St., San Francisco 20, Calif.
Bayles, Allison Lerch ('44) (CODHFE), V.P., Am. Engrg. Co., 2447 Aramingo Ave., Philadelphia; *for mail,* Sproul Rd., Ithan, Pa.
Bayles, Charles B. (J'40) (LKD), Equip. Layout Engr., Wallace & Tiernan Products, Inc., 11 Mill St.; *for mail,* 298 Union Ave., Belleville 9, N.J.
Bayles, William H. (J'45) (CER), 27 Park Pl., Bloomfield, N.J.
Bayliss, Benjamin Philip (J'38) (EFP), Engr., Designer, West. Oil Tool & Engrg. Co., Inc., 4400 San Fernando Rd., Glendale 4; *for mail,* 1879 E. Mendocino St., Altadena, Calif.
Bayliss, William Arthur ('29) (SFE), Ch. Inspr., Hartford Steam Boiler Inspc. & Ins. Co., 87 Kilby St., Boston 9, Mass.
Bayne, Carl R. (J'37) (EHP), Lt., Co. G, 142nd Inf., Camp Bowie, Tex.
Bayrer, L. Garfield ('18), Works Mgr., Blakeslee Forging Co., Plantsville, Conn.
Beach, Charles S. (J'85) (BMW), Owner, Beach's Shop, 753 Main St., Bennington, Vt.
Beake, Laurence (J'41), Detailer, Meisel Press Mfg. Co., 942 Dorchester Ave., Boston; *for mail,* 77 Bay State Ave., West Somerville, Mass.
Bealing, Ernest ('36) (BCD), London Rep., Spencer (Melksham) Ltd., Melksham, Wilts; *for mail,* "Coniston," 50 Woodway Crescent, Kenton, Harrow, Middlesex, England.
Beall, Almon Lee ('30) (PAFBE), Research Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* Box 342, 20 Osborne Terrace, Packanack Lake, N.J.
Beals, Richard O. (J'34) (BCM), Box 46, Cobalt, Conn.
Beam, Albert Steven (J'43) (BMN), Gear & Gage Engr., Vinco Corp., 8855 Schaeffer Highway, Detroit 27; *for mail,* 51 Holbrook Ave., Detroit 2, Mich.
Beaman, David Webster, Jr. (J'39) (NMZYW), Design Engr., Frederick Hart Co., 837 Main St., Poughkeepsie, N.Y.
Beaman, P. Alden ('15; '35) (NCJ), Mgr., Wire Mchy. Dept., Morgan Constr. Co., 15 Belmont St., Worcester 5, Mass.
Bean, Clarence Herbert ('15) (SFLA), Supvg. Engr., Power Plants, Calco Chem. Div., Am. Cyanamid Co.; *for mail,* 410 Church St., Bound Brook, N.J.
Bean, Guy Myron ('36) (RFKS), Sales Engr., Am. Arch Co., Inc., 60 E. 42nd St., New York 17, N.Y.; *for mail,* 178 S. Ardmore Ave., Los Angeles 4, Calif.
Bean, Howard S. ('19; '26) (HBZEO), Ch., Fluid Meters Sec., Natl. Bur. of Stands., Connecticut Ave. & Upton St., N.W., Washington 25, D.C.
Bean, James ('39) (RMCJED), Asst. Supt., M.P., So. Pacific Co., 201 N. Piedras St., El Paso, Tex.
Bean, Laurence Guy ('20; '26; '35) (ZCLJ), V.P., Charge Engrg. & Sales, Bristol Co., Waterbury 91; *for mail,* 72 Hewlett St., Waterbury 30, Conn.
Bean, Nevin Lockridge ('44) (ACL), Ch. Tool Designer, Ford Motor Co., Dept. 2866, 3000 Schaefer Rd., Dearborn; *for mail,* 1362 Manistique, Detroit 15, Mich.
Bean, Philip H. ('22; '35) (FLS), Harrison Gas Works, 4th St. & Passaic River, Harrison; *for mail,* 265 Harrison Ave., Jersey City 4, N.J.
Bean, Theron W. ('31; '35) (FKS), Capt., QM. Corps, Constr. QM., Triangular Div. Camp, Medford, Ore.; *for mail,* 24 Highland Dr., Medford, Ore.
Beane, John Arthur ('44) (BOGHKN), Cons. Engr., John A. Beane & John Y. Sloan, 296 Delaware Ave., Buffalo 2, N.Y.
Beane, John R. L., Jr. (J'39) (FLS), Ensign, Bur. of Ships, Navy Dept., Naval Shipyard, Philadelphia, Pa.
Beaney, Walter E. ('30; '35) (BESKNJ), Ch. Engr., D. E. Whiton Mch. Co., 190 Howard St.; *for mail,* 37 Crest St., New London, Conn.

Beanfield, Byron F. (J'35) (COMESB), Asst. to Officer in Charge, Plant Facilities Sec., Office of Supvg. Inspr. of Naval Matl., 90 Church St., New York 7, N.Y.
Beanfield, Rufus McClellon ('31) (POSEFD), Cons. Engr., 1876 Middleton Pl., Los Angeles 37, Calif.
Bearce, George D. ('42) (CFG), Gen. Mgr., Me. Seaboard Paper Co., Bucksport, Me.
Beard, Geoffrey G. ('42) (CJNM), Dir., V.P., Exec. Asst. to Pres., United Engrg. & Fdy. Co., 2500 1st Natl. Bank Bldg., Pittsburgh, Pa.
Beard, Richard Damon (J'45), Test Engr., Gen. Elec. Co., Schenectady; *for mail,* 51 W. Balcom St., Buffalo 9, N.Y.
Beard, Theodore Hemingway ('17; '25) (ACMR), V.P., Dictaphone Corp., 375 Howard Ave., Bridgeport 5; *for mail,* R.F.D. 1, Bridgeport, Conn.
Beardsley, Harry Irving, Jr. (J'41) (AFEKZB), Flight Test Engr., Bell Aircraft Corp., Niagara Falls, N.Y.
Bearse, Loring Runyon (J'38) (CMNT), V.P., Charge Prod., Bearse Mfg. Co., 3815 Cortland St., Chicago; *for mail,* 4844 Lawn Ave., La Grange, Ill.
Bearse, Robert Emery (J'44), 179 Emerson Pl., Brooklyn 5, N.Y.
Beatie, Cecil Emile ('12; '18; '19) (LKHESF), Beatie & Delius, 586 Market St.; *for mail,* Apt. 201, 1840 Clay St., San Francisco, Calif.
Beaton, Norman H. ('39) (OLHSDC), Secy., Treas., J. M. Montgomery & Co., Inc., 306 W. 3rd St., Los Angeles 13, Calif.
Beattie, Wayne Stevenson ('37) (SACFK), Prof. Mech. Engrg., Univ. of Colo., Boulder, Colo.
Beatty, Charles Edward ('31; '39) (CDSMAF), Dist. Mgr., Pub. Serv. Co. of Colo., 721 Harrison Ave.; *for mail,* 815 Spruce St., Leadville, Colo.
Beatty, Henry Russell ('43) (CMO), Assoc. Prof. Admin. Engrg., Pratt Inst., 215 Ryerson St., Brooklyn 5, N.Y.; *for mail,* R.D. 2, Millington, N.J.
Beatty, Robert James ('46) (BCO), Dir., Fed. Glass Co., Columbus 7, Ohio.
Beatty, William Clare ('41) (NBSDOC), Plant Engr., United Engrg. & Fdy. Co., 1400 Grace Ave., N.E., Canton 5, Ohio.
Beaty, Alden Query (J'40) (NJMC), Engr., Hardinge Co., Inc., 240 Arch St.; *for mail,* 931 S. Queen St., York, Pa.
Beauchemin, Alphonse O. ('21) (BEFHSY), Pres., Univ. Pump & Water Co., Inc., 101 Cedar St., New York 6, N.Y.
Beaufrere, Albert Harry (J'38) (ABJ), Research Engr., De Laval Steam Turbine Co.; *for mail,* 1806 Pennington Rd., Trenton, N.J.
Beaumont, Jay Charles ('18; '25) (HJNE), Mech. Engr., Detroit Universal Duplicator Co., 723 E. Milwaukee Ave., Detroit; *for mail,* 1220 Beaconsfield Ave., Detroit 30, Mich.
Beaven, H. Edgar ('20; '26; '35) (CMDFOL), Staff Engr., Am. Mutual Liability Ins. Co., 142 Berkeley St., Boston; *for mail,* 756 Watertown St., West Newton 65, Mass.
Beavers, George Robert ('30; '35) (CHNBKS), 90 Chirton W. View, North Shields-On-Tyne, Northumberland, England.
Beavers, Knox Tenant (J'42) (JOEM), Metal Inspr., Humble Oil & Refining Co., Baytown; *for mail,* 317 W. Gulf St., Goose Creek, Tex.
Bechaud, Leslie J. ('37) (CM), Supt. & Partner, Pan-Am. Engrg. Co., 820 Parker St., Berkeley 2, Calif.
Bechert, Fred J. ('14; '20; '35) (M), Partner, Mitchell & Bechert, 420 Lexington Ave., New York, N.Y.
Bechtel, Luther Dunham ('38; '45) (AEFORS), Asst. to Mech. Engrg. Dir., Commercial Testing & Engrg. Co., 307 N. Michigan Ave.; Chicago 1, Ill.
Bechtold, Merlin E. (J'38) (CES), Field Engr., Allis-Chalmers Mfg. Co., 1604 Merchants Bank Bldg.; *for mail,* Box 32, R.R. 16, Indianapolis, Ind.
Beck, C. Wendell (J'42) (CM), Ensign, 65 Lake Dr., Mountain Lakes, N.J.
Beck, Carl Henry ('43) (RAHEC), Gen. Sales Mgr., Westinghouse Air Brake Co., Wilmerding, Pa.
Beck, Charles Edgar ('11; '18; '35) (EFCJSZ), Sales Engr., Busch-Sulzer Bros.-Diesel Eng. Co., 3300 S. 2nd St., St. Louis; *for mail,* 5805 Lansdown Ave., St. Louis 9, Mo.
Beck, George Doughty ('27; '30) (BEF), Erecting Supt., Buffalo Works, Worthington Pump & Mchy. Corp., Buffalo 5, N.Y.; *for mail,* 2331 Constance St., New Orleans 13, La.
Beck, Lawrence J. (J'35) (MAENBT), Cons. Engr., 10 W. 48th St., New York 19, N.Y.
Beck, Matthias A. ('86), Retired; 3070 S. Superior St., Milwaukee 7, Wis.
Beck, Rudolf ('27) (BEKLSZ), Ch. Engr., Instrument Div., Manning, Maxwell & Moore, 11 Elias St., Bridgeport 2, Conn.
Beck, Wm. H., Jr. (J'39), 4502 Mainfield Ave., Baltimore 14, Md.
Becker, Carlton Hewitt ('42) (HESK), Mech. Engr., Greeley & Hansen, 220 S. State St., Chicago 4, Ill.
Becker, Charles Samuel ('34; '45) (CMO), Personnel Mgr., Indus. Engr., Diamond T Motor Car Co., 4401 W. 26th St., Chicago 23; *for mail,* 808 S. Kensington Ave., La Grange, Ill.
Becker, Damas Warren (J'42) (BJKGH), Jr. Mech. Engr., Aluminum Co. of Am.; *for mail,* 128 Center St., Massena, N.Y.
Becker, H. Kirke ('19; '35) (CJMN), Pres., Peters Mchy. Co., 4700 Ravenswood Ave., Chicago 40, Ill.
Becker, Julius ('23), 33-51—165th St., Flushing, L.I., N.Y.
Becker, Peter M., Jr. (J'34) (BMR), Lt. (j.g.), Box ND-10, F.P.O., Balboa, C.Z.
Becker, Richard F. (J'41) (WBHK), Engr., Gibbs & Cox, Inc., 21 West St., New York 6; *for mail,* 55 Morton St., New York 14, N.Y.
Becker, William D. (J'41) (BCM), Sr. Draftsman, Locke Insulator Corp., Box 57; *for mail,* 4220 Springwood Ave., Baltimore, Md.
Becker, William M. ('37; '43) (SBKFEJ), Mech. Engr., Boiler Design, Babcock & Wilcox Co., Barberton; *for mail,* 186 Humbolt Ave., Wadsworth, Ohio.
Beckett, Charles Harrison, Jr. (J'46), 708 Winston Ave., San Marino, Calif.
Beckjord, Walter Clarence ('22) (BCEFS), Pres., Cincinnati Gas & Elec. Co., 4th & Main Sts., Cincinnati 2, Ohio.
Beckman, John George (J'45) (EHS), Brother, Rosary House, Maryknoll, N.Y.
Beckman, Leroy John (J'35) (HLS), Field Engr., Wallace & Tiernan Co., Inc., 1229 W. Washington Blvd., Chicago 7, Ill.
Becks, Douglas Edwin (J'46), 510 Cassell St., North Bay, Ont., Can.
Beckstrand, E. H. ('08) (BKS), Prof. Mech. Engrg., Univ. of Utah; *for mail,* 244 Douglas St., Salt Lake City, Utah.
Beckwith, Bernard L. ('27; '35) (REBNA), Mgr., Metro. Railcars Ltd., Paseo Colon 185, Buenos Aires, Argentina, S.A.
Beckwith, C. Gordon (J'40) (ABE), Sr. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford; *for mail,* 38 S. Hawthorne St., Manchester, Conn.
Beckwith, Charles Frederick ('42) (CFEA), East. Sales Mgr., Pump Div., Neptune Meter Co., 50 W. 50th St., New York 20, N.Y.
Beckwith, Charles Raymond (J'42) (AMYD), Engrg. Dept., Hoover Co., North Canton, Ohio.
Beckwith, E. L. ('20; '35), Dist. Mgr., Detroit Stoker Co., 333 N. Michigan Ave.; *for mail,* 2000 Lincoln Park West, Chicago, Ill.
Beckwith, Oliver Phillips (J'36) (TCMG), Ch., Prod. Engrg. Labs., Alex. Smith & Sons Carpet Co., Yonkers; *for mail,* 64 Kensington Rd., Bronxville, N.Y.
Beckwith, Thomas Gilbert ('35; '46) (BJN), Asst. Prof. Mech. Engrg., 206 State Hall, Univ. of Pittsburgh, Pittsburgh 13, Pa.
Beckwith, Ward ('42) (CMODGT), V.P., Exec. Engr., Thompson & Lichtner Co., Inc., 806 Park Sq. Bldg., Boston 16, Mass.
Becnel, P. Alfred, Jr. (J'46) (BEJ), 418 Angela Ave., Arabi 16, La.
Bedinger, Albert F. G. ('34; '45) (OMNLKY), Gp. Engr., Chem. Drafting Rm., Tenn. Eastman Corp., Oak Ridge; *for mail,* 1805 W. Clinch Ave., Knoxville 16, Tenn.
Beecher, Clarence Yale, Jr. (J'33) (MYA), Staff Engr., Roll Film Dept., Eastman Kodak Co., Kodak Park, Rochester 4; *for mail,* 25 Needham St., Rochester 13, N.Y.
Beede, Arnold Henry ('33; '45) (NBAYMH), Design & Devel. Engr., Electrolux Corp., Forest Ave., Old Greenwich; *for mail,* 232 Sound View Ave., Stamford, Conn.
Beede, E. Bennett (J'35) (DBJKZL), Mfrs. Agt., Stand. Conveyor Co., 755 Boylston St., Boston 16; *for mail,* R.F.D., Trapelo Rd., South Lincoln, Mass.
Beedle, Arthur K. ('25; '41), Mech. Engr., Asst. to Ch. Permanent Mold Engr., Aluminum Co. of Am., 2210 Harvard Ave., Cleveland 5; *for mail,* 3525 Toenley Rd., Shaker Heights 22, Ohio.
Beekley, Waldron C. ('16; '35), V.P., Whitlock Mfg. Co., Hartford; *for mail,* 35 Walker Lane, West Hartford, Conn.
Beekman, Henry M. ('17; '24; '32), Bedminster, N.J.
Beeler, Greig B. (J'46), 2112 W. Capitol, Houston 10, Tex.
Beensen, Charles ('22; '35) (DEBLOH), Ch. Engr., Maule Industries, 21st Ave., Miami, Fla.
Beers, Otis E. ('30) (MJCD), Pres., Gen. Mgr., Buescher Band Instrument Co., 225 E. Jackson Blvd., Elkhart, Ind.
Beers, Royce L. ('14; '30) (FKS), V.P., Charge Engrg., Detroit Stoker Co., 5-125 Gen. Motors Bldg., Detroit 2; *for mail,* 345 Arlington Rd., Birmingham, Mich.
Beese, Charles William ('38) (C), Dir., Tech. Ext. Div., Head, Gen. Engrg. Dept., Purdue Univ., Lafayette, Ind.
Beesley, Glyn Walter (J'42) (SHZO), Turbine Field Engr., Gen. Elec. Co., 1801 N. Lamar, Dallas 2, Tex.
Beeson, Frank McDonald ('28) (CS), Gen. Mgr., Beeson Bros. Engrg. Co., 1400 N. Spring St., Los Angeles 12, Calif.
Begeman, Myron Louis ('21; '35) (JLMCDW), Prof. Mech. Engrg., Univ. of Tex., Austin, Tex.
Begg, Robert Arthur (J'43) (NDBJSH), Mech. Draftsman, Dominion Bridge Co. Ltd.; *for mail,* 625—44th Ave., Lachine, Que., Can.
Beggs, Wm. E. ('29) (SLC), Owner, W. E. Beggs Co., 416 Bell St., Seattle 1, Wash.
Begley, Richard W. ('21), Supt. Engr., Booth Am. Shipping Corp., 17 Battery Pl., New York, N.Y.
Behar, Manoel Felix deMayo ('20; '27) (ZMBLKG), V.P., Editor, Instruments Publ. Co., 1117 Wolfendale St., Pittsburgh 12; Gen. Dir. Instrumentation Manual Project, 245 Melwood St.; *for mail,* The Wellington, Pittsburgh 13, Pa.
Behr, Francis J. ('14), 1013 Bella Vista Ave., Oakland 10, Calif.
Behr, Ralph Kottman ('40) (SFEKCJ), Dist. Rep., Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3, Ill.
Behrens, Henry F. ('22; '25; '35) (BFHS), Mech. Design Engr., Commonwealth & So. Corp., 212 Michigan Ave.; *for mail,* 761 Union St., Jackson, Mich.
Behrens, Karl Frederick (J'45) (MLYC), Maint. Engr., Shell Chem. Corp., Shell Bldg., Houston 1; *for mail,* 1209 S. Randall St., Pasadena, Tex.
Behringer, Charles David (J'33) (COM), Staff Engr., Albert Ramond & Associates, Rm. 6300, Chrysler Bldg., New York, N.Y.; *for mail,* 285 Johnson Ave., Teaneck, N.J.
Beich, William August (J'45) (MNLCDZ), Paul F. Beich Candy Co., Front & Lumber Sts., Bloomington, Ill.
Beichley, F. Wendell (J'43) (RACSZB), Cons. & Application Engr., Westinghouse Elec. Corp., 1 Montgomery St., San Francisco 4, Calif.
Beier, Kurt Alfred ('43) (EM), Ch. Engr., Automotive Div., Schwitzer-Cummins Co., 1125 Massachusetts Ave., Indianapolis 7; *for mail,* 7002 College Ave., Indianapolis, Ind.
Beier, Paul Herman ('45) (MDCAJB), Plant Engr., Austenal Labs., Inc., 224 E. 39th St., New York 16, N.Y.; *for mail,* 52 Palisade Rd., Elizabeth 3, N.J.
Beighley, Paul Allen, Jr. (J'41) (ACEMNS), Lt. Comdr., U.S.N., Washington, D.C.; *for mail,* 504 Happy Creek Rd., Front Royal, Va.
Beir, Richard Martin (J'40) (MBJ), Design, Engr., E. I. du Pont de Nemours & Co., River Rd., Buffalo, N.Y.
Beischer, George M. (J'39) (EFR), Lt.; 111 Lincoln Ave., Clifton, N.J.
Beisser, Stanley Allen (J'44) (BACEJL), Graduate Student, Box 882, Stamford Univ., Calif.
Beitler, Samuel R. ('23; '34) (HSFEZK), *Vice-President,* '45-'47; Prof. Hyd. Engrg., Ohio State Univ., Columbus 10, Ohio.
Beitzer, Vern Fredrick (J'40) (CDOJLM), Ch. Inspr., Arty. Branch, Army Serv. Forces, Milwaukee Regional Office, Chicago Ord. Dist., 322 E. Michigan St., Milwaukee 2; *for mail,* Apt. 105, 2327 W. Michigan St., Milwaukee 3, Wis.
Bejarano, Julio G. ('26; '42) (BKL), Shell Chem. Co., 100 Bush St., San Francisco, Calif.
Belaef, Nikolai N. ('38) (MZHLC), Control Engr., Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2; *for mail,* 68 Baker Ave., Westport, Conn.
Belair, Roy Albert (J'43) (LKSZ), Design Draftsman, Hercules Powder Co., Wilmington; *for mail,* 305 Wyoming Ave., Wilmington 278, Del.
Belcher, Wallace Edward, Jr. (J'41) (ZLA), Devel. Engr., Brown Instrument Co., Wayne & Roberts Aves., Philadelphia 44; *for mail,* 7424 Devon St., Philadelphia 19, Pa.
Beliaeff, Stephen B. ('24; '35) (CMY), Pres., Mgr., Wonder Products Corp., 323 Berry St., Brooklyn 11, N.Y.
Belilove, Saul (J'45) (ECSORA), Ch. Application Engr., Sales Engrg., Enterprise Eng. & Fdy. Co., 2901 Florida St., San Francisco, Calif.
Beline, Martin G. (J'40) (KOM), Mech. Engr., Htg. & Vent., Louis C. Kingscott & Associates, Archs. & Engrs., 208 Elm St., Kalamazoo; *for mail,* 839 W. Lovell St., Kalamazoo 45, Mich.
Beline, Walter E. ('30; '40) (KLC), Alco Products Div., Am. Loco. Co., 30 Church St., New York 8; *for mail,* 103 Beach Ave., Larchmont, N.Y.
Beling, John K. (J'41) (AB), Lt., U.S.N.R., Naval Aviator; *for mail,* La Roche Ave., Harrington Park, N.J.
Belinkin, Julius Alexander ('43) (EBMJ), Tech. Engr., 163—8th Ave., New York 11, N.Y.
Belitz, Walter B., Jr. (J'39) (CDKLMS), Maj., Corps of Engrs., U.S.A., McGuire Gen. Hospital, Richmond, Va.; *for mail,* 5100 Gwynn Oak Ave., Baltimore 7, Md.
Belk, Wilber C. (J'40), 3698 Van Buren St., Arlington, Calif.
Bell, Andrew Lane ('13; '19) (LBD), Prin. Design Engr., Solvay Process Co., Hopewell; *for mail,* 602 S. Boulevard, Petersburg, Va.
Bell, Clinton W., Sr. ('17; '35) (SBFKY), Asst. to Supt. of Generation, Pa. Power & Light Co., Cedar & Buttonwood Sts., Hazleton, Pa.
Bell, Dante Louis (J'41) (BYNMDA), Engr., Ord. Dept., U.S.A., Erie Proving Ground, Lacarne; *for mail,* 271 E. Tallmadge Ave., Akron 10, Ohio.
Bell, Frank B. ('18), Pres., Edgewater Steel Co., Box 249, Pittsburgh, Pa.
Bell, Frank Suydam (J'32), Maribell Farm, Perrysburg, Ohio.
Bell, Frederick John ('31) (CHS), Mgr., C. A. Parsons of Can. Ltd., Royal Bank Bldg., Toronto 1, Ont., Can.
Bell, Harold S. ('43) (EKP), 729 Bond Bldg., 14th St. & New York Ave., N.W., Washington 5, D.C.
Bell, Harold Sill, Jr. (J'44) (ABG), 49 Woodland Ave., Summit, N.J.

Bell, James Broughton ('45) (ECMRSN), Asst. Mech. Engr., War Assets Corp., 4095 St. Catherine St. W., Montreal, Que., Can.
Bell, James Burt ('36) (JMACDH), Sales Engr., Young & Vann Supply Co., 1729—1st Ave., Birmingham 2; *for mail,* 1601 Sterling Pl., Birmingham 5, Ala.
Bell, John Alex (J'37) (FHR), Major, Ord. Sec., Hdq., III Bomber Comd., MacDill Field, Tampa 8, Fla.
Bell, John Edward (J'42) (MNKOSW), Prod. Engr., Barber-Colman Co., Rockford, Ill.; *for mail,* 717 Elm St., Beloit, Wis.
Bell, John Louis (J'41) (BC), Asst. Mar. Engr., U.S. Maritime Comm., 348 Baronne St., New Orleans 12; *for mail,* 2808 Calhoun, New Orleans 15, La.
Bell, Kenneth Deane (J'41) (ACE), Sr. Test Engr., Wright Aero. Corp., Beckwith Ave., Paterson 3; *for mail,* 57 S. Mountain Ave., Montclair, N.J.
Bell, Melvin ('39; '45) (ACO), Lt., U.S.N.R., Research & Devel. Div., Bur. of Ord., Navy Dept., Washington, D.C.; *for mail,* 1223 Beacon St., East Chicago, Ind.
Bell, Paul Woodworth ('42) (SFKCEW), Maint. Sales Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 500 Marsh Rd., Charlotte, N.C.
Bell, Thomas E. ('26; '33; '35) (ZSLFE), Branch Mgr., Republic Flow Meters Co., 213 Red Rock Bldg., Atlanta 3, Ga.
Bell, Walter R. (J'26) (CEHKLS), V.P., Charge of Sales, Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.
Bell, Wilbur Carroll (J'42), Lt., Army Air Forces, 1452nd Base Unit, Air Trans. Comd., A.P.O. 462, Minneapolis, Minn.; *for mail,* 6 Pleasant Ave., South Portland, Me.
Bell, William Corbett (J'44) (ACE), Box 5018, Tucson, Ariz.
Bell, William Duis ('40) (NOSFAZ), 2359 Canterbury Rd., Columbus 8, Ohio.
Bell, William Kenneth (J'43) (MCDLJ), Imperial Brass Mfg. Co., 1200 W. Harrison St., Chicago 7; *for mail,* 6720 S. Jeffery Ave., Chicago 49, Ill.
Bellaimey, Henry Edward, Jr. (J'41) (HCJMBA), Gen. Mgr., Detroit Tube Products, 304 S. Junction; *for mail,* 1076 Hubbard St., Detroit 9, Mich.
Bellamy, Leon ('14) (BDOS), Propr., Bellamy Engr. Co., 302 Walnut St.; *for mail,* 46 Central Ave., Newtonville 60, Mass.
Bellan, Theodore Myles (J'44), Technician, 5th Grade, 2766th Engr. Base Photomapping Co., 29th Engr. Base Topo. Bn., A.P.O. 75, c/o Postmaster, San Francisco, Calif.
Bellanca, Giuseppe Mario ('30) (ACHMYW), Pres., Bellanca Aircraft Corp., New Castle, Del.
Bellegia, Faust (J'35) (LST), Capt., Ord. Dept., U.S.A., Hdq., Advance Sec. 2, A.P.O. 629, c/o Postmaster, New York, N.Y.
Bellin, Albert Irving (J'44) (BAK), Graduate Student, Sch. of Engrg., Harvard Univ., Pierce Hall, Cambridge 28, Mass.; *for mail,* 50 Reynolds Ave., Providence 5, R.I.
Bellinger, Lorentz Dobbs ('29; '45) (EJS), Comdr., U.S.N.R., U.S.S. *Palau,* CVE-122, F.P.O., New York, N.Y.
Bellows, John C. (J'41), Lt., U.S.A., Hdq. 7th Ord. Bn., A.P.O. 307, San Jose, Calif.
Belsheim, Robert Oscar (J'44) (BNJME), Mech. Engr., Naval Research Lab., Bellevue, Washington 20; *for mail,* 1868 Columbia Rd., N.W., Washington 9, D.C.
Belsley, Steven Eric (J'38) (ABK), Aero. Engr., Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field, Calif.
Beltle, William Otto, Jr. (J'43) (NBAH), Ch. Tool Engr., Krautter Weber Tool Co., 605 Broad St., Newark 2; *for mail,* 1076 Stuyvesant Ave., Irvington 11, N.J.
Belton, Joseph F., Jr. (J'41), 2nd Lt., Army Air Forces, B.F.S., Shaw Field, Sumter, S.C.
Beltran, Edward Vincent ('37; '44) (KLC), Partner, Colley & Beltran, 225 Lafayette St., New York, N.Y.
Beltz, Fred Wallace, Jr. (J'44) (BKS), 2412 Buhl Bldg., Detroit 26, Mich.
Belz, Herman M. (J'40), 102 Meigs St., Clarksburg, W. Va.
Belz, Paul Dewey (J'43), 1st Lt., Coast Arty. Corps, A.U.S., Antiaircraft Sec., Hdq., 8th Army, A.P.O. 343, c/o Postmaster, San Francisco, Calif.; *for mail,* 1010 N. Tuckahoe St., Falls Church, Va.
Belz, Robt. A. (J'39), N. Augusta St., Staunton, Va.
Bemis, Walter S. ('21; '32) (CEF), Pres., Bemis Co., 1175 E. 63rd St., Chicago, Ill.
Bencze, Stephen (J'38) (ESBKNO), Asst. Div. Engr., Turbine Div., Elliott Co., N. 4th St.; *for mail,* 303 N. 3rd St., Jeannette, Pa.
Benda, Harry R. ('14; '31) (BNLWHC), V.P., Ch. Engr., P. Pryibil Mch. Co., Inc., 42-11—9th St., Long Island City 1, N.Y.; *for mail,* 250 Clark Terrace, Cliffside Park 1, N.J.
Bendelius, Albert (J'43) (LCD), Secy., Treas., Pfister Chem. Works, Ridgefield, N.J.
Bender, Eugene W., Jr. ('45) (BCDJMN), Dist. Engr., Am. Mutual Liability Ins. Co., 772 Ledger Bldg., Philadelphia, Pa.
Bender, René Jose (J'37) (FERSA), Asst. to Ch. Cons. Engr., Sinclair Refining Co., 630—5th Ave., New York 20, N.Y.
Bendersky, David (J'43) (AEB), Pvt., Co. B, 105th Bn., 76th Regt., Inf. Replacement Trng. Center, Camp Robinson, Ark.
Bendixen, Alf Hugo ('44) (NBESO), Staff Engr., Garmey, Bodin & Johansen, Inc., 32 Broadway, New York 4; *for mail,* 675—86th St., Brooklyn 9, N.Y.
Benedetti, George Gunnar ('30; '35) (BDKSHY), Spec. Staff Engr., N.Y. Cent. R.R. Co., Rm. 1842, 70 E. 45th St., New York 17, N.Y.; *for mail,* 111 Larch Ave., Bogota, N.J.
Benedick, Fred ('44) (SHKE), Mech. Engr., Puget Sound Naval Shipyard; *for mail,* 1412 Burwell Ave., Bremerton, Wash.
Benedict, Coleridge H., Jr. (J'40), Hdq. & Hdq. Co., 303rd Sig. Opera. Bn., Ft. Ord, Calif.; *for mail,* 56 Baker Ave., Dover, N.J.
Benedict, Loyal Clark ('19; '26; '35) (CDM), Prod. Mgr., Robot Appliances, Inc., 943 S. High St., Akron 11; *for mail,* 123 E. Prospect St., Wadsworth, Ohio.
Benedict, Walter E. ('28; '34; '35) (MLYDZ), Ch. Research Engr., Sloane-Blabon Corp., Hutchinson's Mills, Trenton, N.J.
Benedictus, Andrew Roger ('42; '45) (MCRSA), Rm. 1539, 55 W. 42nd St., New York 18, N.Y.
Benes, Gaspar Peter ('45) (BM), Ch. Tool Designer & Mech. Engr., Addressograph-Multigraph Corp., 1200 Babbitt Rd., Cleveland 17; *for mail,* 10541 Elk Ave., Cleveland, Ohio.
Benes, Ralph Joseph, Jr. (J'44), 90 Clover Ave., Floral Park, N.Y.
Benét, Laurence Vincent ('92), Retired; 2021 Connecticut Ave., Washington 8, D.C.
Benfer, Maurice Frink (J'27) (OMCJDG), Consultant, Indus. Engr., 3210 Cross St., Madison 5, Wis.
Benford, Robert L. (J'40) (BJS), Mem. Gear Engrg. Dept., Gen. Elec. Co., River Works, West Lynn; *for mail,* 265 Pleasant St., Marblehead, Mass.
Benger, William Frederick Alexander ('44), Ch. Mech. Engr., Canadian Pacific Ry. Co., Windsor St., Montreal 3, Que., Can.
Bengle, Charles Victor (J'38) (KZSBL), Sales Engr., Worthington Pump & Mchy. Corp., 8th & Race Sts., Cincinnati 1, Ohio.
Benischek, Howard William (J'37) (AMEFLC), 3962 Mt. Vernon, Ft. Worth 3, Tex.
Benjamin, Julien Philip, Jr. (J'38) (ABD), Lt., Conversion Officer, Water Div., Trans. Corps, U.S.A., N.Y. Port of Embarkation, Brooklyn; *for mail,* 6702 Ridge Blvd., Brooklyn 20, N.Y.
Benjamin, Max William ('31; '37) (BEFHKS), Steam Power Engr., Engrg. Div., Detroit Edison Co., 2000—2nd St., Detroit 26; *for mail,* 621 S. Denwood Dr., Dearborn, Mich.
Benjamin, Ray Neel ('25) (SHFEBK), Mech. Engr., Georgia Power Co., 75 Marietta St., N.W., Atlanta 3; *for mail,* 74—17th St., N.E., Atlanta, Ga.
Benjes, Edward McDevitt (J'34), 5005 Ross Rd., Baltimore 14, Md.
Benkert, John Edwin (J'43) (MALJCW), Pvt., Army Air Forces, Sqd. O, 3706th Base Unit, Sheppard Field, Tex.; *for mail,* 515 Winkler Ave., Louisville 8, Ky.
Benneche, Christian Price ('43) (OS), Engr., J. K. Blum, 205 E. 42nd St., New York 17, N.Y.
Benner, Louis Howard (J'41) (KSDEA), Field Engr., Am. Blower Corp., 611 Olive St., St. Louis 1, Mo.
Bennett, Arthur F. ('21; '35) (BDHJMN), Mch. Designer, Brown & Sharpe Mfg. Co., Promenade, Providence; *for mail,* 360 Washington Rd., West Barrington, R.I.
Bennett, Arthur L. (J'40) (FKS), Lt., U.S.N.R., 2706 Campbell Ave., Schenectady 6, N.Y.
Bennett, Charles Wilbur ('04) (CJ), Retired; 119 Hoodridge Dr., Mt. Lebanon, Pittsburgh 16, Pa.
Bennett, Clinton Wendell ('22; '35), Partner, Cooley & Marvin, 140 Federal St., Boston 10, Mass.
Bennett, Daniel Arthur ('30; '35; '38) (CNE), Lt. Comdr., U.S.N.R., Pat. Lawyer, Office of Research & Inventions, Navy Dept., Washington, D.C.; *for mail,* 3434 S. Wakefield St., Arlington, Va.
Bennett, Daniel Paul (J'44) (CMS), Pvt., 1st Class, Regt. Hdq., 16th Inf., A.P.O. 1, c/o Postmaster, New York, N.Y.
Bennett, Douglas Warren ('44) (BJM), Sr. Naval Arch., Norfolk Naval Shipyard, Portsmouth; *for mail,* 433 Wells Pkwy., Norfolk 3, Va.
Bennett, Frank S. ('22) (ABFKRS), Mech. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Bennett, George Lewis ('05) (OMC), Long Mountain, New Milford, Conn.
Bennett, George Wyckoff (J'42) (MBC), Prod. Engr., East. Air Devices, 585 Dean St., Brooklyn; *for mail,* 29 Clarkson Ave., Brooklyn 26, N.Y.
Bennett, Harold M. ('45) (BKM), Mech. Engr., E. E. Ashley, 10 E. 40th St., New York; *for mail,* 574—81st St., Brooklyn 9, N.Y.
Bennett, Harry A. ('24), V.P., Charge Mfg., Haughton Elev. Co., 671 Spencer St.; *for mail,* 2715 Sagamore Rd., Toledo, Ohio.
Bennett, Henry George ('24; '35) (CFHKSZ), Engr., Commercial Relations, N.Y. Steam Corp., 130 E. 15th St., New York; *for mail,* 40 Earley St., City Island 64, N.Y.
Bennett, Hubert Duke ('44), Pres., Toledo Scale Co., Defiance Mch. Works, Toledo 12; *for mail,* 2247 Collingwood Blvd., Toledo 10, Ohio.
Bennett, James ('41) (BHJKLT), Ch. Plant Engr., Am. Viscose Corp.; *for mail,* Box 1043, Front Royal, Va.
Bennett, James William (J'44) (LSZ), Mech. Engr., Energy Control Co., 866 Broad St., Newark; *for mail,* Box 524, R.F.D. 2, Westwood, N.J.
Bennett, Joseph S., 3rd ('24; '26) (FSC), V.P., Am. Engrg. Co., Philadelphia 25, Pa.
Bennett, Kenneth Stephen (J'42) (MCDN), Engr., Gen. Elec. Co., Schenectady; *for mail,* 2706 Campbell Ave., Schenectady 6, N.Y.
Bennett, Leland Wilson ('18; '35) (SOHAC), Valuation Engr., Ebasco Services, Inc., 2 Rector St., New York 6; *for mail,* 420 W. 24th St., New York 11, N.Y.
Bennett, Norman H. (J'40) (BHSMC), Goldendale, Wash.
Bennett, Raymond Franklin ('39) (CMSOLK), Gen. Supt., Ecusta Paper Corp., Box 200, Pisgah Forest, N.C.
Bennett, Vincent W. (J'40) (EBC), Asst. Gen. Supt., Gen. Ship & Eng. Works, 336 Border St., East Boston; *for mail,* 29 Banks Rd., Swampscott, Mass.
Bennett, Wm. C. M. (J'37), Observer, Carnegie-Ill. Steel Corp., Munhall; *for mail,* 6314 Marchand St., East Liberty, Pa.
Bennett, William Francis (J'41) (FKOCE), Jr. Engr., Atlanta Gas & Light Co., Box 4569, Atlanta 2, Ga.
Bennett, William H. ('45), Supt., Gautier Dept., Bethlehem Steel Co.; *for mail,* Box 533, Johnstown, Pa.
Bennett, William H. K. ('20) (DH), Pres., W. H. K. Bennett Co., 100 W. Monroe St., Chicago, Ill.
Benning, Victor Leopold ('22; '32; '35), Charge Engrg., Electricity Supply Comm., Salt River Power Sta., Dock Rd.; *for mail,* "Naby," Bennington Rd., Capetown, South Africa.
Bennon, Meyer (J'43) (ZLJA), Radio Engr., Naval Air Exper. Sta., Naval Shipyard, Philadelphia; *for mail,* 542 Tasker St., Philadelphia 48, Pa.
Benns, Jack J. ('45) (DOR), Ch., Arty. Branch, Army Serv. Forces, Chicago Ord. Dist., 38 S. Dearborn St., Chicago 3; *for mail,* 5423 Leland Ave., Chicago, Ill.
Bennum, George C. (J'43) (BCRMN), 190 Searlwyn Rd., Syracuse 5, N.Y.
Benoit, A. W. ('21) (T), Assoc., Chas. T. Main, Inc., 201 Devonshire St., Boston, Mass.
Benoit, Lester William ('25; '45) (SHEJ), Gen. Secy., Mfrs. Standardization Soc. of the Valve & Fittings Industry, 420 Lexington Ave., New York 17, N.Y.
Benscoter, Daniel B. ('28; '35) (EFS), Mun. Engr., Dayton Power & Light Co., Gas & Elec. Bldg., 25 N. Main St., Dayton 1, Ohio.
Bensen, Igor B. (J'40) (BACJ), Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Bensinger, Samuel M. ('34; '42) (EBCRAT), Plant Engr., Arms Textile Mfg. Co., Manchester, N.H.
Benson, Allen Arthur ('43) (SBFKJ), Power Plant Design Engr., Indianapolis Power & Light Co., 17 N. Meridian St., Indianapolis 6; *for mail,* 3410 Brookside Pkwy. N. Dr., Indianapolis 1, Ind.
Benson, Arthur Edward ('26; '34; '35) (YCLM), Project Engr., Tire Devel., U.S. Rubber Co., 6600 E. Jefferson Ave., Detroit 32; *for mail,* 754 Ashland Ave., Detroit 15, Mich.
Benson, Carl Ivar, Jr. (J'45) (SKEBA), Engr., East. Refractories Co., Inc., 70 Pearl St., Brookline 46; *for mail,* 85 Judson Rd., Weymouth 88, Mass.
Benson, Carl N. ('18; '26; '35) (HTW), Mech. Engr., SKF Industries, Inc., Philadelphia 34; *for mail,* 427 Newbold Rd., Jenkintown, Pa.
Benson, Chenery F. ('19; '22) (TSC), Mgr., Chicago Curled Hair & Gen. Felt Products Co., 2301 S. Paulina St., Chicago 8, Ill.
Benson, Frank Robert ('45) (EHNSW), Ch. Engr., Newport News Shipbldg. & Drydock Co., Newport News, Va.
Benson, George Albert ('42) (BMWC), Mem. Tech. Staff, Bell Tel. Labs., 463 West St., New York 14, N.Y.
Benson, John Goffe (J'36) (CSKLZ), Cost Analyst, Linde Air Products Co., 30 E. 42nd St., New York 17; *for mail,* Edgepark Rd., White Plains, N. Y.
Benson, Paul Chadbourne (J'39) (CMAZY), Asst. Prod. Mgr., Ericsson Screw Mch. Products Co., Inc., 25 Lafayette St., Brooklyn 1; *for mail,* 5 Old Mamaroneck Rd., White Plains, N.Y.
Benson, Philip A., Jr. (J'41) (CJM), U.S.N.; 9 DeKalb Ave., Brooklyn, N.Y.
Benson, Robert Elliott (J'38) (MCOS), Indus. Analyst, Securities Investment Dept., Equitable Life Assurance Soc. of the U.S., 393—7th Ave., New York 1; *for mail,* 280 Prospect Ave., Sea Cliff, L.I., N.Y.
Benson, Stuart W., Jr. (J'40) (ABY), Lt. Col., 1221 Boundary St., Aliquippa, Pa.
Bent, Wendell Eugene (J'43) (CKS), Lt. (j.g.), U.S.N.R., Southdown Ave., Huntington, L.I., N.Y.
Bentey, Thomas J. ('39; '46) (MJD), Design Engr., Pollak Mfg. Co., 541 Devon St., Arlington; *for mail,* 70 Lindsley Pl., East Orange, N.J.
Bentivegna, Anthony Francis (J'45) (SKA), Pvt., Sqd. SB-6, Box 25, Keesler Field, Miss.

Bentley, David Merrell (J'40) (ACD), Engr., Reports & Appraisals, J. G. White Engrg. Corp., 80 Broad St., New York; *for mail*, 22 Dongan St., Staten Island, N.Y.
Bentley, Geo. L. (A'21) (HSK), Sales & Serv. Engr., Ingersoll-Rand Co., 1700—3rd Ave. S., Birmingham 3, Ala.
Bentley, Harold ('30) (DEFK), Gen. Mgr., Keymer Bagshawe & Co., Ltd., 22 Strand Rd.; *for mail*, Box 399, Calcutta, India.
Bentley, Oliver D. H. ('10) (CJS), Mgr., Engr., Turbine Dept., B. F. Sturtevant Co., Damon St., Hyde Park, Boston, Mass.
Bently, Julius Gordon ('24; '25; '35) (FSRDKO), Mech. Engr., Fuel Serv., Army Serv. Forces, Hdq. 6th Serv. Comd., 20 N. Wacker Dr., Chicago 6; *for mail*, 7748 East End Ave., Chicago 49, Ill.
Benton, Eugene Dewey ('42) (FRSEZC), Fuel Engr., Louisville & Nashville R.R. Co., 908 W. Broadway, Louisville 3; *for mail*, 1886 Rutherford Ave., Louisville 5, Ky.
Benton, Louis Arnett ('45) (CDLJN), Supervisory Engr., Barrett Div., Allied Chem. & Dye Corp., Frankford, Philadelphia 37; *for mail*, Route 1, Newton, Bucks County, Pa.
Bentson, Harold Jacob ('37) (FKLS), Power Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Benzien, Fritz ('30; '36) (BCL), Maint. Engr., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City; *for mail*, 339 Cumberland Rd., South Orange, N.J.
Benzon, George H., Jr. ('18) (BCM), Prod. Mgr., Lucas Mch. Tool Co., 523 E. 99th St.; *for mail*, 13805 Shaker Blvd., Cleveland, Ohio.
Beran, Charles Francis ('27) (CTLYDF), V.P., Charge of Prod., Celanese Corp. of Am., 180 Madison Ave., New York 16, N.Y.
Berard, Samuel John ('19) (NGB), Prof. Emeritus, Engrg. Drawing, Brown Univ., Providence 12; *for mail*, 51 University Ave., Providence 6, R.I.
Berberich, Charles Edward (J'37) (SFCZK), Engr., Potomac Elec. Power Co., 10th & E Sts., N.W., Washington 4; *for mail*, 1728 Lamont St., N.W., Washington 10, D.C.
Bercaw, Corliss Arthur (J'39) (RCEMF), Dist. Sales Mgr., Electro-Motive Div., Gen. Motors Corp., 135 S. La Salle St., Chicago 3, Ill.
Berdahl, Edgar Oliver (J'45) (ABZOCN), Vibration Measurement & Research, David Taylor Model Basin, Navy Dept., Washington 7, D.C.; *for mail*, 20 Froude Circle, Bethesda, Md.
Berdan, Frank, Jr. (J'39) (C), Asst. to Mgr., Naval Aircraft Factory, Naval Air Matl. Center, Naval Base Sta., Philadelphia 12, Pa.; *for mail*, West Orchard St., Allendale, N.J.
Beretta, John Ward ('32) (ABC), Pres., J. W. Beretta Engrs., Inc., 1205 Natl. Bank Bldg., San Antonio 5; *for mail*, 404 W. French Pl., San Antonio 1, Tex.
Berg, Henning J. ('17; '21; '35) (HS), Mech. Engr., Stand. Oil Co. of Calif., 225 Bush St.; *for mail*, 2335 Laguna St., San Francisco, Calif.
Berg, Lothar Edward (J'44) (HKL), c/o Minneapolis-Honeywell Regulator Co., 1305 Capital St., Houston, Tex.
Bergdolt, George James, Jr. (J'43) (AEMCJ), 122 Shippen St., Weehawken, N.J.
Bergdolt, Vollmar Edgar (J'39) (J), 104 N. Salisbury St., West Lafayette, Ind.
Bergen, Martin John ('37; '45) (ABF), Ch. Draftsman, E. I. du Pont de Nemours & Co., Wilmington 98; *for mail*, Elkton Blvd., R.R. 1, Newark, Del.
Berger, Alfred (J'41), 1828 Andrews Ave., Bronx 53, N.Y.
Berger, Frank Alfred ('09) (BEFKS), Prof. Mech. Engrg., Washington Univ., St. Louis 5, Mo.
Berger, George G. ('42; '43) (CBGAN), Gen. Mgr., Duroyd Gasket Mfg. Co., 1828 Amsterdam Ave., New York 31, N.Y.
Berger, Joseph W. ('13) (SKBFNH), Mech. Engr., 579 Wolcott Ave., Beacon, N.Y.
Berger, Julius George ('14; '24) (OSFETK), Cons. Engr., Mech. Engr., Firm of J. G. Berger, 24 Commerce St., Newark 2, N.J.
Berger, Knute ('26) (ABHJN), Pres., Gen. Mgr., Berger Engrg. Works, Inc., 3236—16th Ave., S.W., Seattle 4; *for mail*, 3432 Mt. Baker Blvd., Seattle 44, Wash.
Berger, Peter (J'43) (SZKEAY), Aerodynamicist, Ranger-Lark Co.; *for mail*, 9102—169th St., Jamaica 3, L.I., N.Y.
Berggren, Karl G. ('27; '35), Engr., Thos. A. Edison, Inc.; *for mail*, 62 Valley Way, West Orange, N.J.
Bergh, Paul Sigvard (J'42) (ABH), Radio Technician, 20-A-4, Rm. 612 Yorktown, Naval Trng. Sch., Del Monte, Calif.
Bergin, Robert Francis (J'38) (FKS), Serv. Engr., Hall Labs. Inc., 2512 Book Bldg., Detroit 26; *for mail*, 9599 Cardwell, Garden City, Mich.
Berg-Johnsen, Jorn, Jr. (J'41) (ACDJM), Requisition Engr., Carbon Products Div., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 1567 Union St., Schenectady 8, N.Y.
Bergland, Wm. S. ('18; '35) (JLW), Project Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.
Bergman, Donald J. ('24; '35) (LZKBH), Ch. Engr., Universal Oil Products Co., 310 S. Michigan Ave., Chicago, Ill.
Bergman, Elmer Otto ('43) (BJKHO), Cons. Engr., C. F. Braun & Co., 1000 S. Fremont Ave., Alhambra, Calif.
Bergmann, Adolph A. ('23; '29) (SKFJE), Mech. Engr., Pub. Utility Engrg. & Serv. Corp., 231 S. La Salle St., Chicago 4, Ill.
Bergner, Frederick A., 3rd (J'41), 7427 Rising Sun Ave., Philadelphia, Pa.
Bergsland, Charles Hans (J'39) (NHYB), Mech. Engr., Lake Erie Engrg. Co., Box 68; *for mail*, 25 Eiseman, Kenmore 17, N.Y.
Bergstrom, Albert L. ('37) (JMR), V.P., Charge Engrg., Timken Roller Bearing Co., 1835 Dueber Ave., Canton 6, Ohio.
Bergstrom, Paul Hugo (J'38), Reservoir St., Holyoke, Mass.
Bergstrom, Ralph W. (J'41), Capt., Army Air Forces, Apt. 3, 1950 Hinsachee St., San Antonio, Tex.
Berkey, Donald Clayton (J'43) (BJKNS), Turbine-Generator Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady 1; *for mail*, 201 Jackson Ave., Schenectady 4, N.Y.
Berkley, Henry Walter ('29) (SBH), Field Erection Supt., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.; *for mail*, c/o Mrs. J. E. Staggs, 52 Yukon St., San Francisco 14, Calif.
Berkley, William Ernest ('21; '26; '35) (BS), Dist. Turbine Engr., Gen. Elec. Co., 1801 N. Lamar St., Dallas 2, Tex.
Berkness, Irving Russell ('44) (SOFCDE), Mech. Engr., Designer, Slaughter, Saville & Blackburn Inc., Law Bldg.; *for mail*, 1853 W. Grace St., Richmond, Va.
Berlau, Charles Jacob (J'46), 232 E. Front St., Berwick, Pa.
Berman, Benjamin P. (J'38) (ABM), Mar. Engr., Naval Shipyard; *for mail*, 4417 Baltimore Ave., Philadelphia 3, Pa.
Berman, Robert (J'43), 204 N. Beechwood Ave., Catonsville, Md.
Bermeo-Cevallos, Carlos H. (J'45), Grand Hotel, Guayaquil, Ecuador, S.A.
Bermingham, John Richard (J'45) (AYCNHF), Mgr., Indus. Div., Am.-Ind. Inc., 420 Madison Ave.; *for mail*, 137 E. 28th St., New York 17, N.Y.
Bernard, Harold B. ('15; '21; '28), V.P., Sinclair-Prairie Oil Co., Sinclair Bldg.; *for mail*, 1500 S. Frisco Ave., Tulsa, Okla.
Bernardo, Everett (J'42) (AFJKS), Mech. Engr., Aircraft Eng. Research Lab., Natl. Advisory Comm. for Aero., Cleveland Airport; *for mail*, 3576 W. 147th St., Cleveland 11, Ohio.
Bernat, Richard Raymond (J'43) (BHCD), Technician, 4th Grade, Corps of Engrs., Spec. Engrg. Det., Oak Ridge, Tenn.; *for mail*, 24 Mercer St., Jersey City 2, N.J.
Bernauer, Raymond O. ('44), Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7; *for mail*, 1236 South 36th St., Milwaukee 4, Wis.
Berndt, Paul (J'45) (AEL), 5450 S. Blackstone Ave., Chicago 15, Ill.
Berner, Gerard Herman (J'43) (ABJ), Sr. Stress Analyst, Glenn L. Martin Co., Baltimore 3; *for mail*, 5401 Catalpha Rd., Baltimore 14, Md.
Berner, John ('43) (NMJCD), Shop Supt., Associated Wirecloth Products, Inc., 262-72—44th St., Brooklyn; *for mail*, 1360 Ogden Ave., Bronx 52, N.Y.
Bernhard, Richard (J'01) (BHJDS), Ch. Engr., Traylor Engrg. & Mfg. Co.; *for mail*, 125 S. West St., Allentown, Pa.
Bernhard, Richard Lowry (J'34) (BEKZ), Engr., Turbo Blower Dept., Ingersoll-Rand Co., Phillipsburg, N.J.; *for mail*, 77 N. 2nd St., Easton, Pa.
Bernhardt, Charles H. ('45) (BCJ), Plant Mgr., Dir. of Engrg., Spar-Tan Engrg. Co., 1504 E. Washington Blvd., Los Angeles 21, Calif.
Bernhardt, George Keith (J'44), 1148 Boulevard, West Hartford 7, Conn.
Bernhardt, Lee Frank (J'43) (JNMBLA), Owner, Design, 2707-09 S. Calhoun St., Ft. Wayne 6, Ind.
Berninger, Robert D. ('32; '35) (FKS), Asst. Supt., Stanton Div., Scranton Elec. Co., Box 381, Pittston, Pa.
Bernitt, Elmer W. (J'33) (CMDOFS), Plant Engr., Nash Motors Div., Nash-Kelvinator Corp., 5626—25th Ave., Kenosha, Wis.
Bernstein, Herbert J. (J'37) (CLS), Project Engr., York Corp., 1238 N. 44th St., Philadelphia 4; *for mail*, 2260 Bryn Mawr Ave., Philadelphia 31, Pa.
Bernstein, Ralph Henry (J'45) (HKS), 2 Crow Lane, Glen Cove, L.I., N.Y.
Berntsen, Carl Bernard (J'45), Field Serv. Engr., Ford Instrument Co., Inc., Rawson St. & Nelson Ave., Long Island City 1; *for mail*, 435 Delafield Ave., Staten Island 10, N.Y.
Berolzheimer, Henry ('18; '25; '35), Secy., Treas., Eagle Pencil Co., 710 E. 14th St., New York, N.Y.
Berrian, Kenneth J. (J'35), Croll Reynolds Co., 17 John St., New York, N.Y.; *for mail*, 36 Sherman Pl., Jersey City, N.J.
Berry, Bernard C. ('39) (SHFD), Sales Mgr., Allen-Sherman-Hoff Co., 271 Madison Ave., New York 16, N. Y.
Berry, C. Harold ('19; '21) (FKS), Prof. Mech. Engrg., Harvard Univ., Pierce Hall, Cambridge 38, Mass.
Berry, Earle William ('41) (EOSHC), Cons. Engr., 603 Baltimore St., El Paso, Tex.; *for mail*, 184 A St., Hayward, Calif.
Berry, Edward Harold, Jr. (J'37) (BJC), Plant Metallurgist, Dodge Steel Co., 6501 State Rd., Philadelphia 35, Pa.
Berry, Edward John ('45) (NMJT), Partner, Supvr., Berry Engrg. Serv., 159 Georgia Ave., Providence 5; *for mail*, 84 Massasoit Ave., Cranston 5, R.I.
Berry, Ethan Allen ('45) (AHCJNZ), Mgr., Plant 3, Johnson Tool & Eng. Co., 128 Tulsa Lane, Dayton 3, Ohio.
Berry, Francis Eugene, Jr. ('42) (CKM), Div. Mgr., Research Inst. of Am., Inc., 826 Pk. Sq. Bldg., Boston 15; *for mail*, 11 Chapel Pl., Wellesley Hills 82, Mass.
Berry, Francis Rigdon ('26), Asst. Ch. Engr., Am. Water Works & Elec. Co., 50 Broad St., New York 4, N.Y.
Berry, George Wolf (J'45) (TYCDLO), Engr., Product Devel., Duro-Persian Mfg. Co., 9 Central Ave., East Newark; *for mail*, 472 Mayhew Court, South Orange, N.J.
Berry, James F. ('40) (OM), 373 Frederick St., San Francisco 17, Calif.
Berry, William Randolph (J'40) (BNSM), Design Engr., Mech. Design Calculations, South Philadelphia Works, Westinghouse Elec. Corp., Lester, Pa.; *for mail*, 1389 Kenwood Ave., Camden, N.J.
Berryman, Harold O. ('21; '44), V.P., Natl. Can Corp., 110 E. 42nd St. New York 17, N.Y.; *for mail*, 2513 Hermosa Ave., Baltimore 14, Md.
Berryman, Richard Henry (J'39) (CEKOZ), Ch. Engr., Thermocraft Co., 1805 E. McNichols Rd.; *for mail*, 264 Worcester Pl., Detroit 3, Mich.
Bertagni, Americo Joseph (J'43), 145 Louisiana Ave., Brooklyn, N.Y.
Bertelsen, Albert Norton ('44) (CES), 84 Garrison Rd., Hingham, Mass.
Bertelsen, Christian ('29; '35), Dept. of Water Supply, Gas & Elec., City of N.Y., Altantic Ave. & Logan St., Brooklyn 8; *for mail*, 262—78th St., Brooklyn 9, N.Y.
Bertram, Edward Austen ('44) (KLSJ), Mgr., Heat Transfer, Engrg. & Sales, Indus. Div., Natl. Radiator Co., 60 E. 42nd St., New York 17; *for mail*, 170 E. Hartsdale Ave., Hartsdale, N.Y.
Bertram, H. Graham ('16; '21) (CMJBR), Pres., Gen. Mgr., John Bertram & Sons Co., Ltd., Dundas, Ont., Can.
Bertrand, Louis (J'39) (DELZ), Engrg. Dept. Consultant, E. I. du Pont de Nemours & Co., Wilmington; *for mail*, 11 Wier Ave., Wilmington 253, Del.
Berwick, John Douglas (J'43) (AESN), 57 Robbin St., Avon, Mass.
Beshers, Hugh Monahan ('32) (LCDM), Exec. Engr., E. R. Squibb & Sons, New Brunswick, N.J.
Besio, Charles Arthur (J'44) (SEFA), Asst. Prof. Mech. Engrg., Sch. of Engrg., So. Methodist Univ., Dallas 5, Tex.
Beskin, Leon ('44) (ABC), Design Specialist, Aerodynamics Gp., Consltd. Vultee Aircraft Corp., Vultee Field, Downey; *for mail*, 2850 Russel St., Whittier, Calif.
Besse, Gilbert L. (J'37), 6 Alden St., Plymouth, Mass.
Bessemer, Milton Emanuel (J'26) (SKCH), Mar. Engr., Supvr. Shipbldg., U.S.N., Fed. Shipbldg. & Dry Dock Co., Kearny; *for mail*, 148 Kensington Ave., Jersey City, N.J.
Bessent, Carl Francis (J'43) (ACJ), 1119 H St., Baltimore 19, Md.
Bessent, Thomas Anthony (J'43) (CMSKD), Lt., U.S.N.R., Todd-Pacific Shipyards; *for mail*, Apt. 407, 1119 Boren Ave., Seattle, Wash.
Bessio, Oscar (J'33) (MRS), Mar. Engr., Bur. of Ships, Navy Dept., Constitution Ave., Washington 25; *for mail*, 434 Mellon St., S.E., Washington 20, D.C.
Bessom, Richard Hartland ('44) (SKHEBN), Steam Supt., Westinghouse Elec. Corp., 10 High St., Boston 10; *for mail*, 16 Barnstable St., Swampscott, Mass.
Best, Carl E. (J'40), Engr., Cincinnati Milling Mch. Co., Marburg & South St.; *for mail*, 1468 Burney Lane, Mt. Washington Sta., Cincinnati, Ohio.
Best, Jesse L. ('14; '21) (SLCK), Ch. Oper. Engr., Anheuser-Busch, Inc., 721 Pestalozzi St., St. Louis 18, Mo.
Best, Luther Samuel ('44) (BCHMNS), Ch. Inspr., Supvr. Shipbldg., Navy Dept., Box 8387, Seattle 14, Wash.
Best, Robert D. (J'27) (BEAN), Test Engr., Fredric Flader, Inc., 775 Main St., Buffalo 3; *for mail*, 97 Northledge Dr., Snyder, N.Y.
Bester, Leonard R. ('23) (CEKMRS), Sr. Engr., Bur. of Ships, Navy Dept., Washington; *for mail*, 1417 Sheridan St., N.W., Washington 11, D.C.
Beth, Walter Frederick (J'36) (NMC), Asst. Mech. Engr., Norton Co., New Bond St., Worcester 6; *for mail*, 25 Dean St., Worcester 2, Mass.
Better, Paul (J'46), 24 Evans Rd., Peabody, Mass.
Bettis, John Randolph (J'40) (HSE), Student Engr., Commrs. of Pub. Works, 14 George St.; *for mail*, Box 123, Route 2, Naval Shipyard, Charleston, S.C.
Bettison, A. Morse (J'44) (ABE), Ohio State Univ.; *for mail*, 47 E. Torrence Rd., Columbus 2, Ohio.

Bettman, Robert ('23; '34; '35) (CMSJ), Combustion Engrg. Co., Inc. 200 Madison Ave., New York 16, N.Y.; *for mail*, 10 Hester St., Little Ferry, N.J.
Betts, George Elmer, Jr. (J'37) (AMYWEJ), Ord. Engr., Bur. of Ord., Navy Dept., Navy Bldg., Washington, D.C.; *for mail*, Apt. 101, 2711 Lee Blvd., Arlington, Va.
Betts, Walter L. ('16; '22; '35) (MNCRZA), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14; *for mail*, 167 Maple St., Brooklyn 25, N.Y.
Betty, Bernard Breast (J'30) (BMJ), D-1, Ambassador Apt., 25th Ave. S., Nashville 5, Tenn.
Betz, L. Drew (A'30) (SKCOG), Mem. Firm, Gen. Mgr., W. H. & L. D. Betz, Gillingham & Worth Sts., Philadelphia 24, Pa.
Betzler, Henry William ('34; '35) (ZSBAJY), Devel. Engr., Fed. Tel. & Radio Corp., 1000 Passaic Ave., East Newark; *for mail*, 127 Rutherford Blvd., Passaic, N.J.
Beutel, Albert Phillip ('29) (CEFKSY), Gen. Mgr., Tex. Div., Dow Chem. Co., Freeport; *for mail*, Box N, Lake Jackson, Tex.
Beuter, Arthur John ('46) (CER), Sales Engr., Gen. Steel Castings Corp., Eddystone; *for mail*, 4701 Pine St., Philadelphia 43, Pa.
Bevans, Herbert Miller (J'43) (ABCEF), 95 Quincy St., Quincy 69, Mass.
Bevere, John Paul (J'42) (BHL), 84 Satterthwaite Ave., Nutley, N.J.
Beversdorf, Roy C. (J'38), Jr. Engr., Shell Oil Co., Inc., East Chicago, Ind.; *for mail*, 1512 E. 76th Pl., Chicago, Ill.
Bevin, Sydney ('09; '21) (CL), Ch. Engr., Fiske Bros. Refining Co., Box 1526 C.S., Toledo, Ohio; *for mail*, Box 13, Northport, N.Y.
Beyer, Ben. W., Jr. ('21; '39) (DCMR), Beyer Mchy. Co., 1709 Book Bldg., Detroit 26, Mich.
Beyer, Frederick C. (J'44) (JNBOCM), 6640 S. California Ave., Chicago 29, Ill.
Beyer, John Robert (J'45) (CDO), Supvr., Methods Engrg., Bendix Radio Div., Bendix Aviation Corp.; *for mail*, 613 Debaugh Ave., Towson 4, Md.
Beynon, Charles Edmund (J'37) (ODEFKS), Tech. Engr. with H. H. Angus, Cons. Engr., 1221 Nay St.; *for mail*, 47 Playtee Cres., Toronto, Ont., Can.
Beyster, Henry Edward ('44) (STLMOD), Pres., H. E. Beyster Corp., 2100 Indus. Bank Bldg., Detroit 26, Mich.
Bezbatchenko, John W. (J'41) (ABMCJO), Flight Engr., Goodyear Aircraft Corp., 1210 Massillon Rd., Akron 15; *for mail*, 1088—6th Ave., Akron 6, Ohio.
Bhappu, Kavasji K. ('21; '35) (CEK), Managing Partner, Ch. Engr., Crystal Ice & Cold Storage Co., Kutchery Rd., Karachi, Sind, Bombay, India.
Bialog, Edward Stephen (J'35) (CJBM), Foreman, Welding Tube Mills, Steel & Tubes Div., Republic Steel Corp., 72 Scott Ave., Brooklyn; *for mail*, 575 Barbey St., Brooklyn 7, N.Y.
Bibeault, Gerald Joseph (J'42) (ANMCKO), Aero. Weight Engr., Chance Vought Aircraft Div., United Aircraft Corp.; *for mail*, 100 Nichols Ave., Stratford, Conn.
Bible, Willis B., Jr. (J'39) (LMC), Engr., Design, E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington; *for mail*, 110 Tennessee Ave., Pleasant Hill, Newport, Del.
Bice, Graham William (J'40) (SFKJ), Engr., Mech. Engr. Dept., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 7, N.Y.
Bice, Richard A. (J'46), Box 1663, Santa Fe, New Mex.
Bickel, Caldwell Conrad, Jr. (J'44) (NMHY), Engrg. Dept., Cincinnati Milling Mch. Co., Marburg Rd.; *for mail*, 4607 Erie Ave., Madisonville, Cincinnati, Ohio.
Bickel, Clifford Alcett ('35) (BHNJCG), Engr., Monarch Mch. Tool Co., 109 Oak; *for mail*, Bon Air Dr., Sidney, Ohio.
Bickel, Herman H. ('27; '34) (FSKBMC), V.P. Charge Engrg., Wickes Boiler Co., Saginaw; *for mail*, R. 6, Saginaw, West Side, Mich.
Bickel, Leonard A. ('37) (CEF), Lone Star Gas Co., 301 S. Harwood, Dallas 1, Tex.
Bickell, Gilbert James (J'42) (BNHS), Mech. Draftsman, Inland Steel Co., Block & Watling Sts.; *for mail*, 4728 Magoun Ave., East Chicago, Ind.
Bickford, Horace Lesley, Jr. (J'41) (SFMDEO), Engr., Gibbs & Cox, Inc., 21 West St., New York 6, N.Y.; *for mail*, 136 Delavan Ave., Newark 2, N.J.
Bickhardt, Homer Frank ('40) (SFCEKR), Supt. Power, Ford Motor Co. of Can., Ltd.; *for mail*, 2221 Pelissier St., Windsor, Ont., Can.
Biddison, P. McDonald ('15; '17; F'41) (EFB), Cons. Engr., 135 S. La Salle St., Chicago 3, Ill.
Biddle, Ralph Waldo (J'43) (ACMZND), Examiner (Indus. Engr.), War Assets Corp., Reconstr. Finance Corp., Chamber of Commerce, Springfield; *for mail*, Box 26, R.F.D. 1, Vandalia, Ohio.
Biddle, Richard Scull (J'38) (SCF), Combustion Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 1815 Palmer Ave., Larchmont, N.Y.
Biddle, Walter Allen ('37; '46) (AFK), Secy., Treas., Gen. Mgr., Gen. Engrg., Inc., Box 1037; *for mail*, 1025 Manzano Court, Albuquerque, New Mex.
Bidstrup, Herluf August ('42; '45) (AKL), Mech. Engr., F. L. Smidth & Co., 11 W. 42nd St., New York 18; *for mail*, 248 Hutchinson Blvd., Mt. Vernon, N.Y.
Bieber, Victor Emanuel (J'44), 521 Cumberland Ave., Chevy Chase 15, Md.
Biedermann, Robert Gustav (J'42) (CMA), Radio Technician, U.S.N., Box 6, Submarine Base, Navy 3002, F.P.O., San Francisco, Calif.; *for mail*, 425 W. 57th St., New York 19, N.Y.
Bier, Peter J. ('27; '35) (HSB), Sr. Engr., Bur. of Reclamation, Custom House, Denver 2, Colo.
Bierbaum, Christopher Henry ('94; '98; F'41) (JKM), V.P., Cons. Engr., Lumen Bearing Co., 197 Lathrop St., Buffalo 12, N.Y.
Bieshelt, Oscar (J'37) (BHNJDO), Ch. Engr., Enterprise Co.; *for mail*, 404 N. Firestone Ave., Columbiana, Ohio.
Biever, Edmund J. ('40) (DFJ), Ch. Mech. Engr., Kohler Co., Kohler, Wis.
Bigelow, Carle Muzzey ('17; '18; '21) (CM), Dir., Pharmaceutical Dept., Calco Chem. Div., Am. Cyanamid Co., Bound Brook, N.J.
Bigelow, Frank B. ('07) (CFS), Pres., Treas., Bigelow Liptak Corp., 2842 W. Grand Blvd., Detroit, Mich.
Biggar, William James (J'41) (BC), Lt., 47th Engrs., A.P.O. 957, c/o Postmaster, San Francisco, Calif.
Bigger, Trafford William ('26; '35) (SKFENB), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 12 Cornell Blvd., Schenectady 2, N.Y.
Biggerstaff, Edward D., Jr. (J'41) (NSEFHK), Asst. Mar. Engr., Naval Shipyard, Charleston; *for mail*, 16 Montagu St., Charleston 10, S.C.
Biggert, Florence Cotter, Jr. ('15) (JHBY), Chmn. of Bd., Charge Engrg., United Engrg. & Fdy Co., 1st Natl. Bank Bldg., Pittsburgh; *for mail*, 108 Hawthorne Ave., Crafton, Pittsburgh 5, Pa.
Biggs, George Allen ('23) (H), Secy., Ch. Engr., James Leffel & Co., 426 East St., Springfield, Ohio.
Biggs, William Francis ('35) (EFKLS), Power Engr., Hiram Walker & Sons, Inc., Foot Edmand St., Peoria 1, Ill.
Bilhuber, Paul H. (A'22) (WCA), V.P., Dir., Steinway & Sons, 109 W. 57th St., New York; *for mail*, 14 Richmond Rd., Douglaston, N.Y.
Bilimoria, Phiroze Dadyba (J'45), Ch. Engr., Wah Cement Works, Associated Cement Cos. Ltd., Wah, Punjab; *for mail*, 5 Elphinstone Rd., Poona, India.
Bilisoly, Charles Stoakes ('45) (FSZKCE), Test Engr., Potomac Elec. Power Co., 1st & V Sts., S.W., Washington, D.C.; *for mail*, 3612 Oval Dr., Alexandria, Va.
Bill, Clifford Ernest ('43) (DKL), V.P., Louisville Drying Mchy Co., 451 Baxter Ave.; *for mail*, 621 Cochran Hill Rd., Louisville, Ky.
Bill, Robert George (J'40) (CLM), Mech. Engr., Maint., Davison Chem. Co., Curtis Bay, Baltimore 26; *for mail*, 2223 E. Chase St., Baltimore 13, Md.
Billeter, Julius C. ('28; '38) (CFDE), Stokermatic Div., Rheem Mfg. Co., 1415 S. State St., Salt Lake City 4; *for mail*, 368 G St., Salt Lake City 3, Utah.
Billeter, Robert (J'32) (NMBC), Ch. Engr., Automotive Maint. Mchy. Co., 2100 Commonwealth Ave., North Chicago, Ill.
Billey, Peter R. (J'35) (BMT), Devel. Engr., Plymouth Cordage Co., Court St., North Plymouth; *for mail*, One Holmes Terrace, Plymouth, Mass.
Billhardt, Fred A. ('34; '45) (DCEJ), Ch. Engr., Alcoa Steamship Co., 17 Battery Pl., New York 4, N.Y.
Billich, William Herbert (J'28) (EFKS), Dist. Mgr., Combustion Engrg. Co., Inc., 1603 K St., N.W., Washington 6, D.C.
Billingham, William Gasson ('43) (HKLOSZ), Ch. Draftsman, S. D. Hicks & Son Co., 1671 Hyde Park Ave., Boston 36, Mass.
Billings, A. W. K. ('09) (BHS), Pres., S. P. Tramway Light & Power Co. Ltd., Caixa Postal 26-B, São Paulo, Brazil, S.A.
Billings, Edward J. ('12; '17) (PSFC), Exec. Asst., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Billings, Edward Russell (J'43) (KCBM), Capt., Ord. Dept., A.U.S., Weldon Springs Ord. Plant, Weldon Springs, Mo.
Billings, Frederic Church ('91) (CM), Chmn., Bd. of Dirs., Billings & Spencer Co., 1 Laurel St., Hartford 6, Conn.
Billings, John Edward (J'40) (SCF), Asst. Test Engr., Pub. Serv. Elec. & Gas Co. of N.J., 155 Raymond Blvd., Newark; *for mail*, 109 Park Ave., Verona, N.J.
Billings, John Harland ('14; '17; '21) (BJMO), Prof. Mech. Engrg., Head of Dept., Drexel Inst. of Tech., Philadelphia, Pa.
Billipp, Ernest H. ('21) (OLKDHC), 545 Boston Post Rd., Rye, N.Y.
Billmyer, Carroll Davis ('16; '19; '26) (CBFM), Assoc. Prof. Mech. Engrg., R.I. State College, Kingston, R.I.
Bills, Lester H. (J'41) (CKS), Field Engr., Gen. Elec. Co., 140 Federal St., Boston 1; *for mail*, 62 Huntley Rd., Quincy 69, Mass.
Bills, Max E. (J'39) (CJ), Spec. Engr., Sheet & Tin Mill, Carnegie-Ill. Steel Corp.; *for mail*, 1904 W. 5th Ave., Gary, Ind.
Bilocq, George Albert ('40) (DJMNT), Asst. Ch. Engr., Forano Ltd.; *for mail*, 132 Maple St., Plessisville, Que., Can.
Bilodeau, Alphonse Louis (J'41) (MJYL), Design Engr., Bryant Elec. Co., State St., Bridgeport 2; *for mail*, 324 Lynne Pl., Bridgeport 4, Conn.
Bilty, Charles Henry ('33) (ERS), Retired; 1008 N. 39th St., Milwaukee 8, Wis.
Bilyk, Mathew Harold (J'42) (NMB), Asst. Prof., Stevens Inst. of Tech., Hoboken, N.J.
Binda, Paul A. (J'41) (AEFSR), Mech. Engr., Texas Co., Box 509, Beacon, N.Y.
Binder, A. Francis (J'44), Engrg. Asst., Union Iron Works, 1500 Cascade St., Erie; *for mail*, E. 3rd St., Waterford, Pa.
Binder, Adolf R. (J'32), 90 Van Riper Ave., Rutherford, N.J.
Binder, Raymond Charles ('30; '37) (BAH), Prof. Mech. Engrg., Mech. Engrg. Sch., Purdue Univ., West Lafayette, Ind.
Binns, George W. ('17; '35) (BHM), Mech. Engr., Consultant, Cincinnati Milling & Grinding Mchs., Inc., Oakley, Cincinnati 9, Ohio.
Birch, Thomas ('40) (ACE), Comdr., Civil Engrs. Corps, U.S.N.R., Naval Shipyard, Navy 128, F.P.O., San Francisco, Calif.
Birckhead, Lennox ('40) (CSFGKY), Elec. Engr., Consltd. Gas, Elec. Light & Power Co., 407 Madison St., Baltimore 3; *for mail*, 529 Dunkirk Rd., Baltimore 12, Md.
Bird, Boyakin Old (J'43) (EKSJNH), Assoc. Engr., Chief, Specification Sec., The Engr. Bd., War Dept., Ft. Belvoir; *for mail*, 19 E. Masonic View Ave., Alexandria, Va.
Bird, Byron ('43) (HABCMR), Lt. Col., Corps of Engrs., U.S.A., Exec. Officer, U.S. Engr. Office, 1st & Douglas Sts., N.W., Washington 25, D.C.; *for mail*, 218 Kennedy Dr., Kenwood, Chevy Chase 15, Md.
Bird, Frederick Huey ('45) (FSC), Ch. Engr., Binkley Coal Co., 230 N. Michigan Ave., Chicago 1, Ill.
Bird, Frederick Joel ('45) (DJSNF), Engr., Stand. Dept., Link-Belt Co., 300 W. Pershing Rd., Chicago 9; *for mail*, 8108 S. Eberhart Ave., Chicago 19, Ill.
Bird, James Milton (J'42) (ARZDGJ), Struc. Test Engr., Chance Vought Aircraft Div., United Aircraft Corp., Main St., Stratford; *for mail*, 65 Rivercliff Dr., Devon, Conn.
Bird, Lee Garrison ('30; '32) (BCFSJN), Comdr., U.S.N.R., Naval Boiler & Turbine Lab., Naval Shipyard, Philadelphia 12, Pa.
Bird, Myron ('26; '35) (MJWC), Mgr., Calif. Saw Works, 721 Brannan St., San Francisco 3; *for mail*, 210 Kenyon Ave., Berkeley 8, Calif.
Bird, Ralph C. ('22) (DOC), Sales Engr., Williams Patent Crusher & Pulverizer Co., 2701 N. Broadway, St. Louis 6, Mo.
Bird, Stanley P. (J'34) (BHM), Project Engr., Breeze Corp., Inc., 41 S. 6th St., East Orange; *for mail*, 126 Forest Ave., Verona, N.J.
Birdsall, Howard Clarence (J'41) (OHB), Asst. Engr. with Claude W. Birdsall, 1700 F St.; *for mail*, 1821 Greenwood Terrace, Belmar, N.J.
Birdsall, Paul Everett (J'41) (FKS), Piston Ring Sales Co., 495 Peachtree St., N.E., Atlanta, Ga.
Birdseye, Clarence (A'30) (KLBCNM), Pres., Processes, Inc., Milwaukee, Wis.; *for mail*, Eastern Point Blvd., Gloucester, Mass.
Birdsong, John Miles (J'44) (CD), La Salle Steel Co., Hammond, Ind.
Birget, Charles Darrow ('37) (SBE), Engr., Commonwealth & Southern Corp., 212 W. Michigan Ave.; *for mail*, R.F.D. 6, Jackson, Mich.
Birk, Paul Martin (J'34) (AB), Sr. Design Engr. (Armament), Airplane Div., Curtiss-Wright Corp., 4300 E. 5th Ave., Columbus; *for mail*, 220 N. Kellner Rd., Columbus 9, Ohio.
Birkicht, Edward Roy ('34; '43) (CKLDMN), Supvg. Engr., Eastman Kodak Co., 343 State St., Rochester 4, N.Y.
Birkland, Stellan ('27; '29; '35) (ACJNRY), Mech. Designer, Am. Can Co. 320 Elizabeth Ave., Newark 8; *for mail*, 179 S. Harrison St., East Orange, N.J.
Birlik, Ertugrul (J'45) (BCHJMN), Capt., Turkish Army, Military Factory, Ankara, Turkey; *for mail*, 304 S. Oakland Ave., Pasadena 5, Calif.
Birmann, Rudolph ('23; '30) (ABK), Highland Farm, Newtown, Pa.
Birmingham, John N. (J'41) (YTSZ), Ensign, U.S.S. *Crockett*, A.P.O. 148, c/o F.P.O., San Francisco, Calif.
Birmingham, Michael A. ('44) (AHMR), 2825 Park Ave., Minneapolis, Minn.
Birn, Serge Alexander ('42) (COWMLN), Cons. Mgmt. Engr., 224 Reyburn Bldg., Louisville 2, Ky.
Birss, Ronald J. (J'38), Demonstrator in Thermodynamics, Mech. Bldg., Univ. of Toronto, Toronto; *for mail*, 200 Main St. N., Brampton, Ont., Can.
Birss, Thomas Kenneth (J'45) (HACSMA), Sales Engr., Sheldons, Ltd., Galt, Ont., Can.
Bischof, Gustave Joseph ('24; '31; '35) (MJCE), Secy., Phillips Cooling Tower Co., Inc., 114 Liberty St., New York 6; *for mail*, 471 Ocean Pkwy., Brooklyn 18, N.Y.
Bischoff, Clifton Anthony (J'42) (HSO), Application Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail*, Conover Ave., Middletown, N.J.

Bischoff, Robert ('30; '37) (BHCNOS), Sales Mgr., Ludlow Valve Mfg. Co., Inc.; *for mail*, 8 Lansing Ave, Troy, N.Y.
Bisen, Herbert (J'45) (COMJDT), Indus. Engrg. Consultant, E. R. Steinmann & Associates, 1 DeKalb Ave., Brooklyn 1; *for mail*, 644 Williams Ave., Brooklyn 7, N.Y.
Bises, George R. (J'41) (EHS), 330 E. 79th St., New York, N.Y.
Bishop, Franklin G. (J'43) (COT), Textile Workers Union of Am., 15 Union Sq., New York 3, N.Y.
Bishop, James O. ('34; '40) (EHS), c/o J. W. Beretta Engrs., Inc., 1205 Natl. Bank of Comm. Bldg., San Antonio, Tex.
Bishop, John Winslow (J'44), 1st Asst. Engr., U.S. Maritime Serv., Los Angeles Tanker Operators, Inc., Los Angeles, Calif.; *for mail*, North Bluff, Ottawa, Ill.
Bishop, Robert Ira (J'43), Ensign, U.S.N.R., U.S.S. *New Hanover*, AKA-73, c/o F.P.O., Post Office, San Francisco, Calif.
Bissell, George Welton ('90; '99), Retired; 169 N. Magnolia Ave., Monrovia, Calif.
Bissell, William S. ('45) (HK), Engr., Pipe Line Dept., Shell Oil Co., Inc., 1008 W. 6th St., Los Angeles 14, Calif.
Bisshopp, Kenneth Edward ('45) (BEA), Research Engr., Armour Research Foundation, 33 W. 33rd St., Chicago 16; *for mail*, 8441 S. Wolcott Ave., Chicago 20, Ill.
Bitko, Sol (J'42) (AEM), Ch. Engrg. Draftsman, N.Y. Naval Shipyard, Brooklyn; *for mail*, 557 Snediker Ave., Brooklyn 7, N.Y.
Bittner, Claude E. ('26) (C), Works Mgr., Intertype Corp., 360 Furman St., Brooklyn 2; *for mail*, 8914—63rd Ave., West Forest Hill, N.Y.
Biwer, Lynn W. (J'40) (ABJKH), Asst. Project Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail*, Sunbeam Lane, Swampscott, Mass.
Bixby, Walter ('07; '26), Mech. Engr., Pat. Atty., 1 Mt. Pleasant Terrace, Roxbury, Boston, Mass.
Bixler, Glenn Jennings (J'46), 1109 Madeline Ave., El Paso, Tex.
Bjerkan, Arthur J. (J'39) (BEG), Design Engr., John Deere Wagon Works; *for mail*, 2530—14th St., Moline, Ill.
Bjorklund, Edward Emanuel (J'31) (CST), Partner, Cambridge Woolen Mills, Cambridge, Minn.
Bjorkman, Roy K. A. (J'41) (CKS), 95 Jenness St., Lynn, Mass.
Black, Alexander Robson ('25; '44) (SHJETM), Sr. Applications Engr., Shell Oil Co., Inc., 50 W. 50th St., New York 20, N.Y.
Black, Archibald ('42) (AJMYWO), Provisional Internatl. Civ. Aviation Organization, Dominion Sq. Bldg., Montreal, Que., Can.; 116 Sackville Rd., Garden City, N.Y.
Black, Donald R. ('20; '35) (BCDHJK), 233 W. Hudson Ave., Englewood, N.J.
Black, Edgar N., 3rd ('14; '35) (FKS), Philadelphia Mgr., Fitzgibbons Boiler Co., Inc., 1717 Sansom St., Philadelphia, Pa.
Black, Henry Montgomery ('34; '40) (SOK BCF), Head, Mech. Engrg. Dept., Iowa State College, Ames, Iowa.
Black, John Alexander (J'44) (MYABDJ), Process Engr., Eclipse Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail*, 52 New Lawn Ave., Arlington, N.J.
Black, John F. (J'39) (FK), Summer Hill, Ill.
Black, John Thomas (J'40) (FKS), Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail*, 15—3rd St., Rumson, N.J.
Black, Joseph Elliott (J'39), Officer Candidate, T.C.S., Officer Candidate Sch., Class 18, Army Air Base, New Orleans 12, La.
Black, Lloyd Vernon ('44), Engr., Glass Research, Pittsburgh Plate Glass, Creighton; *for mail*, 1006 Oregon Ave., Tarentum, Pa.
Black, Paul Francis (J'42), Wright Aero. Corp., Cincinnati, Ohio.
Black, Paul Howard ('37) (NBL), Assoc. Prof. Mch. Design, Cornell Univ., Ithaca, N.Y.
Black, Samuel J. (J'42) (BZMHKJ), Asst. Mech. Engr., Naval Ord. Lab., Naval Shipyard, Washington; *for mail*, 312 Rittenhouse St., N.W., Washington 11, D.C.
Black, Stanley Bigelow ('24; '37) (BSJN), Turbine Engr., Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.
Black, Warren Richard (J'45), Ensign, U.S.N., Deck Officer, U.S.S. *Karin*, AF-33, F.P.O., San Francisco, Calif.; *for mail*, 8 S. Gifford St., Elgin, Ill.
Black, William Stewart (J'46) (BJK), Testing Engr., Crane Co., 836 S. Michigan Ave., Chicago; *for mail*, 9804 S. Seeley St., Chicago 43, Ill.
Black, Winfield Scott ('14; '35) (CDFKL), Retired; 7106 Oxford Rd., Baltimore 12, Md.
Blackall, Frederick Steele, Jr. ('29) (CJMDNY), Pres., Treas., Taft-Peirce Mfg., Co., Woonsocket, R.I.
Blackburn, Alfred T. ('31; '41) (CLM), Asst. Shop Supt., Cincinnati Milling Mch. Co., Cincinnati, Ohio.
Blackburn, Charles H. ('15; '24; '35), Pres., Treas., Hormiguero Central Corp., 60 Beaver St., New York, N.Y.
Blackburn, James Edward, Jr. (A'42), V.P., McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.
Blackman, Alfred O. ('22) (CKS), 47 Lafayette Pl., Greenwich, Conn.
Blackman, Robert Chester (J'36) (SCEK), Power Engr., Allison Div., Gen. Motors Corp., Indianapolis 6; *for mail*, 1267 N. Lynhurst Dr., Indianapolis 8, Ind.
Blackman, Vere C. ('27) (NMCHB), Ch. Engr., New Albany Mch. Mfg. Co., New Albany; *for mail*, Lincoln Heights, Jeffersonville, Ind.
Blackman, Vere Crosby (J'42) (COTYMK), Project Engr., Johnson & Johnson, 500 George St.; *for mail*, 108 Somerset St., New Brunswick, N.J.
Blackmon, Hendley Napier ('45) (NHZOCS), Elec. Editor, *Product Engineering*, McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18; *for mail*, 186-01 Jordan Ave., Hollis Manor, St. Albans 12, N.Y.
Blackmore, William ('43) (CMRY), Dist. Sales Mgr., Natl. Malleable & Steel Castings Co., Cleveland, Ohio; *for mail*, 219 E. Broad St., Richmond 19, Va.
Blackstone, Francis Bigelow ('29; '40) (SZO LH), Supt. of Plant & Tech. Adviser, Freeport Sulphur Co., Port Sulphur, La.
Blackwelder, Charles Davis ('19; '23; '24) (BO DEFJ), V.P., Charge of Engrg., Reynolds Metals Co., Reynolds Engrg. Bldg., Richmond, Va.
Blackwell, Hubert Charles ('21) (CEFHLS), Chmn., Bd. of Dirs., Cincinnati Gas & Elec. Co., 4th & Main St., Cincinnati 2, Ohio.
Blackwood, Frank A. (J'41), Ensign, U.S.N.; *for mail*, 1731 Leighton Ave., Los Angeles 37, Calif.
Blade, Ellis ('38; '40) (CMB), Sunol, Ala. Co., Calif.
Bladen, Arthur McKinley ('45) (OSEFKD), Owner, Neiler, Rich & Bladen, 431 S. Dearborn St., Chicago 5, Ill.
Blades, Roger Thomas (J'35) (ECKNS), Design Engr., Phillips Petroleum Co.; *for mail*, 216½ W. 6th St., Bartlesville, Okla.
Blaes, Viggo Alfred (J'45), 86 Sunset Ave., Verona, N.J.
Blaine, David George (J'41) (RSNA), Engr., Test Div., Westinghouse Air Brake Co.; *for mail*, 353 Marguerite Ave., Wilmerding, Pa.
Blair, Abbott Hugh ('27) (FKS), Owner, Firm of Abbott H. Blair, 513 Martin Bldg., Birmingham 3, Ala.
Blair, Ernest Lionel ('19; '26; '35) (SEFKLG), Ch. Bldg. Serv. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail*, 108 Willow Ave., Wollaston 70, Mass.
Blair, J. Arthur ('21; '26) (LCSD), Devel. Engr., Plant Expansion, Campbell Soup Co., 100 Market St., Camden; *for mail*, 245 Rhoads Ave., Haddonfield, N.J.
Blair, James G. (J'35) (ALY), 116 Hoover Rd., Yonkers 3, N.Y.
Blair, James Stanley ('44) (BHJZ), Supt. Engrg. Research, Messrs. Stewart & Lloyds, Ltd., Corby, Northants; *for mail*, 29 E. Carlton Pk., Nr. Market Harborough, Leics., England.
Blair, Maurice (J'39) (CMABD), Controller, A. C. Horn Co., 43-42—10th St., Long Island City 1; *for mail*, 35-54—83rd St., Jackson Heights, N.Y.
Blair, Sidney Martin ('43) (AEFY), Controller, Research & Devel., Trinidad Leaseholds Ltd., Salisbury House, London, E.C. 2, England; *for mail*, Suite 3005, 70 Pine St., New York 5, N.Y.
Blair, Thomas Howard ('32), Pres., Hercules Presses, Ltd., 69 York St., Toronto, Ont., Can.
Blaisdell, Benjamin Henry ('09), Retired; 6023 Robison Rd., Cincinnati 13, Ohio.
Blake, Alfred D. ('14; '22) (EFKS), Editor, *Combustion*, 200 Madison Ave., New York 16, N.Y.
Blake, Arthur Henry ('20; '26; '35) (ASEFT), Maj., Hdq., Army Air Forces, The Pentagon, Washington, D.C.; *for mail*, 3515 S. Wakefield St., Arlington, Va.
Blake, Francis Eugene (J'01) (BDJLWY), Sr. Draftsman, Drafting Div., Watervliet Arsenal, Watervliet; *for mail*, 1 North St., Delmar, Albany Co., N.Y.
Blake, Joel W. ('34; '35) (BES), Asst. Supt. of Generation, Okla. Gas & Elec. Co., 3rd & Harvey Sts.; *for mail*, 909 N.W. 16th St., Oklahoma City, Okla.
Blake, John Henry, Jr. ('32) (ESFK), Simplex Oil Heating Corp., 11 Park Pl., New York 7, N.Y.
Blake, John Holland (J'44), 14 Saratoga Rd., Yorktown Manor, East Greenwich, R.I.
Blake, John Paton (J'37) (SFK), Maj., U.S.A., Hdq., 11th Bn., Armd. Replacement Trng. Center, Ft. Knox, Ky.; *for mail*, 1485 W. 114th St., Cleveland 2, Ohio.
Blake, Larkin LaFayette, Jr. (J'42) (JMY), Ensign, U.S.N.R., WV-32, Rm. 260, Oak Ridge, Tenn.
Blake, Winchester Gould ('26; '34) (SEZJ), Application Engr., Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Blakeley, Samuel David (J'43) (CDM), Gen. Delivery, Hollywood, Fla.
Blakeman, S. P. ('39) (CD), Dist. Mgr., Haughton Elev. Co., 1730 Ludlow St., Philadelphia; *for mail*, 308 Hamilton Rd., Merion, Pa.
Blakeslee, Howard Russell ('18; '35) (MHBC), M.M., Hendey Mch. Co., 105 Summer St., Torrington; *for mail*, 40 Hillside Ave., Thomaston, Conn.
Blanchard, A. G. ('36) (CMS), Pres., Gen. Mgr., Inferno Co., Box 1138, Shreveport, La.
Blanchard, Arthur Horace ('43) (JLM), Box 7042, Roseville Branch P.O., Newark 7, N.J.
Blanchard, E. Payson ('35) (ACM), Gen. Sales Mgr., Bullard Co., Bridgeport, Conn.
Blanchard, Everett E. (J'40) (BDE), Tool Designer, Douglas Aircraft Co., Inc.; *for mail*, 309 S.W. 46, Oklahoma City, Okla.
Blanchard, Philip Dean ('28; '36) (RK), Sales & Serv. Engr., Superheater Co., 122 S. Michigan Ave., Chicago 3, Ill.; *for mail*, 2025 Jefferson Ave., St. Paul 5, Minn.
Blanchard, Rollo Kimball ('12; '14; '25) (PAC), V.P., Neptune Meter Co., 50 W. 50th St., New York, N.Y.
Blanchard, Walter Prescott ('45) (NM), Mech. Designer, Gen. Elec. Co., 920 Western Ave., Lynn; *for mail*, 64 High St., Hingham, Mass.
Bland, Frederick Vincent, Jr., (J'44) (CGJ), Project Engr., 473 Jefferson Ave., Elgin, Ill.
Blanding, Robert Lincoln ('13) (SKMOCF), Retired; 1385 Smith St., North Providence 11, R.I.
Blank, Harry Madison ('39) (HCEM), Seagrave Corp., 2000 S. High St., Columbus 7; *for mail*, 211 W. Dunedin Rd., Columbus 2, Ohio.
Blank, Herbert Arthur (J'43) (LEAC), Area Engr., Arabian Am. Oil Co., Bahrein Island, Persian Gulf; *for mail*, Arabian Am. Oil Co., A.P.O. 816, c/o Postmaster, New York, N.Y.
Blank, Walter Edward ('42) (CDJM), Supvr., Tool Div., Natl. Cash Register Co., Main & K Sts., Dayton 9, Ohio.
Blanton, Burt C. ('45) (GLO), Cons. Engr., 3708 Harvard Ave., Highland Park, Dallas 5, Tex.
Blaskowski, Henry John (J'41) (KSM), Ensign, U.S.N.R., 24-22—27th St., Astoria 2, N.Y.
Blatchley, Charles Gilbert ('45) (BLOHFS), Engr., Schutte & Koerting Co., 12th & Thompson Sts., Philadelphia 22; *for mail*, 403 S. Cedar Lane, Upper Darby, Pa.
Blatz, Walter Louis (J'42) (ENZOSF), Student Engr., Gen. Elec. Co., 1 River Rd.; *for mail*, R.D. 2, Sacandaga Rd., Schenectady, N.Y.
Blayney, James Porter (J'45) (CKYZH), Draftsman, Dow Chem. of Can., Ltd., Box F, Sarnia, Ont., Can.
Blechar, Theodore Joseph (J'45) (BNK), Jr. Engr., Manhattan Project, Box 1663, Santa Fe, New Mex.
Bledsoe, L. F., Jr. (J'40) (JKM), Inspr., Welding Engrs. Staff, Newport News Shipbldg. & Dry Dock Co.; *for mail*, 1062 Ferguson Ave., Newport News, Va.
Bleecher, Rudolph Joseph (J'44), Engr., Charge of Design, Aster Engrg. Co., 625 Parkside Ave., Brooklyn; *for mail*, 2820 Ocean Ave., Brooklyn 29, N.Y.
Blenko, Walter J. ('21; '31) (AJ), Partner, Stebbins, Blenko & Parmelee, Farmers Bank Bldg.; *for mail*, Box 1376, Pittsburgh, Pa.
Blevins, David Jesse (J'43), Route 5, Manhattan, Kan.
Bley, Robert Edmund (J'41) (DMCLBS), Indus. Engr., E. I. du Pont de Nemours & Co., Penns Grove; *for mail*, 389 Ave. I, Carney's Point, N.J.
Blickman, Bruce Barkin (J'44) (NMC), Prod. Engr., S. Blickman, Inc., 536 Gregory Ave., Weehawken, N.J.; *for mail*, 211 Central Park West, New York 24, N.Y.
Blirer, Arthur Edward (J'35) (CJM), Lt. Comdr., U.S.N.R., Research & Devel. Div., Bur. of Ord., Rm. 4247, Main Navy Bldg., Washington 25, D.C.; *for mail*, 33 Woodland Dr., Vanada Woods, N.J.
Bliss, Collins P. ('03; F'41) (AHS), Pres., Engineering Index, Inc., 29 W. 39th St., New York 18; *for mail*, Eton Hall, Scarsdale, N.Y.
Bliss, Conrad De K. (J'41) (CMDJB), Engr., Charge Spec. Apparatus Prod., Air Reduction Co., 181 Pacific Ave., Jersey City, N.J.
Bliss, Donald S. ('41; '43) (ZBLCM), Project Engr., Taylor Instrument Cos., 95 Ames St., Rochester 11; *for mail*, 1233 Crittenden Rd., Henrietta, N.Y.
Bliss, John Warren (J'39) (JKS), Asst. Gen. Foreman, Bethlehem Steel Co., Bethlehem, Pa.
Bliss, Richard (J'43) (SRACMK), Test Engr., Gen. Elec. Co.; *for mail*, 730 N. Brandywine Ave., Schenectady, N.Y.
Bliss, William Carter ('20; '25) (R), Col., 6320 McPherson Ave., St. Louis 5, Mo.
Bliss, William D. ('20) (CMJ), Dean, College of Engrg., Marquette Univ., 1515 Wisconsin Ave., Milwaukee; *for mail*, 416 N. 49th St., Milwaukee 13, Wis.
Bliss, Zenas Randall ('23; '30; '33) (BNCJM), Prof. of Engrg., Div. of Engrg., Brown Univ., Providence, R.I.
Blitz, Emmanuel ('31; '35), Mech. Engr., Engrg. Div., Dept. of Hospitals, City of New York, 125 Worth St., New York 13; *for mail*, 355 E. 187th St., New York 57, N.Y.
Blixt, Gustaf Frederick ('16; '23), Ch. Engr., Lederle Labs., Pearl River, N.Y.; *for mail*, 73 Rodney St., Glen Rock, N.J.
Blizard, John ('20) (BKFSAH), Cons. Engr., Foster Wheeler Corp., 165 Broadway, New York 6; *for mail*, 7 Cedar Pl., Garden City, N.Y.
Blizard, John Risque (J'35) (BCEHKM), Indus. Div., Corning Glass Works; *for mail*, 65 E. 4th St., Corning, N.Y.
Block, M. Sabel (J'39) (MCNDBL), Lt. (j.g.), U.S.N.R., Asst. Ord. Officer, Naval Shipyard, Terminal Island, Calif.

Bloedorn, Charles William (J'44) (SKEBCO), Application Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1; *for mail,* 7123 W. Wells St., Wauwatosa 13, Wis.
Blom, Carl ('36) (HDANWS), Ch. Engr., Byron Jackson Co., Slauson Ave., Huntington Park; *for mail,* 2114 Roanoke Rd., San Marino 9, Calif.
Blomquist, Claes Albert George, Jr. (J'36) (CN DLMR), Research Engr., Monsanto Chem. Co., Anniston; *for mail,* Lafayette, Ala.
Bloodgood, Richard Miller (J'39) (CEFKOS), 107 Colonial Village, Wilmington, N.C.
Bloom, K. Wm. ('40; '41) (RB), Checker, Am. Car & Fdy. Co., Berwick 6, Pa.
Bloomberg, David Jacob ('25; '35) (ASKFBR), Turbine Engr., Gen. Elec. Co., River Works, Lynn; *for mail,* 64 Fairmont Ave., Newton 58, Mass.
Bloomsburg, Marvin S. (J'27) (BKDFJS), Tech. Supt., E. I. du Pont de Nemours & Co., Box 993, Charleston 24, W. Va.
Bloomster, Edgar Leslie ('42) (DMOSJ), Head, Firm of E. L. Bloomster, 116 New Montgomery St., San Francisco 5, Calif.
Blose, James Frank (J'35) (EJNSZ), Assoc. Mech. Engr., U.S. Naval Engrg. Exper. Sta., Annapolis, Md.
Bloss, William Stanley (J'40), (EFSKJZ), Application Engr., Eng. Div., Worthington Pump & Mchy. Corp., Clinton & Roberts Ave., Buffalo 5, N.Y.
Blossom, Francis ('02; F'41; H'43) (CHOS), Mem. of Firm, Sanderson & Porter, 52 William St., New York 5, N.Y.
Blount, Thomas Harvey, Jr. (J'40) (CML), Capt., Ord. Dept., U.S.A., Commanding Officer, Morgantown Ord. Works, Morgantown, W. Va.
Blount, Wayne Lawrence (J'32) (CMKDFA), Dist. Serv. Mgr., Hartford Empire Co., 333 Homestead Ave., Hartford 10, Conn.; *for mail,* 100 Willow St., Olean, N.Y.
Blowney, Walter Edward ('38) (SBCJNA), Admin. Asst., Turbine Generator Engrg. Div., Gen. Elec. Co., 1 River Rd.; *for mail,* Box 61, R.F.D. 1, Schenectady, N.Y.
Blue, A. C. ('28; '35) (FKS), 94 Heddington Ave., Toronto, Ont., Can.
Bluestone, Edwin (J'40) (ARCZ), Mech. Engr., Ord. Dept., U.S.A., Aberdeen Proving Ground, Md.
Blum, Joseph Klee ('09; '19; '26) (SFOE), Cons. Engr., 205 E. 42nd St., New York 17, N.Y.
Blum, Samuel ('34) (SOFE), H. M. Wilson Co., 18th & Brandywine Sts., Philadelphia 30; *for mail,* 6739 N. Smedley St., Philadelphia 26, Pa.
Blum, Walter Werner (J'43) (BFS), 337 Meeting House Lane, Narberth, Pa.
Blumberg, Frank Eagan ('26; '35) (SKHDN), Asst. Ch. Engr., Todd-Pacific Shipyards, Inc., 2400—11th Ave., S.W., Seattle 4; *for mail,* 4330 E. 44th St., Seattle 5, Wash.
Blumberg, Leo ('21; '27) (EKJMND), Assoc. Prof. Mech. Engrg., Univ. of Del., Newark; *for mail,* 4 E. 14th St., Wilmington 33, Del.
Blume, LeRoy O. (J'40) (AER), 6513 N.W. Highway, Chicago, Ill.
Blundell, Eustace Ezra ('17; '35) (CMNHZ), Pres., Mgr., Commercial Centerless Grinding Co., 6603 Cedar Ave., Cleveland 3, Ohio.
Blunt, Richard R. (J'34) (BKR), Loco. Engrg. Div., Gen. Elec Co.; *for mail,* 4112 Iroquois Ave., Lawrence Park, Erie 2, Pa.
Boa, James Caldwell (J'45) (ZLYTFH), Sales Engr., Taylor Instrument Cos. of Can., Ltd., 110-12 Church St., Toronto; *for mail,* 289 Rushton Rd., Toronto 10, Ont., Can.
Boak, Thos. I. S. ('22), Factory Mgr., Winchester Repeating Arms Co., 275 Winchester Ave.; *for mail,* 280 Livingston St., New Haven, Conn.
Boals, Ray B. ('41) (SHFKW), Gen. Supt., Secy. Eugene Water Board, 1116 Willamette St.; *for mail,* 1706 Jefferson St., Eugene, Ore.
Boals, Roy Logan (J'45) (BNLJSR), Student, Test Engr., Gen. Elec. Co., Schenectady; *for mail,* 1046 University Pl., Schenectady 8, N.Y.
Boals, Wayne Stanley (J'39) (ANK), Mech. Engr., Aircraft Div., Cornelius Co., 1621 E. Hennepin Ave., Minneapolis 14, Minn.
Board, Samuel Sikes, Jr. (J'38) (MCDBHJ), Methods & Prod. Control Engr., Farrel-Birmingham Co., Inc., 25 Main St.; *for mail,* 9 Clover St., Ansonia, Conn.
Boardman, Clark Clinton ('20) (CDYLKE), Mgr., Thermatomic Carbon Co., Sterlington, La.
Boardman, Harry Clow ('39) (BJKH), Research Engr., Chicago Bridge & Iron Co., 1305 W. 105th St.; *for mail,* 10357 S. Hoyne Ave., Chicago 43, Ill.
Boas, Robert H. ('29) (DFS), Gilbert Associates Inc., 412 Washington St., Reading, Pa.
Boaz, John Rutherford ('39) (N), Owner, Firm of John R. Boaz—Engineering, 119 E. 23rd St., Los Angeles 2; *for mail,* 1018 E. Santa Anita Ave., Burbank, Calif.
Bobbe, Richard Arthur (J'42) (CMO), Lt., U.S.N.R., Asst. Terminations Officer, Office of Inspr. of Naval Matl., U.S.N., c/o Gen. Elec. Co., Schenectady 1; *for mail,* c/o Norwalk, 760 West End Ave., New York 25, N.Y.
Bobbitt, Bailey M., Jr. (J'41) (ACJ), 1505 Crane Court, Midland, Mich.
Bobrowsky, Alfred R. (J'41) (EAB), Mech. Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 66 E. Bridge St., Berea, Ohio.
Bochenek, Alfred Felix (J'37) (SFEJ), Editorial Assistant, A.S.M.E., 29 W. 39th St., New York 18, N.Y.; *for mail,* 879 Springfield Ave., Summit, N.J.
Bock, Arthur Emil (J'40) (SEF), Ensign, U.S.N.R., Instr., Mar. Engrg. Dept., U.S. Naval Acad., Annapolis, Md.
Bock, Edward John, Jr. (J'41) (CD), Plant Engr., Monsanto Chem. Co., Columbia, Tenn.
Bock, Louis S. (J'31), 2nd Lt., Sig. Sec., N.Y. Gen. Depot, 1st Ave. & 58th St., Brooklyn, N.Y.
Bockstahler, L. A. (J'35), 1st Lt., Ord. Dept., 710th Ord. Co., Air Base, Fresno, Calif.
Bodden, Thomas Robert (J'44) (DN), Engr., Transmission Equip., Continental Gin Co., 4500—5th Ave. S., Birmingham; *for mail,* 232 S. 8th St., Gadsden, Ala.
Boddy, Frederick Acton ('44) (SFLDMO), Ch. Engr., H. E. Beyster Corp., 2100 Indus. Bank Bldg., Detroit 26, Mich.
Bode, Charles Henry (J'41) (NBLCJO), Engr., Sharon Steel Corp., Broadway, Farrell, Pa.; *for mail,* 236 Laird Ave., S.E., Warren, Ohio.
Bodenschatz, August (J'39) (ABHJN), Designer, I.T.E. Circuit Breaker Co., 501 N. 19th St., Philadelphia 30; *for mail,* 7734 Burholme Ave., Philadelphia 11, Pa.
Bodensieck, Ernst Justus (J'43) (HAZBFE), Devel. Engr., Thompson Products, Inc., 2196 Clarkwood Rd., Cleveland 3; *for mail,* 421 E. 222nd St., Euclid 17, Ohio.
Bodger, W. Kenneth (J'41) (BNKSEJ), Project Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 594 Main St., South Glastonbury, Conn.
Bodine, Albert George, Jr. (J'41) (ABE), Bodine Soundrive Co., 3300 Cahuenga Pass, Los Angeles; *for mail,* 13120 Moorpark Blvd., Van Nuys, Calif.
Bodine, Joseph Hall, Jr. (J'44) (BHL), Rm. 462, Univ. of Chicago, Internatl. House, 1414 E. 59th, Chicago 37, Ill.
Bodinson, Frederick William (J'41) (DJNMHB), Pres., Gen. Mgr., Bodinson Mfg. Co., 2401 Bayshore Bldg., San Francisco 24, Calif.
Bodmer, Erwin Edward (J'41) (BJKMZ), Capt., Hdq., 250th Engr. (Combat) Bn., A.P.O. 58, New York, N.Y.; *for mail,* 4206 N. Wolcott Ave., Chicago 13, Ill.
Bodmer, Jakob Emil ('45) (EBKSAM), Ch. Engr., Devel. & Design, Precision Devel. Co., College Ave., Salisbury, Md.; *for mail,* 703 Winchester Ave., Trenton 9, N.J.
Bodnar, John (J'43) (CBM), Asst. Supt., Belknap Mfg. Co., 800 Union Ave., Bridgeport; *for mail,* 160 Oakland Pl., Stratford, Conn.
Bodwell, Howard Lawrence ('07) (MCJ), Branch Mgr., A. Milne & Co., 604 Frick Bldg., Pittsburgh 19, Pa.
Boe, Harvey A. (J'39), 817—2nd St. N., Fargo, N.D.
Boeckling, George A. (J'40) (MCJY), Mech. Engr., West. Automatic Mch. Screw Co., Lake Ave., Elyria, Ohio.
Boehm, William Henry ('00) (CBNS), Retired; 225 W. 106th St., New York 25, N.Y.
Boehringer, H. A. (J'30) (EAFMH), Product Engr., Saginaw Steering Gear Div., Gen. Motors Corp., N. Hamilton St.; *for mail,* 111 N. Andre St., Saginaw, Mich.
Boekelman, H. L. ('33; '44) (SFKNEL), Supvr., Meter & Customer Relations Dept., Ill. Maint. Co., 72 W. Adams St., Chicago 3, Ill.
Boelter, Llewellyn M. K. ('30) (BDKLMO), Dean, College of Engrg., Univ. of Calif., 405 Hilgard Ave.; *for mail,* 254 Bronwood St., Los Angeles 24, Calif.
Boenig, Robt. W. ('92; '02), Fuel Engr., Acme Min. & Fuel Corp., 46 Sutton St.; *for mail,* 647 Meeker Ave., Brooklyn, N.Y.
Boer, Elmer James (J'37) (FRSE), Research Engr., Fuels Div., Battelle Memorial Inst., 505 King Ave., Columbus 1; *for mail,* 75 Brevoort Rd., Columbus 2, Ohio.
Boesch, John C., Jr. (J'43) (CMT), Lt. Col., 765th Ord. (LM) Co., A.P.O. 200, c/o Postmaster, New York, N.Y.
Boettcher, Richard A. (J'41) (BCL), Ensign, U.S.N.R.; *for mail,* Gen. Delivery, Seattle, Wash.
Boettiger, Russell William (J'37) (ZSLEKY), Sales Mgr., Leslie Co., Grant Ave., Lyndhurst; *for mail,* 12 Ferncliff Terrace, Glen Ridge, N.J.
Bogard, Benj. T. (J'38) (CES), Asst. Prof. Mech. Engrg., La. Poly. Inst.; *for mail,* Box 465, Ruston, La.
Bogardus, Frederick James ('45) (NMCBLO), Engr., Mech. Engrg. Dept., Scott Paper Co., Front & Market Sts., Chester; *for mail,* Box 815, Swarthmore, Pa.
Bogart, William Mask (J'42), Apt. 23-B, Corbin Gardens-Plaza, Wilmington, Del.
Bogdanoff, John L. (J'38) (ABE), 123 N. Middletown Rd., Pearl River, N.Y.
Bogden, Edward Paul (J'44) (JNSKZ), Engr., Ceramics Plant, Linde Air Products Co., Box 95; *for mail,* 12 Landers Rd., Kenmore 17, N.Y.
Boger, Clair E. (J'29) (CGYMEA), Engrg. Dept., Hoover Co.; *for mail,* 742 Portage St., North Canton, Ohio.
Boggiano, James Edison ('24; '32) (BOS), Cons. Engr., 2514 S. Adams St., Arlington, Va.
Boggs, Robert William ('42) (LGCWJS), Asst. Mgr. Gen. Publicity Dept., Union Carbide & Carbon Corp., 30 E. 42nd St., New York 17; *for mail,* 170 Morris Ave., Rockville Centre, N.Y.
Bogle, Roy Thomas (J'40) (BCMNDY), Asst. Engr., Canadian Gen. Elec. Co., Ltd., Peterborough, Ont., Can.
Bogot, Alexander ('44) (FSK), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 343 S. Austin Blvd., Oak Park, Ill.
Bogue, Joseph Creswell ('42) (EFS), Cons. Steam Engr., 25 Huntington Ave., Boston, Mass.
Bohanon, Hoy Rolla (J'44) (AHENCM), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 16719 Clifton Blvd., Lakewood 7, Ohio.
Bohjelian, Albert G. (J'42) (GJM), U.S.N., Asst. Engrg. Officer, U.S.S. *Biloxi,* CL-80; *for mail,* 6203 Chestnut St., Philadelphia 39, Pa.
Bohler, Robert Allen (J'40) (AMSEB), Engr., Supvr., Consltd. Vultee Aircraft Corp., Pacific Highway, San Diego 12; *for mail,* 4054 Goldfinch, San Diego 3, Calif.
Bohn, Louis G. ('29; '35; '35) (CLMYJD), Mech. Engr., Allen Engrg. Co., 3 William St., Newark 2; *for mail,* R.F.D. 1, Valley View Park, Whippany, N.J.
Bohn, Robert G. ('16; '24; '28) (SGMD), Plant Engr., Mich. Carton Co., 79 E. Fountain St.; *for mail,* 187 N. Union St., Battle Creek, Mich.
Bohne, Laurence H. (J'40), Pub. Works Officer, U.S.N., Twenty-Nine Palms, Calif.; *for mail,* 1822 Dublin St., New Orleans, La.
Bohnlofink, John ('41) (KLSZDG), Ch. Engr., U.S. Indus. Chemicals, Inc., 5101 Andard Ave., Baltimore 26; *for mail,* 4009 Loch Raven Blvd., Baltimore 18, Md.
Bohnstengel, Walter ('15; '24) (RFJESK), Test Engr., Atchison, Topeka & Santa Fe Ry., Motive Power Bldg., Topeka, Kan.
Bohuslav, Hans Aloys ('43) (BCEO), V.P., Sterling Eng. Co., 1270 Niagara St., Buffalo 13, N.Y.
Boies, Harry Bainbridge ('18; '29; '35), Indus. Engr., Athey Products Corp., 5631 W. 65th St., Chicago 38; *for mail,* 10605 S. Wood St., Chicago 43, Ill.
Boisclair, Hugh Cappes ('44) (BKOSTM), Mgr., Firm of Hugh C. Boisclair, 927-9 Martin Bldg., Birmingham 3; *for mail,* 110 Ridge Rd., Homewood Sta., Birmingham 9, Ala.
Boise, Robert W., Jr. ('40; '42) (CMA), Indus. Engr., Puget Sound Naval Shipyard, Bremerton; *for mail,* Box 135, Route 1, Bellevue, Wash.
Boisvert, Jean Baptiste ('31; '35) (BDLMNW), Plant Engr., Tech. Adviser, Forano, Ltd.; *for mail,* 128 Maple St., Plessisville, Que., Can.
Bokair, William J. (J'45) (CJY), 113—84th St., Brooklyn 9, N.Y.
Bokorney, Fred R. (J'40) (BCM), Engr., Balancing Mch. Div., Gisholt Mch. Co., 1245 E. Washington St., Madison 3, Wis.
Bokum, William H. ('42) (OCM), V.P., George S. Armstrong & Co., Inc., 52 Wall St., New York 5, N.Y.; *for mail,* Tod's Driftway, Old Greenwich, Conn.
Boland, John Rudolph ('44) (JN), Owner & Gen. Mgr., Modern Mfrs., 1216 Nadeau St., Los Angeles 1; *for mail,* 4149—2nd Ave., Los Angeles 43, Calif.
Boland, L. C., Jr. ('41) (OSHKFB), Firm of L. C. Boland, Jr., 774 Spring St., N.W.; *for mail,* 1140 Rosedale Dr., N.E., Atlanta, Ga.
Boldi, Hugo Anthony (J'45) (SEKFHZ), 272-8 Bleecker St., New York 14, N.Y.
Boles, Roger S. (J'40), Lt., VF-19, F.P.O., San Francisco, Calif.
Bolgiano, Gilbert Frederick ('24; '35) (ONMC), 2918 N. Rogers Ave., Baltimore 7, Md.
Boli, Louis Gwinner (J'43) (EABJKN), Research Engr., Farr Co., 2615 Southwest Dr.; *for mail,* 9520 Ceylon Ave., Los Angeles 45, Calif.
Bolin, Marcel E. ('26; '35), 1021 St. Paul St., Baltimore, Md.
Bollenback, Alfred William ('23; '35) (SKFO EM), Project Engr., Jackson & Moreland, 31 St. James Ave., Boston 16, Mass.
Boller, Harry B. (J'38) (NZCOB), Partner, Boller & Chivens, 1118 Mission St., South Pasadena; *for mail,* 1526 S. 7th St., Alhambra, Calif.
Bolles, Sidney Lynn (J'39) (ABE), Ch. Engrg. Draftsman, N.Y. Naval Shipyard, Brooklyn; *for mail,* 2041 Gates Ave., Brooklyn 27, N.Y.
Bollinger, Robert Hamilton (J'43) (ABNJEO), 1st Lt., Army Air Forces, 1809 Parkside Dr., N.W., Washington 12, D.C.
Bolstad, Milo M. (J'38) (SEZBJK), Asst. Prof. Mech. Engrg., Mech. Engrg. Dept., Univ. of Mo.; *for mail,* 806 W. Broadway, Columbia, Mo.
Bolton, Benjamin Franklin (J'40) (LNCDSA), Project Engr., Chas. Lennig & Co., 5000 Richmond St., Philadelphia 37, Pa.
Bolton, James Alfred (J'41) (MCOJND), Lt. (j.g.), U.S.N.R., N.Y. Naval Shipyard, Brooklyn 1; *for mail,* 69 Woodlawn Ave., Valley Stream, L.I., N.Y.
Bolz, Harold August ('37; '45) (NBCMJ), Assoc. Prof. Mech. Engrg., Purdue Univ., Lafayette, Ind.
Bolz, Ray Emil (J'42) (ABHJK), Head, Jet-Eng. Combustion Sec., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 23524 Marion Ave., Rocky River, Ohio.
Bolz, Roger William (J'42) (CHLMN), Asst. Editor, *Machine Design,* Penton Publ. Co., Penton Bldg., Cleveland 13; *for mail,* 3058 Livingston Rd., Cleveland 20, Ohio.
Bolz, Wilbur J. (J'38) (JMCN), Mech. Engr., Triangle Tool & Mfg. Co., 3415 N. Ashland Ave.; *for mail,* 4237 N. Hermitage Ave., Chicago 13, Ill.

Bombick, Henry Norman (J'46), 137 Groveland St., Pittsburgh 27, Pa.
Bonanno, Joseph Louis ('30; '37) (JBMYZL), V.P., Lionel Corp., 28 Sager Pl., Irvington 11; *for mail*, 56 Woodland Rd., Madison, N.J.
Bonar, Robert Barnett (J'42) (AHS), Ensign, LST-218, c/o Postmaster, San Francisco, Calif.
Bond, Francis Marvin ('09; '15) (CM), Asst. Dir., Indus. Bur. of Baltimore, 22 Light St., Baltimore 2; *for mail*, 5811 Pimlico Rd., Baltimore 9, Md.
Bond, Horace A. ('29; '30), Albany Office Mgr., Warren Webster Co.; *for mail*, 12 Ramsey Pl., Albany, N.Y.
Bond, J. Stephen ('43) (FSRE), Combustion Engr., Enos Coal Mining Co., 1405 Merchants Bank Bldg., Indianapolis 4, Ind.
Bond, James Clarence, Jr. (J'42) (LDY), Mech. Engr., E. I. du Pont de Nemours & Co., Box 993, Charleston 24; *for mail*, 1630 Franklin St., Charleston 1, W.Va.
Bond, P. Clark ('25; '34; '35) (BDEFNS), Designing Engr., Plant Engrg. Co., 408 S. Spring St., Los Angeles 13; *for mail*, 1357 Linden Ave., Glendale 1, Calif.
Bond, Reuben (J'39), Instr. Mech. Engrg., Univ. of Calif., Berkeley; *for mail*, 2511 Hearst Ave., Berkeley 4, Calif.
Bond, Rufus E. ('27) (CEA), 60 Wall Tower, New York, N.Y.
Bond, William Goodyear ('28) (KBM), Exper. Engrg., Holly Oak, Del.
Bond, William Howard (J'46), 4196 Rochester Rd., San Diego 4, Calif.
Bond, William Roger (J'44) (NJBLHM), Engr., Mch. Designer, Bird Mch. Co., Neponset, South Walpole; *for mail*, 107 Samoset Ave., Mansfield, Mass.
Bone, Herbert Lawrie ('23; '35) (RJMBC), Mech. Engr., Union Switch & Signal Co., Swissvale, Pa.
Bonfrisco, Joseph John (J'44) (CMLBJK), Methods Engr., New Departure Div., Gen. Motors Corp., Bristol, Conn.; *for mail*, 2076—1st Ave., New York 29, N.Y.
Boniface, John Berkley ('26; '33; '35) (SKF), Planning Engr., Gas Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Bonifaci, Leonard Peter (J'44) (ZMNC), Engr., Seidelhuber Iron & Bronze Works, 1421 Dearborn St., Seattle 44; *for mail*, 3940 W. Kenyon, Seattle 6, Wash.
Bonine, Charles E. ('14) (BKOYSD), Cons. Engr., 100 Surrey Rd., Melrose Park, Philadelphia 26, Pa.
Bonini, Joseph D. (J'41) (BHK), Sales Mgr., Sier-Bath Gear Co., 9252 Hudson Blvd., North Bergen, N.J.; *for mail*, 84-68—127th St., Richmond Hill, N.Y.
Bonnell, John Raymond ('42) (SEHCR), Asst. Boiler Rm. Engr., Commonwealth Edison Co., 3400 N. California Ave., Chicago 18, Ill.
Bonner, Harold William (J'45) (ZSCMBL), 597 San Luis Rd., Berkeley 7, Calif.
Bonner, Harry H. ('28; '35) (BAE), Teacher, Philadelphia Bd. of Education, Parkway at 21st St., Philadelphia; *for mail*, 4634 Adams Ave., Philadelphia 24, Pa.
Bonnett, L. B. ('11) (AR), Retired; 310 W. Jersey St., Elizabeth, N.J.
Bonnett, Leland Brewer ('40) (CS), V.P., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Bonney, Robert Harlan (J'41) (JMCDE), Seaman, 1st Class, Radio Materiel Sch., Class 18-45, Naval Research Lab., Washington 20, D.C.
Bonsal, Richard Irving (J'42) (TNBDGM), Research Engrg. Sec., Exper. Sta., E. I. du Pont de Nemours & Co.; *for mail*, 1600 Broom St., Wilmington, Del.
Bonsall, Judson ('27; '35) (CEJ), Supt. Compressing Stas., Pittsburgh & W.Va. Gas Co., Union Bank Bldg.; *for mail*, 118 Waverly Way, Clarksburg, W.Va.
Bonstow, Thomas Lacey ('44), Cons. Engr., Whitehall Securities Corp., Ltd., 47 Parliament St., London, S.W. 1; *for mail*, 47 Fairdene Rd., Coulsdon, Surrey, England.
Bony, Ernest Marcel (J'43) (AEM), Lt., 1216 Post St., Alameda, Calif.
Bonza, Louis F. (J'46), Struc. Research Engr., Lockheed Aircraft Corp., Burbank; *for mail*, 380½ W. California Ave., Glendale 3, Calif.
Booij, Max Lorenz (J'44) (SEKABH), Mech. Engr., Netherlands Pur. Comm., 41 E. 42nd St., New York 17; *for mail*, Apt. 28, Tennis Pl., 6 Burns St., Forest Hills, N.Y.
Booker, Robert James (J'44) (NMCE), Engr., Nordberg Mfg. Co., 3073 S. Chase Ave.; *for mail*, 3019 S. Herman St., Milwaukee 7, Wis.
Bookmiller, Wayne Hale (J'41) (BRACKF), Jr. Engr., Gen. Elec. Co., 920 Western Ave., Lynn 3; *for mail*, 16 Nason Rd., Saugus, Mass.
Bookmyer, Ray Francis (J'40) (BSJMF), Engr., Turbine-Generator Div., Bldg. 56-3, Gen. Elec. Co., Schenectady 5; *for mail*, 112 Victory St., Schenectady 7, N.Y.
Bookout, Ervin J. ('43) (CKH), Sales Engr., Cochrane Corp., 3146 N. 17th St., Philadelphia 32, Pa.
Books, Thomas Victor (J'43) (HEANY), Draftsman, Celanese Corp. of Am., Ferry St., Newark, N.J.; *for mail*, 21 Pearl St., Stapleton, S.I., 4, N.Y.
Booraem, J. Francis ('95; '06), Cons. Engr., Specialist in Swimming Pool Design, Constr. & Equip., Shore Rd., Greenwich, Conn.
Boorman, Earl Dudley, Jr. (J'46), 148 DeWindt Rd., Winnetka, Ill.
Booth, Charles Arthur ('09) (KLSCDT), V.P. Charge Sales, Buffalo Forge Co., 490 Broadway, Buffalo 4, N.Y.
Booth, Daniel M. ('33) (CEFLRS), Spec. Rep., Worthington Pump & Mchy. Corp., Apartado 1724, Mexico, D.F., Mex.
Booth, Howard R. (J'38) (MCTJ), Indus. Engr., Crawford Mfg. Co., 3rd & Decateur, Richmond; *for mail*, 17 N. Boulevard, Richmond 20, Va.
Booth, Howard Worthington (J'42) (CHESLM), Mgr. Sales, Wilson-Snyder Mfg. Div., Oil Well Supply Co., Braddock, Pa.
Booth, James Holden ('44) (AYBCJM), Ch. Engr., Thompson Products, Inc., 2950 E. Woodbridge, Detroit 7, Mich.
Booth, John Wright (J'38) (BCMN), Shop Supt., Boyles Bros. Drilling Co. Ltd., 1291 Parker St., Vancouver, B.C., Can.
Booth, Paul E. ('40) (BHMA), Partner, Oatis-Booth Mch. Co., 617 Merchants Bank Bldg., Indianapolis 4, Ind.
Booth, Robert Lea (J'42) (BSOY), Ch. Engr., Korfund Co., Inc., 48-15—32nd Pl., Long Island City 1; *for mail*, 548 Cherry Lane, Floral Park, L.I., N.Y.
Booth, Theodore H. ('41) (CMSJDN), Works Mgr., Walworth Co., Greensburg, Pa.
Boothroyd, Howard Wallace (J'45) (SENHYC), Ensign, U.S.N.R., Asst. Gunnery Officer, U.S.S. *Wisconsin*, BB-64, F.P.O., New York, N.Y.; *for mail*, 27 Bertram St., Beverly, Mass.
Booton, Glen Harold (J'43) (MLADNC), Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail*, 8 Woodmere Rd., West Caldwell, N.J.
Booy, John E. (J'46), 4136 Wabash Ave., Kansas City 4, Mo.
Borchardt, Albert Hugo ('12; '16; '35) (HLS YCE), Asst. V.P., Worthington Pump & Mchy. Corp., Worthington Ave., Harrison; *for mail*, 90 Cooper Ave., Upper Montclair, N.J.
Borchardt, William Joseph (J'45) (AHK), Indus. Design Labs., 2062 Sawtelle Blvd., Los Angeles 25, Calif.
Borcherdt, Walter O., Jr. (J'38), 92 Lake Dr., Mountain Lakes, N.J.
Borchert, Edward Walter (J'44), 2735 W. Harrison Ave., Milwaukee 7, Wis.
Bordé, Harry Jamieson (J'45) (OJR), Stress Analyst, Preco, Inc., 960 E. 61st St., Los Angeles 1; *for mail*, 2185 Mandeville Canyon Rd., West Los Angeles 24, Calif.
Borden, Arthur Robert ('31; '35) (SFAZCE), Dist. Mgr., Hagan Corp., 2512 Book Bldg., Detroit 26, Mich.
Borden, James Paul, Jr. (J'44) (ACHM), Ensign, U.S.N., Navy 3923, c/o Capt. of Yard, F.P.O., San Francisco, Calif.; *for mail*, 606 Lyons Ave., Charlottesville, Va.
Borden, Jos. H. ('38) (BHJ), 1929 Fernwood Ave., Toledo 7, Ohio.
Borden, Moro Miller ('13) (HLMNSZ), V.P., Simplex Valve & Meter Co., 68th & Upland Sts., Philadelphia 42, Pa.; *for mail*, 310 Lees Ave., Collingswood, N.J.
Bordt, Frederick John, Jr. (J'41) (EBFAHK), Asst. Prof., Rensselaer Poly. Inst., Troy; *for mail*, Thais Rd., R.F.D. 1, Averill Park, N.Y.
Borg, Elmer H. ('21; '26) (DEF), Partner, Brooks-Borg, Archs. & Engrs., 820 Hubbell Bldg., Des Moines 9; *for mail*, 3101 Easton Blvd., Des Moines 17, Iowa.
Borgel, Bernard Francis (J'45), Plant Supt., Pa. Elec. Co.; *for mail*, 616 Ohio St., Erie, Pa.
Boring, Kenneth L. (J'40), Supvg. Engr., Gen. Elec. Co., 535 Smithfield, Pittsburgh 22, Pa.
Boring, Maynard McKay ('42), Staff Asst. to V.P., Charge Engrg., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Boring, Robert Franklin (J'43) (LSZOHW), Engr., Aircraft Gas Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, 21 Beach Rd., Lynn, Mass.
Boris, William E. (J'42) (AEFR), Research Engr., Stand. Oil Co. (Ind.), 119 Front St., Whiting, Ind.
Borland, John (J'43) (NBEAHR), Engr., Clark Equip. Co., Buchanan, Mich.
Borman, Raymond John, Sr. (AKFCZ), Research Engr., Douglas Aircraft Co., 3000 Ocean Park Blvd.; *for mail*, 2202-A 20th St., Santa Monica, Calif.
Bormann, Henry R. ('24; '35) (KLS), V.P., Krenz & Co., Inc., 5114 W. Center St., Milwaukee; *for mail*, 1748 N. 59th St., Milwaukee 8, Wis.
Born, Maynard Robert ('41) (CYLJKS), Ch. Engr., Shell Chem. Corp., Box 2527, Houston 1, Tex.
Born, William G. ('19), Mech. Engr., John Mohr & Sons, Inc., 3200 E. 96th St.; *for mail*, 8401 S. Oglesby Ave., Chicago 17, Ill.
Borovoy, Samuel ('42) (DJLMN), Engr., Indus. Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Borst, Maurice Andrew (J'45), Rd. Test Engr., Research Lab., Socony-Vacuum Oil Co., Paulsboro; *for mail*, Glassboro Rd., Woodbury Heights, N.J.
Borst, W. Lyle ('43) (SFKOE), Ch. Engr., Power Serv. Corp., 711 Wesley Temple Bldg., Minneapolis 4, Minn.
Borton, George Wills ('21) (AB), Pres., Gen. Mgr., Pa. Crusher Co., 17th Fl., Liberty Trust Bldg., Philadelphia 7, Pa.
Borton, Walter C. (J'40) (NJMG), Mech. Design Engr., Gen. Elec. Co., 100 Woodlawn Ave.; *for mail*, 11 Newton Ave., Pittsfield, Mass.
Bos, Peter H. (J'38), Bos-Hatten, Inc., Elk & Keating Sts., Buffalo 10, N.Y.
Bose, Kumudini Kanta ('18; A'22), 9-A Rajpur Rd., Delhi, India.
Bosler, Krell ('39; '45) (MJCD), Ch. Engr., John Douglas Co., 71st St. & B. & O. R.R., Cincinnati 16; *for mail*, Box 71, Route 15, Cincinnati 15, Ohio.
Bosler, Lester Comly ('19) (FS), Mech. Engr., Coleman & Co., Inc., 123 S. Broad St., Philadelphia 9, Pa.
Bosomworth, George Palliser ('45), Mgr., Engrg. Lab., Firestone Tire & Rubber Co., 1200 Firestone Pkwy., Akron 17; *for mail*, 81 Sand Run Rd., Akron 13, Ohio.
Boss, Millard O. (J'43) (BLZ), Pvt., A-12-4, 2nd Platoon, Field Arty. Replacement Trng. Center, Ft. Bragg, N.C.
Bostian, William Alexander, Jr. (J'43) (KJZ), Student Engr., Gen. Elec. Co., 970 Western Ave.; *for mail*, 12 Park St., West Lynn, Mass.
Bostic, John A. (J'36) (FKS), 18901 Cherokee Ave., Cleveland, Ohio.
Bostock, Raymond N. ('36), Brewery Supt., P. Ballantine & Sons, 57 Freeman St., Newark, N.J.
Boston, Orlan William ('20; '23; F'44) (MJNC), Prof. Metal Processing, Chmn., Dept. of Metal Processing, Univ. of Mich.; *for mail*, 2021 Vinewood Blvd., Ann Arbor, Mich.
Boswell, William L. ('22; '45) (SOBLC), Asst. Mech. Engr., Ebasco Services, Inc., 2 Rector St., New York, N.Y.; *for mail*, 410 W. Milton Ave., Rahway, N.J.
Bothfield, Robert (J'43) (AEMCJF), Lt. (j.g.), U.S.N.R., Structures Div. Officer, Assembly & Repair Dept., Naval Air Sta.; *for mail*, Bachelor Officers' Quarters, Naval Air Sta., Pensacola, Fla.
Botta, Angelo F. (J'36) (OSNBMK), Mar. Engr., Naval Shipyard, Philadelphia; *for mail*, 4824 Walnut St., Philadelphia 4, Pa.
Botteron, Leonard K. (J'25) (BRS), Engr., Rd. Tests, Union Pac. R.R. Co., 15th & Dodge Sts.; *for mail*, Apt. 9, 128 N. 31st St., Omaha, Neb.
Botticher, Wilhelm Karl (J'37) (LZKDBC), Plant Engr., Anabolic Food Products, Inc., 514 Riverdale Dr., Glendale 4, Calif.
Botts, Earl Albert ('43) (SCLF), Htg. Engr., Cincinnati Gas & Elec. Co., 323 Plum St., Cincinnati 2; *for mail*, 3547 Saybrook Ave., Cincinnati 8, Ohio.
Botwinik, Norman Irving (J'43) (CDONMZ), V.P., Gen. Mgr., Botwinik Bros., Inc., 333 Welton St., Hamden 11, Conn.
Bouchard, Constant Leo (J'42) (AEH), 1060 Winton Ave., Speedway City, Ind.
Bouillon, Lincoln ('43) (0), Partner, Lincoln Bouillon & Associates, 1411 Fourth Ave. Bldg., Seattle 1, Wash.
Bourgarde, Lester Francis (J'43) (KCOBL), Regional Mgr., York Corp., 1238-46 N. 44th St., Philadelphia 4; *for mail*, 1301 Medford Rd., Wynnewood, Pa.
Bourgeois, Claude (J'44), Jr. Prod. Engr., Dominion Containers, 6240 Park Ave., Montreal, Que., Can.
Bourke, Francis Evans (J'36) (NMRAC), Mech. Engr., Hercules Powder Co., Port Ewen; *for mail*, 140 Wall St., Kingston, N.Y.
Bourke, Norman T. ('39) (BES), 451 Washington, Gainesville, Fla.
Bourland, Earl David ('42), Res. Mgr., Eagle-Royal & Globe Indemnity Cos., 1155 Dexter Horton Bldg., Seattle 4, Wash.
Bourne, Robert G. B. (J'37) (CHL), Lt., Hdq. Engrs. Reserve Trng. Center, Weapons Branch 5-3, Ft. Leonard Wood, Mo.; *for mail*, 317 Calvin Rd., Raleigh, N.C.
Bourne, Thos. G. (J'39) (RMS), 171 Coligni Ave., New Rochelle, N. Y.
Bourne, William H. (J'37) (CRS), 60 E. 42nd St., New York 17, N.Y.
Boushell, Clinton Campbell (J'45), Ch. Draftsman, Design Engr., Internatl. Boiler Works Co.; *for mail*, 334 E. Brown St., East Stroudsburg, Pa.
Boutelle, Albert (J'41) (NBAJMO), Serv. Engr., Hamilton Stand. Propellers, East Hartford 8, Conn.; *for mail*, 56 Brownell St., Providence 8, R.I.
Bouton, Edgar Montalbert ('43), Mgr. of Engrg., Westinghouse Elec. Elev. Co., 150 Pacific St., Jersey City, N.J.
Bouton, Geo. I. ('01: '04) (CDM), Sales Engr., D. P. Brown & Co., 516 Howard St.; *for mail*, 2926 Baldwin St., Detroit, Mich.
Boutros, Richard Douglas (J'44), Mech. Engr., Engrg. Dept., Mixing Equip. Co., Inc., 1024 Garson Ave.; *for mail*, 1405 Highland Ave., Rochester 7, N.Y.
Boutwell, Frederick Kent (J'43) (EJL), Lt., U.S.N.R.; Box 545, Durham, N.C.
Bouvier, George Alfred ('36) (CDLMNY), Asst. Dir. of Mfr., Gen. Cable Corp., 26 Washington St., Perth Amboy; *for mail*, 2137 Gamble Rd., Westfield, N.J.
Bovee, John Leroy, Jr. (J'42) (HYNJCM), Dir., Product Devel., Calif. Factory Rep., Red Jacket Mfg. Co., Davenport, Iowa; *for mail*, 500 N. Sabina St., Anaheim, Calif.

Bovin, Henry (J'44) (DZMNOJ), Test Engr. with Walter Dorwin Teague, 444 Madison Ave., New York 22; *for mail,* 1519 E. 19th St., Brooklyn 30, N.Y.
Bowar, Earl C. (J'41) (AZSCDO), Design Mech. Engr., Minneapolis-Honeywell Regulator Co., 2753—4th Ave. S., Minneapolis; *for mail,* 4001 Chowen Ave. S., Minneapolis 10, Minn.
Bowden, Oscar Louis (J'40) (CFKSZ), Prod. Engr., Edgewater Power Plant, Ohio Pub. Serv. Co., Lorain, Ohio.
Bowen, Ernest Whiting ('25; '31; '35) (CLBNKD), Ch. Engr., Nitrogen Div., Solvay Process Co., Hopewell, Va.
Bowen, Frank Milton ('26; '37), Partner, Jensen, Bowen & Farrell, Engrs., 209 Mich. Theatre Bldg., Ann Arbor, Mich.
Bowen, Harold Gardiner (H'45) (AEFJSY), Rear Admiral, U.S.N., Chief, Office of Research & Inventions, Rm. 3449, Main Navy Bldg., Washington 25, D.C.
Bowen, Harry S. ('31; '35) (CDL), V.P., Puget Sound Sheet Metal Works, 3631 E. Marginal Way, Seattle, Wash.
Bowen, Henry Blane ('44), Ch. of M.P. & R.S., Canadian Pacific Ry. Co., Windsor Station; *for mail,* 3018 Breslay Rd., Montreal, Que., Can.
Bowen, John Thomas (J'44) (BNHK), Mech. Engr. (Design), Naval Ord. Test Sta., 1030 E. Green St., Pasadena 1; *for mail,* 551 S. Hill Ave., Pasadena 5, Calif.
Bowen, Percy P. ('31) (W), Pres., Becker, Moore & Co., Inc., North Tonawanda, N.Y.
Bowen, Raymond Henry (J'43) (ACEMO), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 11 N. Ten Broeck St., Scotia, N.Y.
Bowen, Robert Densham (J'42) (CDM), Tool Research, Sr., Quenching Dies, Wright Aero. Corp., Lockland; *for mail,* Box 227, R.R. 10, Kugler Mill Rd., Cincinnati 27, Ohio.
Bowen, William Spencer ('18; '35) (ES), Pres., Bowen Research Corp., North Ave., Garwood, N.J.
Bowen, William Victor ('24; '28; '35) (FSCKHD), Mech. Engr., Pa. Water & Power Co., Lexington Bldg., Baltimore 3, Md.
Bower, Clark D. (J'37) (BHJ), 1618 Gordon Court, Pomona, Calif.
Bower, Edward Illston (J'36) (CDMS), Lt., S (O), U.S.N.R., Naval Ord. Matls. Handling Lab., Naval Am. Depot, Hingham; *for mail,* 1282 Great Plain Ave., Needham 92, Mass.
Bower, George C. (J'39), Lt., 166 Bainbridge St., Brooklyn, N.Y.
Bower, Harvey Settone (J'40) (BJM), Draftsman, Gen. Chem. Co., 1100 Line St., Camden; *for mail,* 131 Church St., Teaneck, N.J.
Bower, Jerome Gordon (A'03) (RABYJ), Buckeye Steel Castings Co., 50 Church St., New York 7, N.Y.
Bower, Raymond G. ('21; '26; '35), Mech. Draftsman, Burroughs Adding Mch. Co., Detroit, Mich.
Bowerman, Myron Ralph ('10; '14; '35) (DBC), Asst. to Ch. Engr., Alliance Mch. Co., Alliance; *for mail,* Homeworth, Ohio.
Bowers, Roscoe Haines ('44) (WLNMCD), Asst. Ch. Engr., Kieckhefer Container Co., Delair; *for mail,* 625 Main St., Riverton, N.J.
Bowes, Elmer Henry (J'42) (KHZC), Mech. Engr.; Dravo Corp., Broad St. Sta. Bldg.; *for mail,* 8009 Albion St., Philadelphia 3, Pa.
Bowes, Robert James (J'43) (SAKB), Engr., A.G.T. Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 548 Darby Rd., Ridley Park, Pa.
Bowes, Thomas David, Jr. (J'35) (EBNFMJ), Mech. Engr., Maint., Sun Oil Co., 1608 Walnut St., Philadelphia; *for mail,* 333 Trevor Lane, Cynwyd, Pa.
Bowie, Harry Joseph (J'43) (HJLMNZ), Asst. Ch. Engr., Fischer & Porter Co., County Line Rd., Hatboro; *for mail,* Orvilla & School Rd., R.F.D. 1, Hatfield, Pa.
Bowlus, B. H. ('41) (ACH), 954 W. 6th St., Erie, Pa.
Bowman, Clarence Bliss (J'38) (DLM), Project Boss, Western-Knapp Engrg. Co., 760 Folsom St., San Francisco; *for mail,* 1891 Catalina Ave., Berkeley 6, Calif.
Bowman, Cletus E. (J'41) (FK), Ensign, U.S.N.R., Cabin John Gardens, 9 Webb Rd., Washington 25, D.C.
Bowman, Frederick Hardy (J'39) (EHS), Mech. Engr., McIntyre Porcupine Mines, Ltd.; *for mail,* Box 485, Schumacher, Ont., Can.
Bowman, Henry T. ('28; '35) (BES), Asst. Prof. Mech. Engrg., Engrg. Bldg., Univ. of Pa., Philadelphia 4, Pa.
Bowman, James Schenck ('26) (HSBFDN), Ch. Water Control Engr., Tenn. Valley Authority, 410 Union Ave., Knoxville, Tenn.
Bowman, John Cowan (J'44) (BAN), Sr. Stress Analyst, Ryan Aero. Co., Lindbergh Field, San Diego; *for mail,* 4045 San Juan Rd., San Diego 3, Calif.
Bowman, Robert Alexander ('31; '39) (BKS), Mgr., Condenser Engrg., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13, Pa.
Bown, Robert Carpenter (J'43), Lt. (j.g.), U.S.N.R., U.S.S. *Jallao* SS-368, F.P.O., San Francisco, Calif.
Box, Robert Edwin (J'43), 217 Cedar Ave., St. Charles, Ill.
Box, William Thomas (J'40) (HWLJC), Sales Engr., Byron Jackson Co., Box 2017, Terminal Annex, Los Angeles 54, Calif.
Boxer, Jack (J'43) (BAM), Assoc. Engr., Turbine & Gear Sec., Bur. of Ships, Navy Dept., Code 641, Washington 25, D.C.
Boyajian, James Aram ('42) (MYJNOK), Partner, Product Engrg. & Mfg. Co., 3709 W. Armitage Ave., Chicago 47; *for mail,* 5760 N. Rogers Ave., Chicago 30, Ill.
Boyajian, Richard D. (J'39) (CLM), 609 S. 52nd St., Philadelphia 42, Pa.
Boyce, Frank George ('21) (EFHS), V.P., Consumers Power Co., 212 W. Michigan Ave., Jackson, Mich.
Boyd, Benjamin Earl ('44) (CTYDEF), Gen. Prod. Mgr., Owens-Corning Fiberglas Corp., Madison Ave., Toledo 1, Ohio.
Boyd, Charles Oliver (J'45), 128 Tyrone Ave., Greenside East, Johannesburg, S. Africa.
Boyd, James Ellsworth ('15) (BH), Emeritus Prof. of Mechanics, Ohio State Univ., N. High St., Columbus 1, Ohio.
Boyd, James William (J'37) (YBMZDC), Plant Engr., Sealright Co., Inc., Fairfax & Rickel Rds., Kansas City, Kan.
Boyd, John Taylor ('87) (RSDJ), Retired; c/o Mrs. Forbes McCreery, 151 E. 83rd St., New York 28, N.Y.
Boyd, Joseph Johnson ('45) (CHODE), Gen. Mgr., Dempsey Constr. Co., Box 3098, Tulsa 8, Okla.
Boyd, Kenneth Ross ('45) (MYCNDG), Maint. Engr., Eastman Kodak Co., Kodak Park; *for mail,* 180 Merrill St., Rochester 13, N.Y.
Boyd, Lee Roy ('44) (BGJMNY), Design Engr., Westinghouse Elec. Corp.; *for mail,* 333 Taylor Rd., Box 70, Mansfield, Ohio.
Boyd, Robert Norman (J'39) (CTL), Project Engr., Nylon Div., Canadian Industries Ltd.; *for mail,* 803 Johnson St., Kingston, Ont., Can.
Boyd, William Wallace ('10; '17; '35) (BCDFJK), Mech. Engr., 79 Glen Ridge Ave., Glen Ridge, N.J.
Boyd, William Wray ('37) (SLE), Field Engr., Flexitallic Gasket Co., 8th & Bailey Sts., Camden, N.J.; *for mail,* 804 S. Knoxville Ave., Tulsa 4, Okla.
Boye, Burton Louis, Jr. (J'43) (A), 316 Ashland Rd., Summit, N.J.
Boyen, John Louis (J'45) (KSFRM), Application Engr., Besler Corp., 4053 Harlan St., Emeryville 8; *for mail,* 4 Via Floreado, Route 2, Orinda, Calif.
Boyer, Earle G., Jr. (J'40) (BHJNYZ), Mech. Designer, Arty. Devel. Sec., Franklin Inst., 20th & Parkway Rd., Philadelphia; *for mail,* 112 E. Mt. Airy Ave., Philadelphia 19, Pa.
Boyer, Edwin D. ('21), Mech. Engrg. Dept., N.Y. Air Brake Co.; *for mail,* 145 Sterling St., Watertown, N.Y.
Boyer, Edwin S. ('97; '02; '07) (CDL), Chmn. of Bd., Am. Hard Rubber Co., 11 Mercer St., New York 13, N.Y.; *for mail,* 1070 Hillside Ave., Plainfield, N.J.
Boyer, Frederick G. L. ('28) (NBGK), Research Engr., Pat. Dept., Champion Paper & Fibre Co., Hamilton, Ohio.
Boyer, Glenn Clinton ('40) (EFKS), Assoc. Engr., Burns & McDonald Engrg. Co., 107 W. Linwood Blvd.; *for mail,* 4133 Oak St., Kansas City 2, Mo.
Boyer, Rodney Leonard (J'42) (ABF), 125 Wiggins St., West Lafayette, Ind.
Boyer, Vincent Saull (J'39) (SZJ), Ensign, U.S.N.R., 6320 Lawnton Ave., Philadelphia 41, Pa.
Boyko, John (J'37) (DNBYMC), Ch. Draftsman, Mech. Handling Systems Inc., 4600 Nancy St., Detroit 12; *for mail,* 10238 Beaconsfield, Detroit 24, Mich.
Boylan, Glen Dennis ('39) (CMY), V.P., Dir. of Mfg., Meredith Publ. Co., 1716 Locust St., Des Moines 3, Iowa.
Boyle, Joseph Cornelius ('40; '45) (BNJ), Sr. Mech. Engr., David Taylor Model Basin, Navy Dept., Washington; *for mail,* 16 Webb Rd., Bethesda Branch, Washington 14, D.C.
Boyle, Joseph Edward (J'43) (JMBD), Mech. Engr., Arch. Steel Engrg. Co., Inc., Amber & Venango Sts., Philadelphia 34; *for mail,* 6418 Lawndale Ave., Philadelphia 11, Pa.
Boyle, William Edward (A'45) (ZHKE), Instrument Engr., Shell Oil Co., Wood River; *for mail,* 821 Hale Ave., Edwardsville, Ill.
Boyles, Robert M. ('23; '30; '33) (COMYLS), Dir., Indus. Relations, Midwest Rubber Reclaiming Co., Box 744, East St. Louis, Ill.; *for mail,* 5851 Waterman Ave., St. Louis 12, Mo.
Boynton, Arthur Jefferson ('22) (JMLDCF), Head, A. J. Boynton & Co., 58 E. Washington St., Chicago 2, Ill.
Boynton, Earl Standish ('42) (YJLACM), Factory Mgr., Pierce Lab., Inc., 2 Walnut St., Summit; *for mail,* 31 Madison Ave., East Orange, N.J.
Boynton, Edgar Bowe (J'37) (SFOLZK), Wiley & Wilson, Inc., 517 American Bldg., Richmond; *for mail,* John Rolfe Apts., 10, Tempsford Lane, Richmond 21, Va.
Boynton, Erwin Reynolds ('44), Hyd. Devel. Engr., Aero. & Mar. Eng. Dept., Gen. Elec. Co.; *for mail,* R.D. 2, Sacandaga Rd., Schenectady, N.Y.
Boynton, John E. ('15; '21), Dir., Dept. Mech. Engrg., Vanderbilt Univ., Nashville, Tenn.
Boynton, W. S. ('17; '35), R.R. 1, Mason, Ohio.
Boynton, Wentworth Devries (J'36) (BJZCYM), Mech. Engr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore; *for mail,* Meadow Rd. & Bellona Ave., Baltimore 12, Md.
Boysen, Jens (J'38) (CMNGBO), Instr., Samuel Gompers Trades Sch., 22nd & Bartlett St., San Francisco 10; *for mail,* 2739 Anza St., San Francisco 21, Calif.
Boyum, William Byron (J'44) (AMNY), 421 Rebecca Ave., Wilkinsburg, Pa.
Bozarth, Harold H. (J'42) (SKEFMJ), Ensign, U.S.N.R., Naval Air Sta., Jacksonville, Fla.
Bozenhard, Cushing Charles (J'45), U.S.N.R., 18 Homewood Ave., Yonkers 2, N.Y.
Braccia, Anthony A. (J'46), Mech. Engr., Textile Research & Devel. Dept., Celanese Corp. of Am.; *for mail,* 107 Bellevue Heights, Cumberland, Md.
Brace, Norman Gourlay ('24; '35) (HFSEKR), Mar. Engr., Bur. of Ships, Navy Dept., Washington 25, D.C.; *for mail,* 719 Richmond Ave., Silver Springs, Md.
Bracegirdle, Joseph, Jr. (J'43) (DMSCL), Maint. Supvr., Charles Lennig & Co., 5000 Richmond St., Philadelphia 37; *for mail,* 1013 Kenwyn St., Philadelphia 24, Pa.
Bracken, Victor Pershing (J'42) (SLEMZ), Process Control Engr., La. Div., Stand. Oil Co. of N.J., Baton Rouge; *for mail,* 1930 Olive St., Baton Rouge 11, La.
Brackett, Newell ('23; '35), V.P., Charge Sales Engrg., Crane Packing Co., Drexel Bldg., Philadelphia, Pa.
Brackin, Richard Francis (J'35) (DHBFAM), Mech. Designer, Philadelphia Div., Yale & Towne Mfg. Co., 4530 Tacony St., Philadelphia, Pa.
Bradbury, Donald (J'39) (BSJZL), Design Engr., Mech. Design Sec., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* 1216 Folsom Ave., Prospect Park, Pa.
Bradbury, Gilbert Venable ('42) (M), Foreman, Gear Lab., Pratt & Whitney Aircraft Div., United Aircraft Corp., Kansas City; *for mail,* 2516 Summit St., Kansas City 8, Mo.
Bradfield, Paul Edgar ('44) (EHLOSY), Partner, F. J. Hearty & Co., 714 W. Olympic Blvd., Los Angeles 15, Calif.
Bradford, John Olaf (J'41) (CMYZ), Lexington, Ill.
Bradley, Charles Byron ('45), Charge Physics Sec., Research Lab., Johns-Manville Corp., Manville, N.J.
Bradley, Earl Halliday ('29; '41) (CMNZ), V.P., Works Mgr., Builders Iron Fdy., 9 Codding St., Providence 1, R.I.; *for mail,* 920 County Rd., Seekonk, Mass.
Bradley, Frank Leo ('20; '27; '30) (SFTDC), Plant Engr., Forstmann Woolen Co., 2 Barbour Ave., Passaic; *for mail,* 355 Meadowbrook Ave., Ridgewood, N.J.
Bradley, Harry L. ('21; '35), Allen-Bradley Co., 136 W. Greenfield Ave., Milwaukee, Wis.
Bradley, James Frederick, Jr. (J'43) (ANHL), 1505 Michigan Ave., Le Porte, Ind.
Bradley, James Harrison ('21; '35) (HJMO), Engr., Holcroft & Co., 6545 Epworth Blvd., Detroit 10, Mich.
Bradley, John Alfred (J'36) (EORS), Mech. Engr., Ross & White Co., Chicago Daily News Bldg., Chicago 6; *for mail,* 2651 Broadway, Evanston, Ill.
Bradley, John Joseph (J'45) (ACP), Apprentice Seaman, U.S.N.; 5818 Crittenden St., Philadelphia 38, Pa.
Bradner, Alton Fancher ('32; '42) (AEHS), Asst. Plant Mgr., Orange County Tel. Co., 19 John St.; *for mail,* 40 Liberty St., Middletown, N.Y.
Bradshaw, George Verlan, Jr. (J'43), Jr. Design Engr., RCA Victor Div., Radio Corp. of Am.; *for mail,* 1552 Ellery Walk, Fairview, Camden, N.J.
Bradshaw, William Thomas (J'46), 96 Brook St., Wellesley, Mass.
Bradt, David M. (J'41), Lt. (j.g.), U.S.N.R., Norfolk Naval Shipyard, Portsmouth, Va.; *for mail,* 620 Liberty St., Erie, Pa.
Bradt, Morris ('19; '35) (SWF), Skinner Eng. Co., Erie, Pa.
Brady, Dirk Samulon (J'43) (ZNMC), Tool Engr., Metrical Labs., Inc.; *for mail,* 1208 W. Madison St., Ann Arbor, Mich.
Brady, James Vincent (J'39) (EFNBSM), Designer, Lanova Corp., 27-01 Bridge Plaza, Long Island City 1; *for mail,* 3980 Provost Ave., Bronx 66, N.Y.
Brady, Leo Joseph (A'45), Exec. V.P., Waucantuck Mills, Uxbridge, Mass.; *for mail,* Christie Hill Rd., Darien, Conn.
Brady, Sherwood Joshua (J'42) (EBKS), 2nd Lt., Ord., Cent. Flying Trng. Comd., Texas; *for mail,* Box 235, Garden Grove, Calif.
Bragaw, Chester Griswold, Jr. (J'46) (KOS), 71 Lewis Rd., Belmont, Mass.
Bragdon, George D. ('46) (BJS), Retired; Box 1090, Orlando, Fla.
Bragg, David Kendall ('32; '42) (ZLHSKB), Research Engr., Foxboro Co., Neponset Ave., Foxboro, Mass.
Bragg, Francis Coolidge ('38) (BNSEHJ), Assoc. Prof. Drawing, Engrg. Drawing & Mech. Dept., Ga. Sch. of Tech., Atlanta, Ga.
Bragg, James Harold (J'44) (ABS), 2nd Lt., Hdq., 78th Div. Arty., A.P.O. 78, c/o Postmaster, New York, N.Y.
Bragg, Sidney Jaquith (J'44) (BCE), Engr., Mack Mfg. Corp., 1120 S. 2nd St., Plainfield, N.J.

Brainard, Boyd B. ('25; '34) (FS), Prof. Mech. Engrg., Mech. Engrg. Dept., Kan. State College, Manhattan, Kan.
Braine, Bancroft Gherardi ('96; '05), Retired; 19 N. Mountain Ave., Montclair, N.J.
Braithwaite, Willis Martin (J'45) (BCE), Jr. Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 58 Union St., Montclair, N.J.
Brakeman, Roy Edgar ('11) (BCG), Ch. Engr., So. Dist., Republic Steel Corp.; *for mail,* 629 Turrentine Ave., Gadsden, Ala.
Bralove, William, Jr. (J'41) (BEH), 164 R.F.D. 3, Rockville, Md.
Bramhall, George Hardy (J'45) (MJNG), Engr., Engrg. Gen. Dept., Gen. Elec. Co., Schenectady; *for mail,* 20 Front St., Schenectady 5, N.Y.
Brammer, Robert Carl (J'45) (AHBNS), Research Analyst, Lockheed Aircraft Corp.; *for mail,* 414 E. Valencia Ave., Burbank, Calif.
Branch, Arthur M. ('25), Cons. Engr., Rm. 204, 319 N. 4th St., St. Louis, Mo.
Brand, Frederick Fermor ('37) (BCJLMN), Works Engr., Gen. Elec. Co., 100 Woodlawn Ave., Pittsfield, Mass.
Brand, Fred'k. P. ('39) (SFBEJL), V.P., Ch. Engr., Internatl. Combustion Corp., 436 Pearl St., New York 7, N.Y.
Brand, George B. ('12) (CDM), 8078 Selma Ave., Hollywood 46, Calif.
Brand, Horace H. ('36), Ch. Mech. Engr., Liberian Project, Raymond Concrete Pile Co., 140 Cedar St., New York, N.Y.; *for mail,* 62 Yantecaw Ave., Bloomfield, N.J.
Brand, Ronald Scott (J'43) (BAJ), 30 Baker Ave., Beverly, Mass.
Brandenburg, Stanley Allenn (J'35) (BCM), Sales Mgr., Monarch Mch. Tool Co., Sidney, Ohio.
Brandes, C. H. ('39) (FL), East. & Foreign Mgr., West. Precipitation Corp., 405 Lexington Ave., New York, N.Y.
Brandon, Daniel Morris (J'43) (AONBCZ), Design Engr., Westinghouse Elec. Corp., Lester; *for mail,* 508 Holmes Rd., Ridley Farms, Morton, Pa.
Brandon, J. Campbell, Jr. (J'42) (CMDJA), Devel. Engr., Heating Research Co., 219 E. Cunningham St.; *for mail,* 608 N. McKean St., Butler, Pa.
Brandon, Raymond John ('41) (SF), Fuel Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Brandt, Arnold Edward (J'44) (NWHOJR), Arty. Instr., Ord. Sch., Box 128, Aberdeen Proving Ground, Md.; *for mail,* 6330 Cartridge, Detroit 9, Mich.
Brandt, Ernst Hamilton, Jr. ('21; '26; '35) (KSEFAR), Pres., Reliance Engrg. Co., Inc., Box 1292, Charlotte 1, N.C.
Brandt, Frank C. ('27; '33), 538 Parker Ave., Barberton, Ohio.
Brandt, Hugh B. ('24; '30; '35) (CLSFEK), Procter & Gamble Co., M. A. & R. Bldg., Ivorydale, Cincinnati 17, Ohio.
Branigan, George Francis ('44) (BGH), Dean of Engrg., Bradley Poly. Inst., Peoria, Ill.
Branin, Francis Sunfield ('44) (CJYMOL), V.P., Gen. Mgr., Middlesex Mfg. Co., Inc., Lincoln Blvd., Middlesex; *for mail,* 211 Farragut Rd., North Plainfield, N.J.
Brannan, John Clark (J'44) (SFLDJ), Lt., U.S.N.R.; *for mail,* 765—13th St., Boulder, Colo.
Bransford, Carroll Keister (J'42) (ASFKN), Lt., U.S.N.R., Oper. Instructions Officer, Power Plant Design Branch, Bur. of Aero., Navy Dept., 2W45, Navy Bldg., Washington 25; *for mail,* 5728—3rd St., N.W., Washington 11, D.C.
Brasch, John ('38) (CMJHL), Sr. Engr. (Mech.), U.S. Engr. Office, War Dept., 751 S. Figueroa St., Los Angeles 14, Calif.
Brashear, William Mercer (J'36) (CLMDKB), Asst. Plant Engr., Colgate-Palmolive-Peet Co., 6th & Carlton Sts., Berkeley; *for mail,* 1145 Grizzly Peak Blvd., Berkeley 8, Calif.
Braski, Neil John ('44) (JKS), Power Engr., Diamond Alkali Co.; *for mail,* 875 North State St., Painesville, Ohio.
Braswell, Charles Clarence (J'45) (KCMLDY), Shift Supvr., Casting Area, Photo Products, E. I. du Pont de Nemours & Co.; *for mail,* Box 125, Parlin, N.J.
Bratt, Ralph Turner ('37) (FS), 150 Woodland St., Worcester 3, Mass.
Brattin, Claud Lamar ('38) (GNAESR), Prof. & Head, Engrg. Drawing Dept., Mich. State College; *for mail,* 220 Charles St., East Lansing, Mich.
Brauburger, Robert Arthur (J'45) (ODCZKS), Lab. Engr., Chrysler Corp., Oakland Ave., Detroit 31; *for mail,* 257 Colorado Ave., Detroit 3, Mich.
Braugart, George, Jr. ('22) (EHKNS), Pres., Ch. Engr., Engrg. Contractors, Inc., 611 Bona Allen Bldg., Atlanta 3, Ga.
Braumiller, Richard Earl (J'45) (SRCEKO), Lt. (j.g.), U.S.N., Ch. Engr., U.S.S. *Sigsbee,* DD-502, Naval Base, Charleston, S.C.; *for mail,* 2127—12th St., Moline, Ill.
Braun, John Bernard (J'34) (JCDM), Dist. Maint. Supvr., Otis Elevator Co., 2301 Locust St., St. Louis 3, Mo.
Braun, John Ludwig ('40) (ODLNCA), Engrg. Draftsman, Gen. Foods Corp., 250 Park Ave., New York, N.Y.; *for mail,* 263 Greenwich Ave., Stamford, Conn.
Braun, Richard (J'44) (NBJ), 148-47—88th Ave., Jamaica 2, N.Y.
Braun, Robert Herman (J'45) (AHY), 109 Jordan Ave., San Francisco 18, Calif.
Braun, Robert Leopold (J'43) (AMW), 905 Melrose Ave., Melrose Park, Pa.
Brauninger, Glen Gerald (J'36) (ACMSJE), Lt., U.S.N.R., Assembly & Repair Dept., Naval Air Sta., Norfolk 11, Va.
Braunschneider, Frederick John (J'44) (BEJ), Lt. (j.g.), U.S.N.R., Naval Mine Warfare Test Sta., Solomons, Md.
Braverman, Joseph H. (J'38) (KHO), Mech. Engr., Armo Cooling & Ventilating Co., 30 W. 15th St., New York; *for mail,* 867 Forest Ave., Staten Island 10, N.Y.
Bravo, Carlos Luis ('41) (JLM), Ch. Engr., Mayaguez Sugar Co. Inc., Mayaguez, P.R.
Bravo, David Salvador, Jr. (J'44) (AKMC), Tool Engr., Leslie Co., Grant Ave., Lyndhurst; *for mail,* 427 Riverside Ave., Rutherford, N.J.
Bravo, Oscar F. (A'29) (CDF), Administrator, Gen. Mgr., Mayaguez Sugar Co. Inc., P.O. Box 569, Mayaguez, P.R.
Bray, Charles William ('88), Retired; Exeter Rd., Haverford Township, Pa.
Breaker, Ernest Robert ('25) (CRY), 1406 Windsor Rd., Austin, Tex.
Brecht, David Cedric (J'35) (MBAN), Mech. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 712 Riverside Ave., Scotia 2, N.Y.
Brecht, Winston Allen ('43) (RBNCJF), Mgr., Trans. Engrg. Dept., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 231 Cascade Rd., Pittsburgh 21, Pa.
Breckenridge, Clarence Edward ('04; '16) (COWY), Asst. Gen. Mgr., Am. Express Co., 65 Broadway, New York, N.Y.; *for mail,* 114 Mill St., Westwood, N.J.
Breckenridge, Andrew Lyndon ('13; '24; '35) (CDOS), Supvg. Engr., Reconstr. Finance Corp., 58 Weybosset St., Providence 3, R.I.; *for mail,* 449 Edgewood Ave., New Haven 11, Conn.
Breckenridge, Frank ('43) (BCT), Ch. Engr., Home Appliance Div., Fed. Tel. & Radio Corp., 200 Mt. Pleasant Ave., Newark 4; *for mail,* 358 Churchill Rd., West Englewood, Teaneck, N.J.
Breckenridge, Robert William (A'42) (JLM), Asst. Prof., Mech. Engrg. Dept., Iowa State College, Ames, Iowa.
Breda, Thoralf Kristian ('29; '31; '35) (HBMNCO), Asst. Ch. Design Engr., S. Morgan Smith Co., Lincoln St.; *for mail,* 672 Florida St., York, Pa.
Bredlau, Albert Ernest, Jr. (J'42) (BCMNHS), Project Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 730 Union St., Schenectady, N.Y.
Breece, Grady Lee ('45) (BHS), Ch. Mech. Engr., McWilliams Dredging Co., 860 St. Charles St., New Orleans 13; *for mail,* 3105 Metairie Rd., New Orleans 20, La.
Breed, Everett M. ('20) (CHM), Charge Order & Contract Div., Sales Dept., Joshua Hendy Iron Works, Sunnyvale; *for mail,* 507 Cornell Ave., San Mateo, Calif.
Breeden, George Blunt (J'43) (YLJ), Lt. (j.g.), U.S.N.R., Student, Postgraduate Sch., U.S. Naval Acad., Annapolis, Md.
Breen, Edward M. (J'37) (CKS), Asst. Power Engr., Bethlehem Steel Co., Lebanon; *for mail,* 30 N. 13th St., Easton, Pa.
Breen, John Neuman (J'44), Ensign, Civil Engr. Corps, U.S.N.R., Pub. Works Office, 14th Naval Dist., Navy 128, c/o F.P.O., San Francisco, Calif.
Breen, Patrick James ('31; '35) (JBM), Mar. Inspr., Stand. Oil Co. of N.J., 30 Rockefeller Plaza, New York, N.Y.; *for mail,* 1573 Glen Ave., Folcroft, Pa.
Breer, Carl ('12; '25) (CEARB), Dir. of Research, Chrysler Corp., 341 Massachusetts Ave., Highland Park; or Box 1919, Detroit 31; *for mail,* 15600 Windmill Pointe Dr., Grosse Pointe 30, Mich.
Breffeilh, George A. (J'37) (BHP), 936 Jefferson Pl., Shreveport, La.
Bregler, Wilfred Andrew Campbell (J'36), Engr. & Expediter, Hendry Corp., Rattlesnake; *for mail,* 3808 Lynwood Ave., Tampa, Fla.
Bregman, Irvin S. (J'42) (NMJYCF), Technician, 3rd Grade, A.U.S., Spec. Engrg. Det. (assigned to Carbide & Carbon Chemicals Corp.), Bks. Area, Oak Ridge, Tenn.
Brehm, William W. ('28; '34; '35) (ALS), Engrg. Draftsman, Wilson & Co., 223 S. Santa Fe Ave., Salina; *for mail,* 1704 Carey Blvd., Hutchinson, Kan.
Breidenbach, Paul H. (J'34), 1st Lt., U.S.A., 115th Ord. Co. (MM), A.P.O. 689, c/o Postmaster, New York, N.Y.; *for mail,* 811 N. Main St., Kenton, Ohio.
Breidenstein, Leonard William (J'32) (LSEC), c/o Hercules Powder Co., Box 210, Cumberland, Md.
Breitenstein, Albert Frank ('19; '21) (BJM), Ch. Engr., Geometric Tool Co.; *for mail,* 10 Alden Ave., New Haven, Conn.
Breithaupt, John Theodore (J'42) (CDH), Jr. Mech. Engr., Shell Oil Co., Inc., Box 398, Eagle Lake, Tex.
Breitman, Milton (J'40) (ABM), 3409 Jerome Ave., Bronx, N.Y.
Bremmer, John Livingston ('46) (KHC), Mech. Engr., Post Engrs., Camp Crowder, Mo.
Brendlin, Herman John ('40) (DBJ), Ch. Engr., Hayward Co., 50 Church St., New York 7; *for mail,* 37-04 Bowne St., Flushing, N.Y.
Brendlinger, William Brown (A'07) (SHCERK), Dist. Mgr., Sales, Ingersoll-Rand Co., 1600 Arch St., Philadelphia 3, Pa.
Breneman, Louis Aloysius ('27) (OSEMLN), Ch. Tool Engr., Minneapolis-Moline Power Implement Co., 29th & Minnehaha Ave., Minneapolis; *for mail,* 1787 Jefferson Ave., St. Paul 5, Minn.
Brengel, Frederick J. ('40) (BC), 1131 Salem Ave., Hillside, N.J.
Brengelman, George Dury (J'37) (CJ), Asst. M.M., Mills Dept., Tenn. Coal, Iron & R.R. Co., Ensley 8; *for mail,* 3081 Sterling Rd., Birmingham 5, Ala.
Brennan, James Earl ('30; '35) (BHCNML), Charge Design, Cincinnati Milling Mch. Co., 4701 Marburg Ave., Cincinnati 9; *for mail,* 7171 Cincinnati West Union Rd., Cincinnati 30, Ohio.
Brennan, James Ivan ('11; '17) (ACFOS), Advis. Engr., Pub. Works Direction, 57 University Pl., Athens, Greece; *for mail,* 1600 Villanova Rd., East Liberty, Pittsburgh, Pa.
Brennan, John William ('37) (CHSLBN), Mgr., Hyd. Coupling Div., Am. Blower Corp., 8111 Tireman Ave., Detroit 32, Mich.
Brennan, M. G. ('29; '35) (CDEFMS), V.P., Stearns-Roger Mfg. Co., 1720 California St., Denver 2; *for mail,* 18 Crestmoor Rd., Denver 7, Colo.
Brennan, Wm. E. ('23; '26; '35), Inspr., Indus. Power, Detroit Edison Co., 2000—2nd Ave.; *for mail,* 4387 Pingree Ave., Detroit, Mich.
Brennan, William P., Jr. (J'42), Box 483, Crystal City, Tex.
Brenneke, Herman J. (J'43) (MJBDCN), Asst. Prof. Engrg. Shop Practice, College of Engrg., N.Y. Univ., University Heights, New York 53, N.Y.
Brenner, Fred G. (J'40) (BCL), 104 Greenspring Manor Apts., Baltimore 8, Md.
Brenner, Kenneth W. ('30; '45) (LNKDBW), Sr. Asst. Mech. Engr., Kodak Park Works, Eastman Kodak Co., Rochester 4; *for mail,* 97 Belmeade Rd., Rochester 5, N.Y.
Brentzel, Reese Jr. (J'41) (ABJMYD), Layout Draftsman, Landing Gear, Ft. Worth Div., Consltd. Vultee Aircraft Corp., Ft. Worth; *for mail,* 3102 Bristol Rd., Ft. Worth 7, Tex.
Brenzinger, Julius ('15) (CKMTY), Gen. Mgr., Max Ams Mch. Co., Bridgeport, Conn.
Brescka, Rudolph S. ('26; '36) (CJM), Engr., Charge Mch. Tool Procurement, West. Elec. Co., Inc., 100 Central Ave., Kearny, N.J.
Breslove, Joseph ('06; '13) (SOE), Cons. Engr., Oliver Bldg., Pittsburgh 22, Pa.; Leader Bldg., Cleveland 14; *for mail,* 2946 Carlton Rd., Cleveland 22, Ohio.
Breslove, Joseph, Jr. (J'37) (SABFED), Cons. Engr., Oliver Bldg., Pittsburgh 22, Pa.
Brett, James Q. (J'38) (CDM), 57 Sutter St., San Francisco 4, Calif.
Bretz, Frank E., Jr. (J'42) (CMR), Ch. Engr., Ajax-Consltd. Co., 4615 W. 20th St., Chicago 50; *for mail,* 710 S. Highland Ave., Oak Park, Ill.
Bretzlaff, George Albert (J'41) (JMCDAF), Engr. Matl. Specifications, Allison Div., Gen. Motors Corp., Box 894, Indianapolis 6; *for mail,* 6002 Broadway Ave., Indianapolis 5, Ind.
Breunich, Paul Edward ('26; '40) (SKFG), Mech. Engr., Tech. Analyst, Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 616 E. Lincoln Ave., Mt. Vernon, N.Y.
Breunich, Theodore Roosevelt (J'41) (SBK), Sonntag Scientific Corp., 1 Seneca Pl., Greenwich; *for mail,* 32 Culloden Rd., Stamford, Conn.
Breverman, Harry (J'39) (MABJCL), 847 N. 7th Ave., Phoenix, Ariz.
Brevoort, Frank LeRoy (J'41) (YL), Mech. Engr., E. I. du Pont de Nemours & Co., Box 993, Charleston 24; *for mail,* 2518-B Kanawha Blvd. E., Charleston 1, W. Va.
Brevoort, Howard William (J'37) (NKHJ), Design Engr., John Robertson Co., Inc., 133 Water St., Brooklyn 1; *for mail,* 157 Carlton Terrace, Stewart Manor, L.I., N.Y.
Brewer, Alexander Van ('45) (EKS), Prof., Mech. Engrg. Dept., Agric. & Mech. College of Tex., College Station, Tex.
Brewer, Allen Frank ('45), Technologist, Tech. & Research Div., Texas Co., 135 E. 42nd St., New York 17, N.Y.; *for mail,* 314—2nd Ave., Avon-by-the-Sea, N.J.
Brewer, Arthur ('06) (DJL), Asst. Works Mgr., Bridgeport Brass Co., 30 Grand St.; *for mail,* 100 Unquowa Hill, Bridgeport, Conn.
Brewer, Nathaniel ('43) (ZBNLH), Ch. Engr., Fischer & Porter Co., Hatboro; *for mail,* R.D. 2, Newtown, Pa.
Brewster, Ellis W. ('23; '35) (CT), Treas., Gen. Mgr., Plymouth Cordage Co., North Plymouth, Mass.
Brewster, John Irvin (J'39) (KLEZ), Engr., C. F. Braun & Co., 1000 S. Fremont Ave.; *for mail,* 1139 N. Monterey St., Alhambra, Calif.
Brewster, John Thomas (J'38), Serv. Unit Supvr. (Dist. Engr.), Richmond Exploration Co., Apartado 93, Maracaibo, Venezuela, S.A.
Brewster, Wayne Maurice (J'42) (CJM), Woonsocket, S.D.
Brezina, Joseph J. (J'39) (EKS), 34 Joice St., San Francisco, Calif.
Brice, Norman E. ('18; '35) (MOJHKL), Box 14, Millburn, N.J.
Brick, Gordon S. (J'37) (ABOHNK), Aero. Engr., Naval Aircraft Factory, Philadelphia 12; *for mail,* 2122 Spruce St., Philadelphia 3, Pa.
Brickman, Arthur Donald (J'42) (BJM), Lt., U.S.N.R., 419 Clinton Ave., Brooklyn 5, N.Y.

Bridge, Lawrence R. ('38) (FHKNSY), Mech. Engr., TSEPP-10, Power Plant Lab., Wright Field, Dayton, Ohio.
Bridge, Theodore Emerson (J'41) (BSCKLZ), Power Engr., E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington 98, Del.
Bridges, Luther W. ('15) (DKS), Mech. Engr., Design, Constr., Power Plants, Gas Works; *for mail,* 273 Union Ave., Framingham, Mass.
Bridges, Robert McSteen (J'41) (AHEBM), Design Engr., Lockheed Aircraft Co., Burbank; *for mail,* 7723 Tujunga Ave., North Hollywood, Calif.
Bridgman, Robert Rice ('25; '34; '35) (ERFZH), Plant Engr., Sterling Eng. Co., 1270 Niagara St., Buffalo 13; *for mail,* R.F.D. 4, 190 S. Creek Rd., Hamburg, N.Y.
Bridwell, Samuel Lee (J'46), Ensign, U.S.N.; Box 18, Route 1, Pulaski, Pa.
Brie, Emile Hardtwig ('44) (HSNB), Chief, Mech. & Elec. Engrg., Norfolk Dist. Office, U.S. Engrs., Ft. Norfolk 1; *for mail,* 417 Beck St., Norfolk, Va.
Briggs, Chas. B., Jr. (J'41), Mech. Engr., Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15; *for mail,* 2609 W. 69th St., Mission, Kan.
Briggs, Fred, Jr. (J'37) (DJP), Lt., U.S.N.R., A.E.O.B., Navy 131, c/o F.P.O., San Francisco, Calif.
Briggs, John Ogden (J'43) (BJNMH), Design Engr., Kane & Roach, Inc., 100 Holland Pl., Syracuse; *for mail,* 1049 Ackerman Ave., Syracuse 10, N.Y.
Briggs, William Cyrus, Jr. ('34; '35) (CDM), Supt., Social Security Gp., Fed. Works Agency, Pub. Bldgs. Admin., Office Bldgs. Mgmt., Interior Bldg., Washington, D.C.; *for mail,* 710 N. Overlook Dr., Alexandria, Va.
Briggs, William S. (J'40), 516 Russell Courts, Gilmer Homes, Orange, Tex.
Brigham, Ward Edwin, Jr. (J'39) (ABK), Asst. Project Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 113 Oakwood Dr., Packanack Lakes, N.J.
Bright, Edwin Winslow (J'42) (CMLYWT), N.Y. Trust Co., 100 Broadway, New York; *for mail,* 125 E. 84th St., New York 28, N.Y.
Bright, G. Frank (J'41) (BES), 2305 Chester St., Little Rock, Ark.
Bright, Jas. R. (J'39) (ACS), *Charles T. Main Award,* '39; Asst. Editor, *Product Engineering,* 330 W. 42nd St., New York 18, N.Y.
Brightly, Frederick Charles, Jr. ('42) (LJ), V.P., Stand. Galvanizing Co., 2619 W. Van Buren St., Chicago 12; *for mail,* 734 N. Harvey Ave., Oak Park, Ill.
Brilhart, Gladden Loats (J'44) (CKS), 707 Walnut Ave., Baltimore 29, Md.
Brill, George M. ('91; '96; F'36) (LRS), *Manager,* '04-'07, *Vice-President,* '10-'12; Retired; *for mail,* 19 Kingston Ave., Poughkeepsie, N.Y.
Brill, Harold Allan (J'45) (CMNTD), Partner, Prod. Supvr., Gen. Mgr., Brill Corset Co., 248 N. 11th St., Philadelphia 7, Pa.
Brill, James Bayard ('41) (ZYLAWD), Owner, Natl. Engrg. Co., Box 1475, Indianapolis 6; *for mail,* 55 S. Linwood Ave., Indianapolis 1, Ind.
Brillhart, Samuel Edward ('23; '31), (YBLNJK), Supt., Mfg. Engrg., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore; *for mail,* Lincoln Ave., Lutherville, Md.
Brindel, Harold F. ('25) (CFM), Supt., Gas & Gasoline Div., Gulf Oil Corp., Box 661, Tulsa 2, Okla.
Brindle, G. Ralph (J'35) (EFP), c/o Natural Gas Pipeline Co. of Am., Joliet, Ill.
Bringhurst, G. Kendrick ('21; '27) (OLSDFM), Indus. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 89 Larchmont St., Waban 68, Mass.
Brinig, Frank George ('37) (CSFK), Pres., Gen. Mgr., Erie City Iron Works, 1422 East Ave., Erie, Pa.
Brink, William E. ('38; '44) (CAMB), Mech. Engr., Austin Co., Dexter Horton Bldg.; *for mail,* 2436—29th St. W., Seattle, Wash.
Brinkman, Charles Frederick (J'43) (CJLMOA), Owner, C. F. Brinkman, Cons. Engrs., 70 Lindsley Ave., Newark 6, N.J.
Brinley, C. Coapes ('37), Sales Engr., Lamson Corp., 211 Congress St., Boston, Mass.; *for mail,* 1812 K St., N.W., Washington, D.C.
Brinley, Charles Edward ('13; F'45) (MCR), Chmn. of Bd., Baldwin Loco. Works, Paschall P.O., Philadelphia 42, Pa.
Brinton, Willard Cope ('07; '12) (CORJM), Prin., Brinton Associates, 36 W. 59th St., New York 19, N.Y.
Briscoe, Conway Brown ('33; '39) (CEFS), Mech. Engr., Monsanto Chem. Co., 1700 S. 2nd St., St. Louis 4; *for mail,* 903 Bellerive Blvd., St. Louis 11, Mo.
Briscoe, Ralph ('29) (BCFS), Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Bristol, Edward Sherman ('20; '25; '35) (SZF), Sec. Head, Engrg. Dept., Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44, Pa.
Bristol, Howard H. ('13; '25), Pres., Bristol Co., Waterbury, Conn.
Bristol, Raymond Winship ('19; '35) (CZ), Exec. V.P., Radio Inventions, Inc., 155 Perry St., New York 14, N.Y.
Britt, William H. (J'36) (CBDHMJ), Research & Devel. Engr., Columbus-McKinnon Chain Corp., Fremont St.; *for mail,* 127 Dexter St., Tonawanda, N.Y.
Brittain, John Randolph ('22) (RYB), Mech. Engr., Los Angeles Ry. Corp., 401 E. 54th St., Los Angeles 11; *for mail,* 4130 Edgehill Dr., Los Angeles 43, Calif.
Britten, Clarence Raymond ('29), Secy., Monroe Calculating Mch. Co., 555 Mitchell St., Orange, N.J.
Britton, William Miller ('42), Owner, Wm. M. Britton Co., 8610 Graham Ave., Los Angeles, Calif.
Brizzolara, Ralph D. ('27; '30) (JHRSYC), V.P., Am. Steel Fdys., 410 N. Michigan Ave., Chicago 11, Ill.
Brkich, Alexander (J'45) (HBSAGC), Engr., Pump-Designer, Ingersoll-Rand Co.; *for mail,* 460 Heckman St., Phillipsburg, N.J.
Broas, Richard Fowler (J'40) (SJKF), Estimating Group Leader, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 6 Welshman Court, Caldwell, N.J.
Brobeck, William Morrison (J'31) (SKRZ), Asst. Dir., Radiation Lab., Univ. of Calif.; *for mail,* 2528 Warring St., Berkeley, Calif.
Broberg, Orrin Robert ('36; '44) (HCANMR), Mgr. & Ch. Engr., Mar. Equip. Div., Ellinwood Industries, 150 W. Slauson Ave., Los Angeles 3; *for mail,* 433 N. Griffith Park Dr., Burbank, Calif.
Brock, Clarence A. ('13; '20; '26) (AKS), Office of Base QM., Army Air Base, Camp Springs Army Air Field, Washington 20, D.C.
Brock, Edward Jacob ('43) (SBCFNM), 4285 Olive St., St. Louis 8, Mo.
Brock, Frank Carter (J'41) (AES), Point View Pl., Mountain Lakes, N.J.
Brock, John Edison (J'38) (B), Ensign, U.S.N.R., c/o Mrs. John E. Brock, 758 S. Harrison St., Frankfort, Ind.
Brock, Raymond Carl ('35) (KTSC), Air Conditioning Control Engr., N. Am. Rayon Corp.; *for mail,* 114 Cottage Ave., Elizabethton, Tenn.
Brockel, William E. (J'30) (CKNS), Ch. Engr., Heat Exchanger Div., Lummus Co., 420 Lexington Ave., New York 17, N.Y.; *for mail,* 28—20th Ave., Irvington 11, N.J.
Brockett, Frank Howard, Jr. (J'42) (MC), Radio Technician, 2nd Class, U.S.N.; 55 Irving St., New Haven 11, Conn.
Brockhouse, Jack Wentworth (J'45) (BHY), Ensign, U.S.N.R., PC-1229, F.P.O., San Francisco; *for mail,* 77 Birch St., Redwood City, Calif.
Brocklebank, Arthur Pierson ('12) (HSL), Ch. Engr., Cameron Pump Div., Ingersoll-Rand Co., Phillipsburg, N.J.
Brockschmidt, Henry Frederick (J'42) (MCJYTN), Prod. Engr., Barber-Colman Co., Loomis St.; *for mail,* 120 S. Prospect St., Rockford, Ill.
Brockway, Willard Waldo (J'41), 845—15th St., Boulder, Colo.
Brockwell, Lloyd A. (J'41) (WMG), Design Engr., Kimberly-Clark Corp., Neenah; *for mail,* 1508 N. Division, Appleton, Wis.
Broden, Edwin H. ('12) (CLNMBA), R.F.D. 2, Dover, Del.
Broden, Edwin Rauch ('44) (CEFMLR), Div. Mgr., Spec. Ord. Div., Blaw-Knox Co., R.D. 5, York, Pa.
Broderick, Edwin L. ('41) (KFBOZR), Application Engr., Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1; *for mail,* 92 Patterson Village Dr., Dayton 9, Ohio.
Broderick, Robert Ernest (J'28) (WMCRS), Secy., Treas., Northeastern Lumber Mfrs. Assn., 271 Madison Ave., New York 16, N.Y.
Broders, Claude Owen (J'41) (BAN), Designer, Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 Main St., East Hartford 8; *for mail,* 84 Tremont St., Hartford 5, Conn.
Brodin, Eric Carl ('40) (CMN), Factory Mgr., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia; *for mail,* 209 Holmecrest Rd., Jenkintown, Pa.
Brodsky, Stanley Martin (J'44) (MWJEF), Huntleigh Co., 738 Congress St.; *for mail,* 858 Union St., Schenectady, N.Y.
Broeker, Fred'k G. ('17; '35) (BDM), Engr., Charge Shops & Fdys., Braden Copper Co., Rancagua, Chile, S.A.
Broeze, J. J. ('38), Asiatic Petroleum Corp., (Attention: Miss M. McDonald), 50 W. 50th St., New York 20, N.Y.
Brogan, Joseph E. (J'40), Bldg. Div., Ill. Bell Tel. Co., 208 W. Washington St., Chicago; *for mail,* 30 W. Chicago, Chicago 11, Ill.
Broghamer, Edw. L. (J'38) (BES), Capt., 4863—2nd Ave., Detroit, Mich.
Brohl, Harry T. ('26; '35) (FKS), Steam Supt., Engrg. & Serv. Dept., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.; *for mail,* 4 Mt. Vernon Rd., Upper Montclair, N.J.
Broker, William Frederick, Jr. (J'43) (SNBA), 1st Lt., Army Air Forces, Troop Carrier Comd.; *for mail,* Colon, Mich.
Brombacher, Max H. C. ('21) (CER), Box 423, Brevard, N.C.
Bromberg, Robert (J'43) (KZA), Jr. Engr., Dept. of Mech. Engrg., Univ. of Calif.; *for mail,* 2632 Channing Way, Berkeley 4, Calif.
Bromberger, David (J'45), Lt., Constr. Div., Office of Ch. Engr., USFET, A.P.O. 757, c/o Postmaster, New York, N.Y.
Bronk, Arthur H. (J'39) (AE), Box 52, Lake Hopatcong, N.J.
Bronson, Carlos Emerson ('15; 22) (BEFKS), Ch. Mech. Engr., Kewanee Boiler Corp., Kewanee, Ill.
Brook, Victor ('37), 171 Rockingham St., Rochester, N.Y.
Brooke, Minott ('35) (FSREKL), Fuel Serv. Engr., Chesapeake & Ohio Ry. Co., Box 1800, Huntington 19; *for mail,* 105 Fairfax Court, Huntington 1, W.Va.
Brookman, H. E. ('41), 404 Cottage Ave., Vermillion, S.D.
Brooks, Charles Converse ('14) (DF), Retired; 6430 S. Kimbark Ave., Chicago 37, Ill.
Brooks, Charles P. ('41) (RJFKC), Mech. Engr., Erie R.R., 101 Prospect St., Cleveland 15; *for mail,* 14415 Delaware Ave., Lakewood, Ohio.
Brooks, Charles Welks (J'35) (SMEC), Brookview Farm, Moscow, Pa.
Brooks, Eugene Adolphus ('18; '25) (SZFKW), Propr., Boiler Equip. Serv. Co., 686 Greenwood Ave., N.E., Atlanta, Ga.
Brooks, F. Warren (J'26) (CSEGF), Supt. of Rates & Spec. Studies, Cleveland Elec. Illum. Co., 75 Public Sq., Cleveland 1, Ohio.
Brooks, Frederick A. ('18; '25; '29) (KBHE), Agric. Engr., Agric. Exper. Sta., Univ. of Calif.; *for mail,* Box 334, Route 1, Davis, Calif.
Brooks, Frederick T. (J'36) (FLS), Ch. Petty Officer; 381 Bristol St., Brooklyn, N.Y.
Brooks, Henry W. ('18; '20) (CFS), Cons. Engr., 1300 McPherson Blvd., Fremont, Ohio.
Brooks, J. Ansel ('07; '11) (CDM), Box 587, Madison, Conn.
Brooks, James Guyer (J'36) (POLFS), V.P., Dist. Mgr., Cleaver-Brooks Co., 50 Church St., New York 7, N.Y.
Brooks, Leo Stanley ('19; '35) (CLMNO), M.M., Res. Engr., Sinclair Refining Co., 3400 Kansas Ave., Kansas City 19, Kan.
Brooks, Louis Edward (J'41) (LCMD), Ch. Mech. Engr., Boyle-Midway, Inc., 22 E. 40th St., New York, N.Y.
Brooks, Morgan ('85; F'38) (SEH), 2311 Connecticut Ave., Washington 8, D.C.
Brooks, Moses Edward ('27; '31) (CNMBDO), Ch. Works Engr., Aluminum Co. of Am.; *for mail,* 30 Forest Pl., Massena, N.Y.
Brooks, Samuel Austin ('29) (ABCDF), Engr., Treas., United Conveyor Corp., 37 W. Van Buren St., Chicago, Ill.
Brooks, Tillman Chester ('44) (CGMD), Mgmt. Engr., Barrington Associates, Inc., 52 Vanderbilt Ave., New York 17, N.Y.
Brooks, William Smallwood (J'44) (SEBNAK), Design Engr., DeLaval Steam Turbine Co., Trenton 2; *for mail,* 400 Highland Terrace, Orange, N.J.
Brootzkoos, Sergius D. ('21; '35) (BJLM), Allied Comm. Hdq., A.P.O. 394, c/o Postmaster, New York, N.Y.; *for mail,* c/o Mrs. H. Posner, 1820 Clydesdale Pl., N.W., Washington, D.C.
Bros, Chester W. ('30; '35), V.P., Wm. Bros Boiler & Mfg. Co., 1057—10th Ave., S.E.; *for mail,* 4629 Browndale Ave. S., Minneapolis, Minn.
Brose, Frederic Morgner (J'40) (NMRC), 12120 Laurel Terrace Dr., North Hollywood, Calif.
Brosene, William George, Jr. (J'45) (NMCDHZ), Research Engr., Lederle Labs., Inc., Pearl River, N.Y.; *for mail,* 7501 Ross Ave., Cincinnati 16, Ohio.
Brosius, Edgar Enos ('18) (BHM), Pres., Edgar E. Brosius Co., Sharpsburg St., Pittsburgh 15, Pa.
Brosnan, Walter John ('42) (CJMDYR), Mgr. Prod. Planning, John A. Roebling's Sons Co., 640 S. Broad St., Trenton 2, N.J.; *for mail,* R.F.D. 2, Westover Rd., Morrisville, Pa.
Brossart, John Alden, Jr. (J'36) (RJCDM), Plant Supervision, Stand. Ry. Equip. Mfg. Co., 4527 Columbia Ave., Hammond; *for mail,* 2644 Wicker St., Highland, Ind.
Brossman, Martin Werner, Jr. (J'45), Graduate Research Asst., Pa. State Exper. Sta.; *for mail,* 514 W. Foster Ave., State College, Pa.
Brostrom, F. (J'44) (CLM), Engr., Bay City Iron Works, Oakland; *for mail,* 1723 High St., Alameda, Calif.
Brotzman, Irvin C. ('28; '35) (CDL), Gen. Supt., Lone Star Cement Corp., 342 Madison Ave., New York 17, N.Y.
Broudo, Saul (J'42) (ABN), Project Engr.,-Aero., Naval Aircraft Factory, Naval Base, Philadelphia; *for mail,* 1 Sharpless Rd., Melrose Park, Pa.
Broun, Richard G. (J'42) (HBL), Ensign, U.S.N.R., U.S.S. *Perkins* DD-877, F.P.O., San Francisco, Calif.
Broussard, Joseph A. (J'43) (AM), Box 2047, Baton Rouge, La.
Brousseau, Edw. W. ('20; '25; '35), Installation & Serv. Div., RCA Victor Div., Radio Corp. of Am., Camden, N.J.; *for mail,* Chestnut Hill Apts., Evergreen & Crefeld Sts., Chestnut Hill, Philadelphia, Pa.
Broward, Hoyt E. (J'41) (EJK), 812 E. 39th St., Savannah, Ga.
Browe, Ernest L. ('36) (SFEK), Power Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5; *for mail,* 738 Ormond Ave., Drexel Hill, Pa.
Brower, Jas. ('25) (BFS), Supt., Milwaukee Sewage Comm., Jones Island; *for mail,* 3021 N. 36th St., Milwaukee, Wis.
Brown, Albert ('22; '35) (BFKZS), Combustion Engr., Republic Steel Corp., Pine St.; *for mail,* 736 Hall Ave., N.W., Warren, Ohio.
Brown, Albert Kenneth ('21; '25; '35), Engr., Riggs Distler & Co., Inc., 1518 Walnut St., Philadelphia 2; *for mail,* 7216 Brent Rd., Upper Darby, Pa.

Brown, Allan Robert (J'43) (YNJHB), Layout Draftsman, Farrel-Birmingham Co., Inc., Main St., Ansonia; *for mail*, 80 Valley Rd., Hamden, Conn.
Brown, Arnold Kaywood (A'42) (BCM), V.P., Brown & Sharpe Mfg. Co., Box 1385, Providence 1, R.I.
Brown, Arthur H. (J'44) (DCNO), Indus. Engr., Columbia Steel Co., Pittsburg; *for mail*, 1740 Franklin St., San Francisco 9, Calif.
Brown, Arthur L. (J'40) (CP), Shell Oil Co., Inc., 100 Bush St.; *for mail*, 2676 Pacific Ave., San Francisco, Calif.
Brown, Arthur Laurence ('17; '25) (BHTSFL), Ch. Engr., Asst. Mgr., Associated Factory Mutual Fire Ins. Cos., 184 High St., Boston 10, Mass.
Brown, Aubrey Ingerson ('14; '21) (KSB), Prof. Htg. & Vent., Ohio State Univ., Columbus 10, Ohio.
Brown, Ben M. ('43) (CER), Gen. Supt. M.P., So. Pacific Co., 65 Market St., San Francisco 5, Calif.
Brown, Bertrand Hirsch (J'39) (BAENZS), Project Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., Main St., East Hartford 8; *for mail*, 37 Vernon St., Hartford, Conn.
Brown, Bruce Forrester ('21) (CDG), Mgr. So. Dist., Fibreboard Products, Inc., 4444 Pacific Blvd., Los Angeles 11, Calif.
Brown, Carey H. ('43) (CMBS), Gen. Supt., Serv. Depts., Kodak Park Works, Eastman Kodak Co., Rochester 13; *for mail*, 11 E. Genesee St., Scottsville, N.Y.
Brown, Carl Daniel ('21; '24) (T), Dir. of Research, Draper Corp., 25 Hopedale St., Hopedale, Mass.
Brown, Carl Fay (J'41) (CMJ), Indus. Engr., Natl. Tube Co., E. 28th St., Lorain; *for mail*, 13513 Clifton Blvd., Lakewood 7, Ohio.
Brown, Carl Frank, Jr. (J'44) (CM), Engr., Design & Processing, Chicago Broach Co., 5514 W. Lake St., Chicago 44; *for mail*, 510 Fillmore St., Oak Park, Ill.
Brown, Cecil Welsman ('15; '35) (EFS), Dir. of Engrg., Conn. Power Co., 31 Union St., New London, Conn.
Brown, Chas. A. (A'24), Secy., Asst. Gen. Mgr., Lunkenheimer Co., Box 360, Annex Sta., Cincinnati, Ohio.
Brown, Charles Ellsworth ('30) (SFE), Burns & McDonnell Engrg. Co., 107 W. Linwood Blvd., Kansas City 2, Mo.
Brown, Charles Henry ('24) (MNBCGJ), Owner, Mgr. & Engr., Brown Tool Co., Box 111, Breckenridge, Tex.
Brown, Charles Leonard (J'40) (BKEA), Instr. Mech. Engrg., Purdue Univ., Lafayette, Ind.
Brown, Clarence Blair, Jr. (J'41) (ASCE), Lt. (j.g.), U.S.N., Tech. Asst. to Planning Officer, Armament Test, Naval Air Test Center, Patuxent River; *for mail*, 3954 Cloverhill Rd., Baltimore 18, Md.
Brown, Clinton Bleecker ('17; '35) (CDKES), Sr. Engr., Ford, Bacon, & Davis, Inc., 39 Broadway, New York 6; *for mail*, Assynt House, Staatsburgh, N.Y.
Brown, Dale H. (J'43) (SHBM), Ensign, PC-776, F.P.O., San Francisco, Calif.
Brown, Darwin Scott ('25; '35) (CSF), V.P., Elec. Prod., Cincinnati Gas & Elec. Co., 4th & Main Sts., Cincinnati 2; *for mail*, 1307 Hayward Court, Cincinnati 28, Ohio.
Brown, Davis ('16) (HLO), Cons. Engr., 136 S. Spruce Ave., Montebello, Calif.
Brown, Donal Edward (J'45) (CMF), Owner, Mgr., Brown Engrg. Co., Box 409, Franklin, Pa.
Brown, Edmund D. (J'40) (ABK), Project Engrg., Pratt & Whitney Aircraft Div., United Aircraft Corp., Main St., East Hartford 8; *for mail*, 493 Adams St., Manchester, Conn.
Brown, Edwin Harrison ('43), Vice-Pres., Allis-Chalmers Mfg. Co., 1126 S. 70th St., Milwaukee, Wis.
Brown, Elton Shumway (A'43) (ABFE), Dist. Mgr., Stand. Oil Co. of N.J., Box 32; *for mail*, 727 Benoni Ave., Fairmont, W.Va.
Brown, Everett Rittenhouse, Jr. (J'40) (BHN), Mech. Design Engr., Platt-LePage Aircraft Co., Box 59, Chester; *for mail*, 4231 Hartel St., Philadelphia 36, Pa.
Brown, Francis (J'43) (LCO), Ch. Engr., Brown Linseed Corp., 2581 Richmond Terrace, Elm Park 2; *for mail*, 40 Westminster Court, Dongan Hills 4, S.I., N.Y.
Brown, Francis Andrew James (J'26) (LM), Project Engr., Armstrong Cork Co., Liberty & Mary Sts., Lancaster; *for mail*, 127 Kready Ave., Millersville, Pa.
Brown, Frank Logan ('43) (BHNJSA), Prof. Applied Mechanics, Univ. of Kan., Lawrence, Kan.
Brown, George Bentley ('24; '35) (LSCFB), Power & Process Engrg., Engrg. Div., Commercial Solvents Corp.; *for mail*, 128 Jackson Blvd., Terre Haute, Ind.
Brown, George Irving (J'20) (CMODLJ), Account Supvr., Booz, Allen & Hamilton, 135 S. La Salle St., Chicago 3; *for mail*, 12813 S. Wallace St., Chicago 28, Ill.
Brown, George Westaway (J'41) (HKSAC), Asst. to Mch. Supt., Test Dept., Shipbldg. Div., Bethlehem Steel Co., San Pedro Yard, San Pedro; *for mail*, 435 Neptune Ave., Wilmington, Calif.
Brown, George Wilmott (J'41) (MS), 24A Richardson St., Wakefield, Mass.
Brown, Graham Bauer (J'46) (CDJ), Youngstown Sheet & Tube Co.; *for mail*, 3626 Lynhaven Rd., Youngstown, Ohio.
Brown, Grant S. (J'44), Mech. Engr., Hawthorne Sta., West. Elec. Co., Inc., Chicago; *for mail*, 1508 George St., Downers Grove, Ill.
Brown, Harold David (J'41) (EMJACO), Staff Engr., Caterpillar Tractor Co., Peoria 8; *for mail*, 1500 Peoria Ave., Peoria 4, Ill.
Brown, Harry Lee (J'42) (ACM), Lt., U.S.N.R.; *for mail*, Philip, S.D.
Brown, Harry Wheeler ('09; '26) (CDJLMO), Engr., Corn Industries Research Foundation, 5 E. 45th St., New York 17, N.Y.; *for mail*, 110 Patterson Ave., Greenwich, Conn.
Brown, Harwood Ignatius (J'37) (BHLCSJ), 622 Seguin St., New Orleans 14, La.
Brown, Holcombe James ('16) (OLMCDF), Owner, Firm of H. J. Brown, 35 Doane St., Boston 9, Mass.
Brown, Howard Hayes ('08; '25) (SEFMOD), Editor, *Marine Engineering and Shipping Review*, Simmons-Boardman Publ. Corp., 30 Church St., New York 7, N.Y.
Brown, Hubert L. (J'38) (SF), Asst. Boiler Room Engr., Kansas City Power & Light Co., 1330 Baltimore, Kansas City; *for mail*, Dundee Hills, Route 4, North Kansas City 16, Mo.
Brown, Hugh Stewart ('39) (CEF), V.P. (Engrg.), Briggs & Stratton Corp., 2711 N. 13th St., Milwaukee 1; *for mail*, 2020 Two Tree Lane, Wauwatosa 13, Wis.
Brown, J. Calvin ('28) (AEF), *Manager*, '43-'44; *Vice-President*, '45-'47; Owner, Firm of J. Calvin Brown, 704 S. Spring St., Los Angeles 14, Calif.
Brown, Jacob Henry ('43) (SEHLRD), Elec. Power Designer, Cox & Stevens, Inc., 11 Broadway, New York; *for mail*, 1180 Anderson Ave., New York 52, N.Y.
Brown, James M. ('00; '06), Asst. Supt., Babcock & Wilcox Tube Co., Beaver Falls; *for mail*, Apt. 2, 222—4th St., Beaver, Pa.
Brown, James Otis (J'45) (EJS), Box 143, Rogers, Ark.
Brown, John Rowland ('00; '04; '09) (FJLNSZ), Pres., Reliance Gauge Column Co., 5902 Carnegie Ave., Cleveland 3, Ohio.
Brown, Joseph James (J'43) (FBEKNS), Exper. Automotive Engr., Ford Motor Co. of Can., Ltd.; *for mail*, 1721 Central Ave., Windsor, Ont., Can.
Brown, Lawrence Wilbert ('34; '46) (ACBS), New England Tel. & Tel. Co., 245 State St., Boston; *for mail*, 17 Bromfield Rd., West Somerville 44, Mass.
Brown, Leonard Earl (J'45) (ODLJYM), Engr., Mathieson Alkali Works, 2400 Buffalo Ave., Niagara Falls, N.Y.
Brown, Lewis F. ('30) (AFS), Route 1, Monroe, Va.
Brown, Malcolm Neal ('41) (ABHKLN), Asst. Dept. Head, Cornell Aero. Lab., 4455 Genesee St., Buffalo 5; *for mail*, 190 Deerhurst Park, Kenmore 17, N.Y.
Brown, Neil Hamilton ('39) (STLW), Mfrs. Rep., 1117 Liberty Life Bldg., Charlotte 2, N.C.
Brown, Norman Earle ('42) (ABHJMN), Ch. Designing Engr., Taft-Pierce Mfg. Co., 32 Mechanic Ave., Woonsocket, R.I.
Brown, Norton M. (J'34) (LCSYB), Constr. Supvr., Sharples Chemicals, Inc., Wyandotte; *for mail*, 996 Highland Ave., Lincoln Park 25, Mich.
Brown, Paul Hillman (J'40) (YMLN), Engr., Mch. Design, B. F. Goodrich Co., Clarksville, Tenn.
Brown, Percy ('16; '35), 596 Broad Ave., Ridgefield, N.J.
Brown, Perry H. (J'39) (HNJ), Engr., Charge Product Engrg., Joshua Hendy Iron Works, Sunnyvale; *for mail*, 525 Kenwood Dr., Menlo Park, Calif.
Brown, Ralph Lawrence, Jr. (J'41) (NOH), Designing Engr., Magnolia Petroleum Co.; *for mail*, 1725 College St., Beaumont, Tex.
Brown, Raymond Willis (J'46) (DJO), J. W. Singer Metal Works, 1908 Manchester Ave., Middletown, Ohio.
Brown, Richard Arthur (J'42) (NBMZL), Mech. Engr., Chicago Bridge & Iron Co., 1305 W. 105th St., Chicago, Ill.; *for mail*, 503 W. Arbutus St., Compton, Calif.
Brown, Richard P. ('08; '15) (ZL), V.P., Minneapolis-Honeywell Regulator Co., Wayne & Roberts Aves., Philadelphia 44; *for mail*, 3830 Oak Rd., Philadelphia 29, Pa.
Brown, Richard Robert (J'35) (CADLJ), Lt. (j.g.), U.S.N.R., Engr., Mech., REF, Panama Canal; *for mail*, Box 2867, Cristobal, C.Z.
Brown, Robert Dixon (J'42) (BHJ), Ch. Engr., Ajax Forge Co., 1956 E. 48th St., Los Angeles 11, Calif.
Brown, Robert J. (J'37) (EJS), Engrg. Div., Gen. Elec. Co., 361 Bryant St., San Francisco 6; *for mail*, 545 O'Farrell St., San Francisco 2, Calif.
Brown, Robert Stanley ('91; '04) (EJW), Secy., Mech. Engr., New Britain Mch. Co., South St.; *for mail*, 29 Russell St., New Britain, Conn.
Brown, Roger Hamilton (J'45) (EKS), Ensign, U.S.N.R., U.S.S. *Adm. W.S. Sims*, AP-127, F.P.O., San Francisco, Calif.; *for mail*, 25 Franklin St., Northampton, Mass.
Brown, Roger Stuart ('22; '35) (JLABM), Sales Engr., Calorizing Co., 136 Liberty St., New York 6, N.Y.; *for mail*, 18 Bellvale Rd., Mountain Lakes, N.J.
Brown, Russell W. (J'40) (ABE), Engr., Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail*, 348 Canyon View Dr., Los Angeles 24, Calif.
Brown, Theodore Cecil ('36; '45) (CGMNO), Prof. Mech. Engrg., Mech. Engrg. Dept., N.C. State College; *for mail*, 910 Canterbury Rd., Raleigh, N.C.
Brown, Thomas, Jr. ('30; '35) (CSBDNM), Supvr. Planning, Wage Rate & Equip., Gen. Elec. Co., 920 Western Ave., Lynn 3, Mass.
Brown, Thomas Cedric (J'45), Asst. to Cons. Mech. Engr., S.A. Township, Mining & Finance Corp., Ltd., 31 Simmonds St., Johannesburg, S. Africa.
Brown, Thomas Walter Falconer ('28; '37) (BEKSFA), Research Dir., Ch. Exec., Pametrada Research Sta., Wallsend-on-Tyne, England.
Brown, Walter Henry, Jr. ('33; '45) (HBENYW), Supt., Sewage Treatment Works, Dept. Pub. Works, City of Cranston, City Hall; *for mail*, 12 Hersey Rd., Cranston 10, R.I.
Brown, Warren August ('29; '39) (CHMS), Asst. Plant Supt., Florence Pipe Fdy. & Mch. Co., Front St., Florence, N.J.
Brown, Wendell Stimpson ('16; '18; '22) (EFH), Cons. Engr., F. P. Sheldon & Son, Hospital Trust Bldg.; also, Charles A. Maguire & Assoc., Turks Head Bldg., Providence 3; *for mail*, 201 Wentworth Ave., Providence 5, R.I.
Brown, Wilbert Rhea (J'42) (ABFNES), Sr. Test Engr., Wright Aero. Corp., Paterson 3; *for mail*, 142 Gordenhurst Ave., Upper Montclair, N.J.
Brown, William Bachman, Jr. (J'43) (HRS), Pvt., U.S.A., Co. B, 102nd Inf. Trng. Bn., 81st Inf. Trng. Regt., Camp Robinson, Ark.; *for mail*, 1304 Luttrell St., Knoxville, Tenn.
Brown, William John ('35; '45), Pres., Techni-finish Lab., Inc., 38 Scio St., Rochester 3, N.Y.
Brown, William Wilson ('44) (SKHOE), Engr., Constr. Engrg. Div., Detroit Edison Co., 2000—2nd Ave., Detroit; *for mail*, 11380 Nottingham, Detroit 24, Mich.
Brownback, Henry L. ('21; '30), Cons. Engr., 823 W. Main St., Norristown, Pa.
Browne, Albert Alexander ('45) (CJMNYZ), Mgr. Engrg. & Sales, Joshua Hendy Iron Works, Sunnyvale; *for mail*, 1755 University Ave., Palo Alto, Calif.
Browne, Andrew T. ('28; '34; '35) (BERY), ACF-Brill Motors Co., 62nd St. & Woodland Ave., Philadelphia 42, Pa.
Browne, Bard ('20; '35) (BEKRS), V.P., Sales, Superheater Co., 60 E. 42nd St., New York 17; *for mail*, Braeside Lane, Dobbs Ferry, N.Y.
Browne, Edmund Victor (J'43) (NCJMBZ), Mch. Designer, M. L. Baylard & Co., 20th & Indiana, Philadelphia; *for mail*, 338 Fairmont Ave., Philadelphia 23, Pa.
Browne, Frank A. ('13), 215 S. Aberdeen Ave., Wayne, Pa.
Browne, Kenneth Alan ('45) (RFEAB), Research Consultant, Pres.'s Office, Chesapeake & Ohio Ry. Co., 3305 Terminal Tower, Cleveland 1, Ohio.
Browne, Leland Warren ('32; '35) (CLJ), V.P., Darby Corp., 1st & Walker Sts., Kansas City 15, Kan.
Browne, William Hooper (J'36) (EFNBRS), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Brownell, Clifford Eaton (J'45) (DNBHKL), Ch. Engr., Savory, Inc., 90 Alabama St., Buffalo 4, N.Y.
Browning, Frank H. ('14) (CES), Engr. Examiner, Seattle Civ. Serv. Comm., 605 County City Bldg.; *for mail*, 1514—35th Ave., Seattle, Wash.
Browning, Winston Headly (J'42) (LKBFJH), Sr. Inspr., Shell Oil Co., Box 262, Wood River; *for mail*, 502 Reller Ave., Roxana, Ill.
Brownlie, David ('21) (SWYFEZ), Cons. Engr., 56 Grange Rd., Ealing, London W 5, England.
Brownlie, William Neilson ('42) (CEHORS), Mutual Boiler Ins. Co. of Boston, 60 Batterymarch St., Boston, Mass.
Brownscombe, Philip Joslyn ('36; '45) (NBM), 19 N. Hillside Ave., Chatham, N.J.
Brownstein, Benjamin ('22) (ABEFHL), Mech. Engr., Jones & Laughlin Steel Corp., Aliquippa; *for mail*, 517—4th St., Ellwood City, Pa.
Bruback, Theodore McLane ('41) (BH), Appropriation Control Engr., Gary Sheet & Tin Mill, Carnegie-Ill. Steel Corp.; *for mail*, 578 Broadway, Gary, Ind.
Bruce, Albert White (J'36) (EFKS), Asst. Engr., Div. of Steam Engrg., Pacific Gas & Elec. Co., 245 Market St., San Francisco 5; *for mail*, 3734 Broderick St., San Francisco 23, Calif.
Bruce, Alexander Douglas (A'45), Pres., Vapor Car Heating Co., Inc., 80 E. Jackson Blvd., Chicago 4; *for mail*, 1319 Ridge Ave., Evanston, Ill.
Bruce, Alfred W. ('43) (ERS), 242 E. 19th St., New York, N.Y.
Bruce, Archibald K. ('27) (ERS), Sr. Partner, Robert Bruce & Sons, Moorgate Hall, Moorgate, London, England.
Bruce, Cecil Douglas ('43) (HKMN), Mech. Engr., Am. Cast Iron Pipe Co., 2930 N. 16th St., Birmingham 4, Ala.
Bruce, Irving Whiton (J'45) (JMN), Engr., Head of Engrg. Dept., Pacific Gear & Tool Works, 1035 Folsom St., San Francisco 3; *for mail*, 926 Evelyn Ave., Albany 6, Calif.
Bruce, Norman Clement (J'43), Foreman, Cutting Tools Div., Neon Products of West. Can., Ltd., 260 Terminal Ave.; *for mail*, 4474 W. 5th Ave., Vancouver, B.C., Can.

Bruce, Victor Gardiner (J'42) (BNAG), Graduate Student, Stanford Univ., Box 909, Stanford Univ., Calif.
Bruce, William (J'41) (EFSK), Instr. Mech. Engrg., Univ. of Toronto, Toronto; *for mail,* 16 Graham Gardens, Toronto 10, Ont., Can.
Bruce, William Lee (J'42) (EHS), Engr's. Asst., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 1563 S. 82nd St., West Allis 14, Wis.
Bruck, Albert George ('20; '35) (MCJB), Works Mgr., King Mch. Tool Co., Clifton & B. & O. R.R., Cincinnati; *for mail,* 1529 Wittlou Ave., Cincinnati 24, Ohio.
Brucker, Lyle A. (J'41), 930 E. Russell Ave., Flint 5, Mich.
Bruckmann, Hugo C. ('27; '35) (DMW), Auchentorlie House, Seedhill Rd., Paisley, Scotland.
Bruckner, Robert Earll ('25; '41) (RENSKJ), Cons. Engr., Cons. Tech. Editor, *Produce Engineering,* McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York, N.Y.; *for mail,* Box 45, Upper Montclair, N.J.
Brueggeman, Karl O. (J'40) (R), Sgt., Co. B, 721st Ry. Opera. Bn., A.P.O. 465, New York, N.Y.; *for mail,* 1945 Kearney, Denver 7, Colo.
Bruehl, Lawrence ('30; '35) (BHKNSM), Mech. Engr., Internatl. Business Mchs. Corp., 310—5th Ave., New York 1; *for mail,* 509 E. 77th St., New York 21, N.Y.
Brugler, Maynard Wesley ('36; '44) (CAYR), Field Engr., Timken Roller Bearing Co., 16 W. 60th St., New York 23, N.Y.
Bruhn, Neils (J'24), Elec. Tester, Brooklyn-Manhattan Transit Co., 500 Kent Ave., Brooklyn; *for mail,* 362 Oak Ave., Cedarhurst, L.I., N.Y.
Bruister, Thomas James (J'41) (BJMN), Mech. Engr., Designer, Naval Research Lab., Rm. 206, Bldg. 1, Washington 20, D.C.; *for mail,* 4124 N. 3rd Rd., Arlington, Va.
Brukardt, Arthur Major (J'43) (MACE), 2nd Lt., QM. Corps, U.S.A., 83rd Co., Army Serv. Forces Trng. Center, Camp Lee, Va.
Brumbaugh, C. C. ('41) (CLDKS), Dir. of Engrg., Diamond Alkali Co.; *for mail,* R.F.D. 3, Painesville, Ohio.
Brummerstedt, Egon Fritjoph (J'43) (BDJL), Ch. Estimator, Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Brune, Carleton Edward ('22; '27; '35) (SKHFJB), Engr., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.; *for mail,* 93 Highland Ave., Metuchen, N.J.
Brunelle, Henry Eugene, Jr. (J'38) (ABMNRE), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1317 Regal Ave., Schenectady 8, N.Y.
Brunett, Adrian L. ('19; '25) (BMS), Mech. Engr., Fed. Works Agency, Pub. Bldgs. Admin., Repair Sec., Procurement Bldg., 7th & D Sts., S.W., Washington, D.C.; *for mail,* Box 36, Rockville, Md.
Bruney, Robert Charles (J'41) (AMJNYZ), Engrg. & Equip. Inspr. (Ord.), U.S. Naval Ord. Plant, 21st & Arlington, Indianapolis 6; *for mail,* 5023 E. 17th St., Indianapolis 1, Ind.
Bruning, John M. ('31; '35) (AS), Oper. Engr., Bd. of Education, 1334 York Ave.; *for mail,* 500 E. 72nd St., New York, N.Y.
Brunkhardt, Fred W. (J'39) (CD), Engrg. Consultant, Charles W. Cromb Co., Cunard Bldg., 220 S. 16th St., Philadelphia 2, Pa.
Brunn, Frederick Milton (J'43) (SCKFM), Jr. Research Engr., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Brunner, Clarence Hargis ('41) (BLS), Mech. Engr., Austin Co., 1924 Broadway, Oakland; *for mail,* 218 Pala Ave., Piedmont 11, Calif.
Brunner, Jules E. (J'43) (SFLKC), Ch. Power Plant Engr., Glidden Co. (& Durkee Famous Foods), 5165 W. Moffat, Chicago 39; *for mail,* 7814 Karlov, Skokie, Ill.
Brunner, Mathias John (J'43) (BKS), 768 E. 52nd St., Brooklyn, N. Y.
Brunot, Albert William ('38; '45) (BNJKG), Mech. Sec. Head, Thomson Lab., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 19 Falls St., East Lynn, Mass.
Bruns, H. Richard ('45) (SNKLMD), Squad Leader, Mech. Dept., Ebasco Services, Inc., 2 Rector St., New York; *for mail,* 44 Vincent Ave., New Dorp, S.I. 6, N.Y.
Bruns, John Dickson (J'43) (AK), Engr. (Air Conditioning), Boeing Aircraft Co., Seattle; *for mail,* 5111 Hudson St., Seattle 8, Wash.
Bruns, Robert Stewart, Jr. (J'41) (KSOF), 315 Byron Pl., Maywood, N.J.
Brunyate, William Richard (J'43) (AES), 108 Grant Ave., Bound Brook, N.J.
Brupbacher, Benjamin Simon, Jr. (J'45) (ACY), 1556 Paul Morphy St., New Orleans, La.
Brusca, L. Joseph ('30; '31; '35) (NBHY), Sales Engr., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia; *for mail,* 511 Arbor Rd., Cheltenham, Pa.
Brush, C. Benjamin ('31) (CM), Saw Mill Supply Inc., 957 Saw Mill River Rd., Yonkers; *for mail,* 40 Maywood Rd., New Rochelle, N.Y.
Brush, Harold Frank (J'40) (HJKCMR), Indus. Engr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., 31 Clinton St., Newark 2, N.J.
Brusher, William Francis (J'42) (LKOCBN), Engr., Design Sec., Gen. Engrg. Dept., Stand. Oil of Calif., 225 Bush St., San Francisco; *for mail,* 3912 Greenwood Ave., Oakland 2, Calif.
Brusso, Andrew M. (J'44) (BJM), Engrg. Draftsman, Welin Davit & Boat Corp., Perth Amboy; *for mail,* 391 MacArthur Ave., Long Branch, N.J.
Brust, George A. ('45) (LMO), Ch. Planning Engr., Refrigeration Div., Internatl. Harvester Co., U.S. Highway 41, Evansville, Ind.
Bruton, Donald Lee (J'45) (CD), Lt. (j.g.), U.S.N.R.; *for mail,* Apt. 101, 5610 Enright, St. Louis 12, Mo.
Bruzelius, Edward M. ('34; '44) (EFJCLK), Lt. Comdr., U.S.N.R., Officer in Charge, Office of Inspr. of Naval Matl., U.S.N., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Bryan, Alan Stuart (J'35) (KTHBJY), Project Engr., Am. Bemberg Corp., Elizabethton; *for mail,* Box 148, Johnson City, Tenn.
Bryan, Bruce Winfield (J'46) (CKY), 802 S. Ironwood, South Bend, Ind.
Bryan, Edward Everett ('45) (CKS), Supt. Bldgs. & Power Plant, Vanderbilt Univ., 21st Ave. S. & Broad St., Nashville 4, Tenn.
Bryan, Emmett Wren ('42) (SLKJ), Ch. Engr., Lumsden & Van Stone Co., 426—1st St., Boston 27; *for mail,* 14 North St., Stoneham 80, Mass.
Bryan, Herbert Gayle (J'43) (ESFOBK), Head, Internal Combustion Plant Unit, Rural Electrification Admin., U.S. Dept. of Agric., Washington, D.C.
Bryan, John K. (J'40) (SKJF), Engr. Asst. to Mech. Supt., Venice Power Plant, Union Elec. Co. of Mo., Venice, Ill.
Bryan, Justus L. ('30; '35) (BCM), Ch. Draftsman, IMO Pump Dept., De Laval Steam Turbine Co., 853 Nottingham Way, Trenton 2; *for mail,* 1202 Pennington Rd., Trenton 8, N.J.
Bryan, Marcus K. ('16; '35) (FSE), Cons. Engr., Charles T. Main, Inc., 201 Devonshire St., Boston 10, Mass.
Bryan, R. G. (J'39), Capt., 7235 Euclid Ave., Chicago, Ill.
Bryan, Walter Edwin ('14; '20) (SRCFAB), Engr. Consultant & Analyst, St. Louis Pub. Serv. Co., 3869 Park Ave., St. Louis 10, Mo.
Bryan, William Lyttle, Jr. (J'45) (KN), Research Engr., York Corp., York, Pa.
Bryans, Dayton R. (J'38) (WDM), Packaging Engr., Carrier Corp., 300 S. Geddes St., Syracuse 2, N.Y.
Bryans, Henry Bussell ('17; '18) (CDFOSZ), Exec. V.P., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Bryans, Henry T. (J'40) (FKS), Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5; *for mail,* 8 Farm Rd., Wayne, Pa.
Bryans, William R. ('19) (BNWA), Asst. Dean, Prof. Engrg. Mechanics, College of Engrg., N.Y. Univ., Bronx 53, N.Y.
Bryant, Elmer John ('21; '28) (MCJA), Detroit Dist. Mgr., Greenfield Tap & Die Corp., 5850—2nd Blvd., Detroit 2, Mich.
Bryant, James Lewthwaite (J'45) (WOANCS), Jr. Mech. Engr., Pacific Mills, Ltd.; *for mail,* Box 6, Ocean Falls, B.C., Can.
Bryant, Jerrus Madison ('27; '35) (BCJMNY), Ch. Engr., Link-Belt Co., 519 N. Holmes Ave., Indianapolis 8, Ind.
Bryant, John George (J'44) (ANMGBD), Research & Devel. Engr., Am. Chain & Cable Co., Inc., Ross St.; *for mail,* 48 S. River St., Wilkes-Barre, Pa.
Bryant, Lorton William (J'35) (NSACBK), Mech. Engr., Roots-Connersville Blower Corp., Connersville, Ind.
Bryant, P. John, II (J'39) (CDAM), Asst. Managing Secy., Engrg. Soc. of Detroit, 100 Farnsworth, Detroit 2, Mich.
Bryant, Robt. E. (J'18) (BCM), Purchasing Dept., Buffalo Arms Corp., Box C, Sta. E, Buffalo; *for mail,* 314 Washburn St., Lockport, N.Y.
Bryant, William Whitsel (J'35) (CSO), Staff Asst., Elev. Div., Westinghouse Elec. Corp., 150 Pacific Ave., Jersey City 4, N.J.
Bryce, James ('31; '35) (SFEC), Ch. Inspr., Milwaukee Inspc. Dept., Md. Casualty Co. (Baltimore), 611 N. Broadway, Milwaukee 2, Wis.
Bryden, Colby William (J'23), 329 Mountain Ave., Westfield, N.J.
Bryson, William Donal ('45), 2nd Asst. Supt., Internatl. Harvester Co., 5 Pulaski St.; *for mail,* 152 S. Hoopes Ave., Auburn, N.Y.
Bubar, Hudson H. ('19; '21) (OSFD), 15 Park Row, New York 7, N.Y.
Bubar, Willard Clark ('44) (SFC), Field, Design Engr., Bubar Dust Systems, 15 Park Row, New York, N.Y.
Bubb, Milford John ('45) (BJS), M.M., Commonwealth Edison Co., Box 149, Pekin, Ill.
Bublitz, Charles A., Jr. (J'43) (WCRDM), Maj., 576 McKinley St., Gary, Ind.
Buccola, Charles H. (J'30) (NBA), Elec. Designer, Sperry Gyroscope Co., Inc., Great Neck, L.I.; *for mail,* 4155—70th St., Woodside, L.I., N.Y.
Buchan, Lloyd P. (J'39), 413 Centre Square, Milford, Pike Co., Pa.
Buchanan, D. Dwight ('18; '21; '35), 33—6th St., S.E., Massillon, Ohio.
Buchanan, Kenneth Alex (J'42) (ACM), 18 Boswell Ave., Toronto, Ont., Can.
Buchanan, Wm. C. ('18; '35), 419 Columbia Terrace, Peoria, Ill.
Buchberg, Harry (J'41) (AHNKBZ), Research Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 8616 W. 3rd St., Los Angeles 36, Calif.
Buchen, Joseph Clement ('18; '21) (CLMK), V.P., Leslie Salt Co., Newark; *for mail,* 840 Breed Ave., San Leandro, Calif.
Bucher, George Heisler ('44) (ABMRSH), Vice-Chmn., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30, Pa.
Bucher, Paul ('21; '35) (SFHKEB), Prof. Steam Engrg., Ohio State Univ., Columbus 10, Ohio.
Buchhagen, Walter Harry ('44) (CDLKMO), Coordination Engr., Am. Cyanamid Co., 30 Rockefeller Plaza, New York 20, N.Y.
Buchi, Alfred J. (Non-Member), *Melville Medallist,* '37; Büchi Syndicate, 20 Salstr., Winterthur, Switzerland.
Buchmann, Karl Emil ('25; '31; '35) (NBDFOG), Design Engr., McIntyre Porcupine Mines Ltd., Schumacher; *for mail,* 13 Birch St. S., Timmins, Ont., Can.
Buchsbaum, Arnold (J'37) (ENBJ), Mech. Engr., Diesel Design & Testing, N.Y. Naval Shipyard, Brooklyn; *for mail,* 1360—48th St., Brooklyn 19, N.Y.
Buck, Everett S. ('38) (EFKOS), Cons. Mech. Engr., Ft. Wayne Air Conditioning Co., 223 E. Main St., Ft. Wayne 2, Ind.
Buck, Nelson Lewis ('38; '45) (AO), 38 S. Lincoln St., Washington, Pa.
Buck, Robert S. (J'42) (ABL), 2117 Lemon Ave., Temple City, Calif.
Buckalter, Lloyd Arthur (J'44), 156 W. 86th St., New York 24, N.Y.
Buckalter, Robert Irving (J'41) (JAM), Tech. Engr., Air Conditioner Design, Gen. Elec. Co., 5 Lawrence St., Bloomfield; *for mail,* 25 Oxford St., Montclair, N.J.
Buckholtz, Ira Elmer (J'40), Capt., Corps of Engrs., U.S.A., A.P.O. 7511, c/o Postmaster, New York, N.Y.
Buckingham, Earle ('13; '35) (BMA), *Worcester Reed Warner Medallist,* '44; Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge, Mass.
Buckingham, Walter Harlow (J'38) (ESZFC), Clark Bros. Co., Olean, N.Y.; *for mail,* 173 Davis St., Bradford, Pa.
Buckland, Bruce O. (J'24) (ABS), Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Buckley, Frederick Doyle (J'44) (SKRAFB), Research Engr., Battelle Memorial Inst., 305 King St., Columbus; *for mail,* 502 Colonial Ave., Worthington, Ohio.
Buckley, Guidet M. ('22; '25; '35) (ESHR), Retired; 178 Hillcrest Rd., Mt. Vernon, N.Y.
Buckley, Raymond William ('45) (SNFDBH), Ch. Engr., Imperial Paper & Color Corp., Lower Warren; *for mail,* 137 Ridge St., Glens Falls, N.Y.
Buckminster, Lloyd Allen (J'44) (ADH), Devel. Engr., Dept. A13A, Am. Optical Co.; *for mail,* 34 Prospect St., Southbridge, Mass.
Buckowski, Henry John (J'40) (JLM), Capt., 36th Engr. Combat Gp., A.P.O. 541, c/o Postmaster, New York, N.Y.
Buckwalter, Tracy V. ('13) (BFJRTW), Retired; 2301 Del Mar Pl., Ft. Lauderdale, Fla.
Budd, Edward Gowen ('46), *A.S.M.E. Medallist,* '44; Pres., Edw. G. Budd Mfg. Co., 25th St. & Hunting Park Ave., Philadelphia, Pa.
Budd, Ralph (H'43) Pres., Chicago, Burlington & Quincy R.R. Co., 547 W. Jackson Blvd., Chicago 6, Ill.
Budde, Albert A. (J'39) (ZACNBO), Exec. Engr., Instrument Research Div., Natl. Advis. Com. for Aero., Langley Field; *for mail,* 225 Hampton Roads Ave., Hampton, Va.
Budden, Edgar LeRoy (J'45), Devel. Engr., Doubleday & Co., 14 W. 49th St., New York; *for mail,* 10903—86th Ave., Richmond Hill, N.Y.
Buddine, Norman Theodore (J'28) (LKCZH), Process Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 29 Bancroft Rd., Wellesley Hills 82, Mass.
Budell, Robert (J'44) (JMASFL), Blanch Ave., West Norwood, N.J.
Budell, William (J'37) (CSH), Asst. Plant Engr., Congoleum-Nairn, Inc., Cedarhurst; *for mail,* 4 New Windsor Rd., Westminster, Md.
Budenholzer, Roland Anthony (J'42) (SKBEAH), Assoc. Prof. Mech. Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.
Budny, Walter Vincent (J'39) (AECM), 2nd Lt., Engrg. Officer, Army Air Forces, Maxwell Field, Ala.; *for mail,* 2691 S. 12th St., Milwaukee 7, Wis.
Budwell, Leigh ('16; '21) (CFR), Ch. Mech. Officer, Richmond, Fredericksburg & Potomac R.R. Co., Richmond 20; *for mail,* 1600 Confederate Ave., Richmond 22, Va.
Budzyko, Edward James, Jr. (J'44) (CAMJON), Lt. (j.g.), U.S.N.R., Naval Serv. Depot, Navy 3149, F.P.O., San Francisco, Calif.; *for mail,* Box 160, R.D. 1, Bristol, Pa.
Buechler, Ralph M. (J'39) (ABK), Maj., Army Air Forces; *for mail,* 631 Main St., Hamilton, Ohio.
Buenger, Edw. F. ('30), V.P., Charge Mfg., Wilson-Jones Co., 3300 Franklin Blvd.; *for mail,* 836 Lathrop Ave., River Forest, Ill.
Buensod, Alfred Charles ('15; '27) (TG), Pres., Buensod-Stacey, Inc., 60 E. 42nd St., New York 17; *for mail,* 33—5th Ave., New York 3, N.Y.
Buerer, Wayne Wallace ('43) (EFSAYZ), Design Engr., Westinghouse Elec. Co., Lester Branch P.O., Philadelphia 13; *for mail,* 438 Sutton Ave., Folsom, Pa.

Buerk, Benjamin Christian (J'31) (MNCJ), Partner, Gen. Mgr., Buerk Tool Works, 315 Grote St., Buffalo 7; *for mail,* 192 High Park Blvd., Eggertsville 21, N.Y.
Buffinton, Alfred L. (J'45) (SFLE), Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Buffum, Timothy Nicholson (J'45), Ensign, U.S.N., 890 Urbano Dr., San Francisco 12; *for mail,* 3371 Jackson St., San Francisco, Calif.
Buford, Edwin Hardy ('39) (DLY), Mgr., Design Div., Monsanto Chem. Co., 1700 S. 2nd St., St. Louis 4, Mo.
Bugge, Sven B. ('16) (HKW), V.P., Gen. Mgr., Tomahawk Kraft Paper Co., Tomahawk, Wis.
Buhler, Carl (J'42), 7157—69th St., Brooklyn 27, N.Y.
Bujak, Henry Carl (J'40) (NBAH), Sr. Engr., Vickers Inc., 1400 Oakman Blvd., Detroit 32; *for mail,* 15416 Wildemere Ave., Detroit 21, Mich.
Bukaty, Raymond M. (J'45), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Bulen, William Herman (J'44) (MRNAHJ), 220 E. 10th, Escondido 1, Calif.
Bulger, William Arthur ('43) (CMFK), V.P., Charge of Mfg., Petroleum Heat & Power Co., Southfield Ave.; *for mail,* Route 36, Old N. Stamford Rd., Stamford, Conn.
Bulk, George Kenneth (J'44) (ANJYBG), Designer, Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 717 Swarthmore Ave., Pacific Palisades, Calif.
Bullard, Edward Clarke ('29) (CNM), V.P., Gen. Mgr., Bullard Co., 286 Canfield Ave., Bridgeport 2, Conn.
Bullard, Edw. P. ('13), *A.S.M.E. Medallist,* '37; Pres., Bullard Co., 286 Canfield Ave., Bridgeport 2, Conn.
Bullard, John Ervin (A'13) (AWRCYG), Free Lance Writer, Tech. & Business Articles, Box 38, Central Valley, N.Y.
Bullen, Clarence Kirkwood ('28; '30) (CHOMD), Civilian Prod. Admin., Key Bldg., Oklahoma City 2; *for mail,* 200 N.E. 14th St., Oklahoma City 4, Okla.
Bullinger, Clarence Edward ('24; '35) (CMDO), Prof., Head, Dept. of Indus. Engrg., Pa. State College; *for mail,* 637 West Foster Ave., State College, Pa.
Bullock, Harry Leslie ('44) (YLMCTN), Dir., Bullock-Smith Associates, 136 Liberty St., New York 6, N.Y.
Bullock, Joseph B. (J'36) (SFKJ), Asst. to Supt., Power Prod. Stas., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington Bldg., Baltimore 3; *for mail,* 1940 Breitwert Ave., Baltimore 30, Md.
Bullock, Richard Gould (J'41) (MSWRCD), Lt., U.S.N.R.; 4333—1st Ave., N.E., Seattle 5, Wash.
Bullwinkel, Herman Henry ('44) (EKS), Dist. Turb. Engr., Cent. Dist., Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Bumgardner, Harvey Edward ('40) (SCYFZ), Supvr. of Library & Secy., Pat. Com., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Bump, Burton Nelson ('92; '04) (OFSAKC), 120 Clifton Pl., Syracuse 6, N.Y.
Bumstead, Robert (J'40) (ACY), Engr., Mfrs. Mutual Fire Ins. Co., 10 Weybosset St., Providence 1, R.I.; *for mail,* 5513 Morris St., Philadelphia 44, Pa.
Bunce, John Peter (J'45) (JMR), Asst. Adv. Mgr., Kearney & Trecker Corp., West Allis 14; *for mail,* 3119 N. 26th St., Milwaukee 6, Wis.
Bunevich, Robt. R. (J'39) (BCS), Draftsman, Navy Dept., Anacostia, D.C.; *for mail,* 158 Gregory Ave., Passaic, N.J.
Bunge, Ralph W. ('15; '26) (BCDFJK), Asst. Cons. & Supvg. Engr., Stand. Brands, Inc., 595 Madison Ave., New York, N.Y.
Bunke, Edward William Diedrich (J'36) (BEA CMJ), Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Bunker, Henry Lloyd, Jr. ('42) (EKL), Process Engr., E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Bunker, Philip Dearborn (J'36) (LBDJSW), Engr., Cheney-Bigelow Wire Works, 417 Liberty St.; *for mail,* 105 High St., Springfield, Mass.
Bunker, Richard Davidson (J'45) (CMDSJO), Seaman, 1st Class, U.S.N., Draftsman, Pub. Works Bldg. 131, Naval Trng. Center, Great Lakes, Ill.; *for mail,* 458 Essex St., Lynn, Mass.
Bunker, Warren W. (J'41) (SEM), 6507 Sefton Ave., Baltimore 14, Md.
Bunker, William Beehler ('43) (BDR), Lt. Col., U.S.A., New Orleans Procurement Office, Trans. Corps, Box 285, New Orleans 9, La.
Bunker, William L. ('23) (CFM), Cosmopolitan Shipping Co., 42 Broadway, New York, N.Y.; *for mail,* 54 Ardsley Rd., Montclair, N.J.
Bunn, Edward S. ('41) (JKL), Revere Copper & Brass, Inc., 1301 Wicomico St., Baltimore, Md.
Bunn, William Bennett, Jr. ('45) (JPS), Welding Engr., M. W. Kellogg Co., Box 469, Jersey City, N.J.
Bunnell, Sterling Haight ('94; '03; F'38) (OOJ YAE), V.P., Secy., Staff Mem., George S. Armstrong & Co., Inc., 52 Wall St., New York 5, N.Y.; *for mail,* 2225 Main St., Stratford, Conn.
Buntin, Roger Williams (J'45) (CFKOSW), Salesman, Babcock & Wilcox Co., 1120 Packard Bldg., Philadelphia 2, Pa.
Bunting, Francis Washington ('33; '43) (BSR HKZ), Mech. Engr., No. Equip. Co., 1945 Grove Dr., Erie; *for mail,* 1939 Oxford St., Erie, Pa.
Bunting, John T. (J'41) (BJ), 4112 N. Keeler Ave., Chicago 41, Ill.
Bunting, Joseph W. ('46) (FKS), Prof. & Acting Head of Mech. Engrg., Univ. of Cincinnati, Cincinnati, Ohio.
Bunzel, Erwin ('16) (DHKLS), Ch. Draftsman, Stand. Brands, Inc.; *for mail,* 936 Orchard St., Peekskill, N.Y.
Buonaccorsi, Alphonse Lawrence (J'35) (SOK EHD), Partner, G. M. Richards & Associates, 333 Kearney St., San Francisco, Calif.
Buote, Clarence Edmund ('44), Ch. Engr., H. & B. Am. Mch. Co.; *for mail,* 205 Cleveland St., Pawtucket, R.I.
Burack, William Durwood ('31; '34; '37) (SDB LKJ), Checker, Chem. Constr. Corp., 350—5th Ave., New York 1, N.Y.; *for mail,* 77 Sycamore Ave., Livingston, N.J.
Burandt, Richard John (J'43), Liaison Engr., Vought-Sikorsky Aircraft Div., United Aircraft Corp., Stratford; *for mail,* 395 Ruth St., Bridgeport, Conn.
Burch, Kenyon C. ('35; '45) (ABC), Supvr. Liaison Engrg., Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 715 S. Oxford Ave., Los Angeles 5, Calif.
Burchfield, William Filer (M'39) (ABJLMN), Devel. & Research Div., Internatl. Nickel Co., Inc., 67 Wall St., New York 5; *for mail,* 115-25 Metropolitan Ave., Kew Gardens, N.Y.
Burckhalter, Frank Oliver (J'45) (AES), Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 109 Penn St., Ridley Park, Pa.
Burdett, Horace Herman ('45) (CDMNTW), Charge of Planning, Draper Corp., Hopedale; *for mail,* 12 West St., Milford, Mass.
Burdg, George Weaver (J'42), 1354 Warren Rd., Lakewood, Ohio.
Burdick, Herbert ('17; '23) (CJNMBY), Foreman, Gen. Research Lab., Underwood Corp., 56 Arbor St., Hartford; *for mail,* 198 N. Oxford St., Hartford 5, Conn.
Burdick, Lewis Remington ('36) (FSKE), Fuel Engr., U.S. Bur. of Mines, S. Interior Bldg., Washington 25, D.C.; *for mail,* 10008 Dallas Ave., Silver Spring, Md.
Burdick, Theo. A. ('18; '25) (SFEH), 3328—158th St., Flushing, L.I., N.Y.
Burdick, William Elliott ('39) (RBJ), Asst. to Serv. Mgr., Gen. Steel Castings Corp., Eddystone, Pa.
Burdoin, Allen Jeffers ('45) (HOESZD), Sr. Engr., Metcalf & Eddy, 1300 Statler Bldg., Boston 16, Mass.
Burgan, A. L. ('97; '99; '03), Mill Supt., Calumet & Hecla Consltd. Copper Co., Hubbell, Mich.
Burgener, Leon George (J'42) (BLM), Micro Switch Corp.; *for mail,* 1006 W. Douglas St., Freeport, Ill.
Burger, George E. ('24; '33), Supt., Burger Bluestone Co., Box 781, Kingston; *for mail,* 62 South St., Warwick, N.Y.
Burger, Murray (J'42) (SKD), Asst. Engr., Bur. of Ord., Navy Dept., Navy Bldg., Washington 25; *for mail,* 1431 Varnum St., Washington 11, D.C.
Burger, William Henry, Jr. (J'38) (NWBMDK), Process Devel. Engr., Kimberly-Clark Corp., Commercial St.; *for mail,* Route 1, Neenah, Wis.
Burger, William John ('44) (N), Dir. Engrg., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3, Ohio.
Burgess, Charles G. ('44) (KSBHEL), Asst. Ch. Engr., Alco Products Div., Am. Loco. Co., 30 Church St., New York 8; *for mail,* 90 Stebbins Ave., Tuckahoe, N.Y.
Burgess, Donald ('34; '35), Asst. Supvg. Oper., Buffalo Gen. Elec. Co., Elec. Bldg., Buffalo; *for mail,* Apt. 7-D, Pomeroy Lane, Eggertsville 21, N.Y.
Burgess, James Allan (J'37) (CDJMN), Plant Mgr., Wallaceburg Brass, Ltd., 1255 Wallace St., Wallaceburg, Ont., Can.
Burgess, James Roy (24; '35) (TLMKDB), Mech. Supt., Stauntons Ltd., Leaside, Toronto, Ont., Can.
Burgess, Leslie Mitchell (J'43) (KMYHCE), Engr., Drayer & Hanson, Inc., 3310 Medford St., Los Angeles 54, Calif.
Burgess, M. L. ('44) (CKS), Omaha Mgr., Westinghouse Elec. Corp., Rm. 712, Elec. Bldg., Omaha 2, Neb.
Burgess, Neil, Jr. (J'43) (ACSE), Devel. Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 285 Porter St., Melrose 76, Mass.
Burgess, Walter Everett ('41) (CDH), Engr., Central Fdy. Co., 386—4th Ave., New York, N.Y.; *for mail,* 23 Donaldson Ave., Rutherford, N.J.
Burgess, Wilfrid Mellon ('44) (BJM), Ord. Engr., Watervliet Arsenal, Watervliet; *for mail,* 21 Center View Dr., Troy, N.Y.
Burggraf, John Charles (J'38) (CNJMD), Field Foreman, Constr. Dept., Bethlehem Steel Co., Baltimore 19; *for mail,* 418 Hilton St., Baltimore 29, Md.
Burhoe, Lemuel Newton ('37) (CMJKBS), Gen. Supt., Riverside Metal Co., Pavilion Ave., Riverside; *for mail,* 300 Midway St., Riverton, N.J.
Burke, Arthur James (J'37) (ADH), Ensign, U.S.N.R., LST-1010, F.P.O., San Francisco, Calif.; *for mail,* 13-07 Fair Lawn Ave., Fair Lawn, N.J.
Burke, Edward Boaz (J'46) (ABK), **Jr.** Research Engr., Research Labs., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 5412 Bartlett St., Pittsburgh, Pa.
Burke, Henry ('45) (HKS), Asst. Boiler Mill Engr., West. Cartridge Co., East Alton, Ill.; *for mail,* 5536 Pershing, St. Louis, Mo.
Burke, Henry E. ('40; '42) (FHS), Design Engr., Los Angeles Bur. of Power & Light, 207 S. Broadway; *for mail,* 3746 West Blvd., Los Angeles, Calif.
Burke, Jack B. (J'42) (ACF), 141 Academy Rd., Hatboro, Pa.
Burke, John J. (J'45) (CHM), Standardization Work, Bell Tel. Labs., Inc., 463 West St., New York 14; *for mail,* 117-40—224th St., St. Albans 11, N.Y.
Burke, Joseph J. (J'41) (BHS), Sales Engr., Gardner-Denver Co., 76—9th Ave., New York, N.Y.; *for mail,* 68 Williamson Ave., Hillside, N.J.
Burke, Norman ('37), Bd. of Dirs., James Howden & Co., Ltd., 195 Scotland St., Glasgow C. 5, Scotland.
Burke, R. F. ('21), Boiler Engrg. & Supply Co., Manarvon St., Phoenixville, Pa.
Burke, Thomas Francis (J'43) (EHKBSR), Lt. (j.g.), U.S.N.R., Fleet Torpedo Sch. Staff, San Diego 36; *for mail,* 2513 McAllister St., San Francisco 18, Calif.
Burke, Thomas Ulic (J'45) (AC), 10938 Strathmore Dr., West Los Angeles 24, Calif.
Burkett, William J. (J'42) (FEKCSL), Ch. Engr., Green's Fuel, Inc., Broadway & Green Sts., Sarasota, Fla.
Burkey, Irwin Russell (J'42) (TCMDA), Asst. Mgr., Hamburg Knitting Mill & Bleach Works, Pine St.; *for mail,* 125 N. 5th St., Hamburg, Pa.
Burkhardt, Everett Robert ('23; '30; '32) (EH OS), Mech. Engr., Holyoke Water Power Co., 1 Canal St.; *for mail,* 271 Pleasant St., Holyoke, Mass.
Burkhardt, John Boden (J'45) (SFKZH), Engr., Jr. Grade, Steam Effic. Bur., Commonwealth Edison Co., 1111 W. Cermak Rd., Chicago 8; *for mail,* 7313 S. East End Ave., Chicago 49, Ill.
Burkholder, Chas. Irvine ('12) (SFHKTO), V.P., Ch. Engr., Duke Power Co., 422 S. Church St., Charlotte 1, N.C.
Burkley, John K. (J'42) (ACF), 1857 Goodyear Blvd., Akron, Ohio.
Burkum, Oliver Joseph (J'44) (NKDL), Draftsman, So. Alkali Corp.; *for mail,* 257 W. Vanderbilt, Corpus Christi, Tex.
Burlesco, Mario Sebastian (J'42) (ACBMEK), Lt., Admin. Asst., Army Air Forces, Air Matl. Comd., Detroit Army Air Force Procurement Field Office, 7100 Lonyo Rd., Detroit 32, Mich.
Burleson, Ambrose L. ('44) (FKS), 1109 Parker St., Berkeley 2, Calif.
Burley, John W., Jr. (J'42), 650 Ocean Ave., Brooklyn, N.Y.
Burling, Herbert Sherman ('13; '24) (KLZ), Pres., Burling Instrument Co., 253 Springfield Ave., Newark 3, N.J.
Burlingame, Charles R. ('20; '25) (FKSELC), Combustion Engr., Pittsburgh Coal Co., Oliver Bldg., Pittsburgh 30, Pa.
Burlingame, J. H. (A'20) (CLM), Asst. Gen. Mgr., West. Adjustment & Inspc. Co., 175 W. Jackson Blvd., Chicago; *for mail,* 928 Judson Ave., Evanston, Ill.
Burlingame, Walter S. ('35) (EFS), 758 W. King St., Decatur, Ill.
Burlingame, William Brayton ('38) (NBCROA), J. H. Horne & Sons, Inc., Lawrence, Mass.; *for mail,* Arbor St., Exeter, N.H.
Burmester, Lawrence R. ('34; '43) (AC), 606 Sylvan Ave., San Mateo, Calif.
Burmistroff, John G. ('24; '35), Testing Engr., Vibration Specialty Co., Harrison Bldg.; *for mail,* 1733 Spring Garden St., Philadelphia, Pa.
Burn, Robert William (J'43) (MCA), 32 Diamond St., Arcadia, Calif.
Burnell, John Gurner ('24) (HKS), Tech. Dir., Mgr., Thompsons Engrg. & Pipe Co., Parker St., Castlemaine, Victoria, Aust.
Burnett, Clair Lee ('42) (MCDHWO), Owner, C. L. Burnett & Associates, R.F.D. 1, Cave Spring, Ga.
Burnett, David Joy ('44) (SEHL), Engr., Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.
Burnett, Earle Smead ('13; '35) (ABEHKZ), Mech. Engr., Bur. of Mines, Amarillo Helium Plant, Box 911; *for mail,* 4223 W. 11th Ave., Amarillo, Tex.
Burnett, William Elton (J'45) (RHCAN), Design Engr., Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.; *for mail,* Apt. 306, 45 N. Fullerton Ave., Montclair, N.J.
Burnette, A. R. ('20) (O), Owner, Firm of A. R. Burnette, Engr., 120 Wall St., New York 5, N.Y.
Burnham, C. H. M. ('27) (E), V.P., Ch. Engr., Panhandle East. Pipe Line Co., 1221 Baltimore Ave., Kansas City 6, Mo.
Burnham, Charles ('17; '35) (ABJLW), Retired; 175 Tyler Ave., Groton, Conn.
Burnham, Leland Frank ('21; '25) (BCJLNY), Ch. Draftsman, Buffalo Forge Co., 490 Broadway, Buffalo 4, N.Y.
Burnite, Alvin W. ('46) (KOS), Ch. Engr., Edge Moor Iron Works, Inc., Edge Moor; *for mail,* 813 Bellevue Rd., Wilmington 280, Del.
Burns, Alan Elmer ('24; '35) (PFS), Propr., Burns Supply Co., 27-05—43rd Ave., Long Island City, N.Y.

Burns, Alan Elwin ('20; '35) (ES), 309 Station Ave., North Hills, Pa.
Burns, Alexander (J'46) (BMY), Draftsman, Designer, Loft Candy Corp., 38-38—9th St., Long Island City; *for mail,* 2280 Andrews Ave., Bronx 53, N.Y.
Burns, Bert H. (J'41), 2nd Lt., Army Air Forces, 23rd Bomb. Sqd., Hickam Field, T.H.
Burns, Charles Henry MacLeod ('22) (CDMFJ), Dir., Trade & Industry Branch, Dept. of Planning & Devel., Queen's Pk.; *for mail,* 114 Ardagh St., Toronto, Ont., Can.
Burns, Frank Donald ('44), Long Beach, Michigan City, Ind.
Burns, Henry (J'38) (YAESFB), Hughes Aircraft Co., Florence & Teale St., Culver City; *for mail,* 1545 N. Las Palmas Ave., Hollywood 28, Calif.
Burns, Homer S. ('11), Box 146, Wake Forest, N.C.
Burns, John Joseph, Jr. (J'45) (ABNJOC), Stress Analyst, Cessna Aircraft Corp.; *for mail,* 239 N. Hillside Ave., Wichita, Kan.
Burns, Louis G. (J'40) (AFH), Capt.; 32 Ohio St., Osborn, Ohio.
Burns, Nicholas Smyth, Jr. (J'46) (EJS), U.S.N.; Box 66, Dade City, Fla.
Burns, Rodney C. (J'43) (EHBSNA), Project Supvr., Carrier Corp., South Geddes St., Syracuse 1; *for mail,* Rockwell Rd., Nedrow, N.Y.
Burns, Thomas ('41) (CMOS), Exec. Engr., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.; *for mail,* 50 S. Highwood Ave., Glen Rock, N.J.
Burns, Willard A. (J'41) (ACML), Reichard Coulston, Inc., Mauch Chunk Rd., Bethlehem, Pa.
Burns, William Julian (J'41) (SEZOFH), Mathieson Alkali Works, Inc., Saltville, Va.
Burnside, John William (J'44) (AJM), Sr. Clerk, Engrg. Dept. 78, Buick Motor Div., Gen. Motors Corp., Flint; *for mail,* 115—11th Ave., Flint 4, Mich.
Burnside, M. C. ('19; '35) (ACMO), Col., Army Air Forces, Air Tech. Serv. Comd., Office of A.A.F. Rep., Briggs Mfg. Co., 4621 Conner Ave., Detroit 13; *for mail,* 23 Gladstone Ave., Detroit 2, Mich.
Burpee, Frank Eugene ('14) (CFEW), Prof. Emeritus Mech. Engrg., Supt. Bldgs. & Grounds, Bucknell Univ.; *for mail,* 110 S. 2nd St., Lewisburg, Pa.
Burr, Arthur H. ('30; '41) (NBJA), Ch. Engr., Project 22, Northwestern Tech. Inst.; *for mail,* 2750 Prairie Ave., Evanston, Ill.
Burr, Kenneth George (J'45) (EKS), Ensign, U.S.N.R.; King Philip Rd., South Sudbury, Mass.
Burress, Lloyd F. ('17; '39), Div. Supt., Coke Plant & Blast Furnaces, Gary Steel Works, Carnegie-Ill. Steel Corp., Gary, Ind.
Burrill, Harold Griswold ('18; '21; '28) (FKORSZ), Owner, Firm of Harold G. Burrill, Cons. Engr., 810 Keyser Bldg., Baltimore; *for mail,* Quarters 56, Aberdeen Proving Ground, Md.
Burris, Winston Durr (J'35) (JWAM), Research Analyst, Engr., Menasco Mfg. Co., 805 S. San Fernando Rd.; *for mail,* 423 Bethany Rd., Burbank, Calif.
Burroughs, Edwin E. (J'30) (NMGDC), Design Engr., St. Regis Paper Co., 230 Park Ave., New York 17, N.Y.
Burrows, John Rexford (J'41) (CAMG), Quality Control & Salvage Engr., Ansco Div., Gen. Aniline & Film Co., Binghamton; *for mail,* Box 464, Greene, N.Y.
Burrows, Lewis Manley ('44) (BESJMC), Asst. Ch. Engr., Manning, Maxwell & Moore, Inc., 5 Watson St., Boston; *for mail,* 60 Walker St., North Quincy 71, Mass.
Burrows, Robert Jay ('14) (R), V.P., Clark Equip. Co., Battle Creek, Mich.
Bursack, William Wright (J'44) (HRS), Turbine Engr., Gen. Elec. Co., Fitchburg, Mass.
Bursley, Joseph Aldrich ('06; '10) (CEFS), Prof., Dean of Students, Univ. of Mich.; *for mail,* 2107 Hill St., Ann Arbor, Mich.
Burstadt, Erwin William (J'41) (SMNCBJ), Engr., Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 52 Chestnut St., East Saugus, Mass.
Burstein, Sol (J'45) (SBKNEO), Asst. Engr., Shipbldg. Div., Bethlehem Steel Co., 97 E. Howard St., Quincy 69; *for mail,* 198 Chester Ave., Chelsea 50, Mass.
Burt, Clayton Raymond ('09) (BHM), Chmn. of Bd., Niles-Bement-Pond Co., Charter Oak Blvd., West Hartford 1, Conn.
Burt, Lawrence Strite (J'40) (SKF), Ch. Results Engr., Pub. Serv. Co. of Colo., Box 79; *for mail,* 932—14th St., Boulder, Colo.
Burtenshaw, Charles David ('22; '28; '35) (DLMN), Dir., Pulverizing Mchy. Co., 10 Chatham Rd., Summit, N.J.
Burton, Cuthbert Clarence (J'42) (SLCS), Mech. Engr., Dow Magnesium Corp., Velasco; *for mail,* 214 W. Pyron Ave., San Antonio 4, Tex.
Burton, Raymond Charles ('27) (SEFK), Sr. Engr., Power Div., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Burton, W. Dean ('14) (BCDMOF), Cons. Engr., 431 S. Allen Ave., Pasadena 4, Calif.
Burton, Walter McKinley (J'45) (CNMDBZ), Mech. Control Engr., Oneida, Ltd., Oneida, N.Y.
Burtt, Nelson Walter ('23; '33; '35) (ZLBCTR), Application Engr., Foxboro Co., Neponset Ave., Foxboro; *for mail,* 43 High St., Sharon, Mass.
Burwell, John Spotswood (J'43) (RHSTLN), Asst. Engr., Allis-Chalmers Mfg. Co., Milwaukee; *for mail,* 2061 S. 73rd St., Milwaukee 14, Wis.
Burwell, Robt. T. ('01) (BFS), Mgr., New Orleans Dept., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford, Conn.; *for mail,* 7911 Freret St., New Orleans, La.
Burzynski, Larry Edward ('43) (DNMJSZ), Mech. Engr., Research & Devel. Div., Army Serv. Forces, Office of the Ch. of Engrs., Army War College, Washington, D.C.; *for mail,* 1505 Wayne St., Alexandria, Va.
Busch, Edwin John, Jr. (J'46) (CLO), 1317 S. 79th St., Omaha 4, Neb.
Busch, Frank ('32; '35) (LNBDOS), Engr., Gen. Am. Process Equip. Div., Gen. Am. Trans. Co., 420 Lexington Ave., New York 17; *for mail,* Chicago Ave., Massapequa, N.Y.
Buschmann, Paul Thomas (J'39) (BDJKMN), Plant Engr., Am. Radiator & Stand. Sanitary Corp., Foot of E. 46th St., Bayonne; *for mail,* 62 Dana Pl., Englewood, N.J.
Busck, Paul G. ('35; '35), Asst. Ch. Draftsman, Lehigh Portland Cement Co., 718 Hamilton St., Allentown, Pa.
Bush, Arthur Ogden (J'41) (CMA), Electrician's Mate, 2nd Class, J.C.A. Navy 926, Box 20, F.P.O., San Francisco, Calif.; *for mail,* 5047 N. Lawndale Ave., Chicago 25, Ill.
Bush, B. H. (J'37), Lt.; Box 141, Charleston, S.C.
Bush, George F. (J'40) (OZBA), Cons. Mech. Engr., 120 Prospect Ave., Princeton, N.J.
Bush, Harvey W. (J'36) (CDM), Edw. G. Budd Mfg. Co., Red Lion & Veree Rds., Philadelphia, Pa.
Bush, R. M. (J'37), 1st Lt., Ord. Dept., Ord. Replacement Trng. Center, D.M.T. Office, Aberdeen Proving Ground, Md.
Bush, Ralph Raymond ('42) (BJANYE), Devel. Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia 42; *for mail,* 28 Orchard Rd., Havertown, Pa.
Bush, Richard Tabor ('39; '43) (BLS), Head, Test & Inspc. Div., Carbide & Carbon Chemicals Corp., Box 2831, Charleston 30, W. Va.
Bush, Vannevar (Non-Member), *Holley Medallist,* '43; Dir., Office of Scien. Research & Devel.; Pres., Carnegie Instn. of Washington, Washington 5, D.C.
Bush, William Wallace (J'42) (COGDJM), V.P., Gen. Mgr., Lloyd Engrg. Co., 55 Stephen St., Belleville 9, N.J.
Bushfield, Frank T. (A'22), 46 Orchard Dr., East Williston, L.I., N.Y.
Bushley, Harry Richard ('26; '33; '35) (SRCKE), Dist. Mgr., Elliott Co., 1101 Vance Bldg., Seattle 1, Wash.
Bushman, Andrew Kidd ('45) (CLRSA), Mgr., Application & Serv. Engrg. Divs., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Bushman, William (J'41), Westinghouse Elec. Corp., Pittsburgh; *for mail,* 517 Holmes St., Pittsburgh 21, Pa.
Bushnell, Cornelius Huntington, Jr. (J'44), 6043 Eastwood Terrace, Norfolk, Va.
Bushnell, Frederic Newell ('91; F'37), *Manager,* '17-'20; V.P., Dir., Stone & Webster Engrg. Corp., 49 Federal St., Boston 6; *for mail,* 20 Chapel St., Brookline 46, Mass.
Bushnell, Howard Parker (J'45), Sales Engr., McCarthy & Robinson, Ltd., 2 Neville Pk. Blvd., Toronto 8, Ont., Can.
Bushong, Robert J. (J'39) (BDO), Engr., Charge of Design, Insley Mfg. Corp., Olney St., Indianapolis; *for mail,* 104 N. Broadway, Greenfield, Ind.
Busler, Karl Maxton ('43) (CSFKJ), Steam Serv. Supvr., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4; *for mail,* 107 Woodlyn Ave., Glenside, Pa.
Bussenius, Wilbert Otto (J'43) (RSC), Sales Engr., Gustin-Bacon Mfg. Co., 310 S. Michigan Ave., Chicago; *for mail,* 6 N. Pine St., Mt. Prospect, Ill.
Butcher, Alfred ('24) (SFE), Steam Engr., Gulf Oil Corp., Gulf Bldg., Pittsburgh 22, Pa.
Butcher, Ira A. ('18; '24) (CDLOSZ), Plant Engr., Charge Design & Constr., Calco Chem. Div., Am. Cyanamid Co., Piney River, Va.
Butcher, Joseph Hubert ('35; '45) (CBMNSL), Engr., Aetna Casualty & Surety Co., 810 S. Spring St., Los Angeles; *for mail,* 2249 Redondo Blvd., Los Angeles 16, Calif.
Butcher, Robert Oliver (J'45), Asst. Engr., Gen. Elec. Co., River Works, 920 Western Ave., Lynn; *for mail,* 400 Puritan Rd., Swampscott, Mass.
Butler, Clarence Albert, Jr. (J'40) (LCFSKY), Dir., Planning Staff, Diamond Alkali Co., Box 430; *for mail,* 1009 Mentor Ave., Painesville, Ohio.
Butler, Ernst ('22; '35), Engr., Charge Design, Wayagamack Div., Consltd. Paper Corp., Ltd.; *for mail,* 530 St. François Xavier St., Three Rivers, Que., Can.
Butler, Frank Anthony ('19; '35) (EHM), Mar. Diesel Engr., Fairbanks, Morse & Co., 1713 N. Market, Dallas 2; *for mail,* 4136 Bryn Mawr, Dallas 5, Tex.
Butler, Frank Joseph (J'42) (BMS), Mech. Engr., Plant Research Gp. 3, Tech. Sec., E. I. du Pont de Nemours & Co., Box 993, Charleston, W. Va.
Butler, Harry M. ('19), Mech. Engr., Union Twist Drill Co.; *for mail,* 95 Highland Ave., Athol, Mass.
Butler, Henry W. ('24) (EFHKOS), Partner, Sanderson & Porter, 52 William St., New York 5; *for mail,* 150 E. 73rd St., New York 21, N.Y.
Butler, Howard Wallace (J'40) (ABNOZ), Exper. Engr., New Britain Mch. Co., New Britain, Conn.
Butler, James Cranford (J'36) (SFE), Inspr., Boilers & Mchy., Hartford Steam Boiler Inspc. & Ins. Co., 1904-12 Caren Tower, Cincinnati; *for mail,* 2019 Scioto Trail, Portsmouth, Ohio.
Butler, John Bailey (J'43) (MCKHW), Engr., Kimberly-Clark Corp., Neenah; *for mail,* 519 W. 7th St., Appleton, Wis.
Butler, Joseph Parker (J'36) (ABJCDH), Engr., Boeing Aircraft Co., Seattle; *for mail,* 306 Blanchard St., Seattle 1, Wash.
Butler, Leonard Court ('44) (CJM), V.P., Charge Sales & Engrg., Fullerton Mfg. Corp., 17 Chestnut St., South Norwalk; *for mail,* 169 Mason St., Greenwich, Conn.
Butler, Robert Boone (J'37) (CBHASF), Serv. Mgr., Oliver Corp., 300 Lawler St., Charles City, Iowa.
Butler, Robert Moore (J'40) (EMSO), Co. Cmdr., U.S.M.C., 1st Separate Engr. Bn., Fleet Mar. Force, F.P.O., San Francisco, Calif.; *for mail,* 524 Canal St., Yazoo City, Miss.
Butler, William Edward ('44), Sales, Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 22 E. 38th St., New York 16, N.Y.
Butt, Howard ('22; '23) (SCEYH), V.P., Wm. Powell Co., 50 Church St., New York, N.Y.; *for mail,* 1 Spring Brook Rd., Morristown, N.J.
Butterfield, Thomas Edward ('12) (ESFA), Emeritus Prof. Mech. Engrg., Lehigh Univ.; *for mail,* 1736 W. Union Blvd., Bethlehem, Pa.
Butterworth, John G. ('42) (SEOKHN), Engr., W. P. Nevins Co., 53 W. Jackson Blvd., Chicago; *for mail,* 1022 Maple Ave., Evanston, Ill.
Buttron, Willard C. ('36) (RSEJW), Asst. Supt., Constr. & Repairs, N. Y. Cent. R.R., Mar. Repair Shop, Weehawken; *for mail,* 1258 Emerson Ave., West Englewood, N.J.
Butts, Edgar (J'41), Time Study, Aluminum Co. of Am., 5151 Alcoa Ave., Los Angeles; *for mail,* 6927 Cedar St., Huntington Park, Calif.
Butz, George Alfred, Jr. (J'43) (BJA), U.S. Army; 242 N. Jefferson, Canonsburg, Pa.
Butze, Richard John (J'44) (CMDNZK), Lt. (j.g.), U.S.N.R., LST-119, F.P.O., San Francisco, Calif.; *for mail,* 7 Windsor Rd., Scarsdale, N.Y.
Buvinger, George A. ('01; '04), Retired; 971 Harvard Blvd., Dayton 6, Ohio.
Buxbaum, William ('43) (CMO), Partner, Johns-Hartford Tool Co., 390 Capitol Ave., Hartford 5; *for mail,* 125 Livingston St., New Haven 11, Conn.
Buxton, Paul H. ('31; F'45) (BDS), Ch. Engr., West. Cartridge Co., East Alton; *for mail,* 702 Euclid Pl., Alton, Ill.
Buyers, Archie Stanton ('12; '25) (CBD), Col., 1202 W. 3rd St., Sterling, Ill.
Bye, Norman Charles ('27; '35) (WNMDCJ), Ch. Engr., Henry Disston & Sons, Inc., Tacony, Philadelphia 35, Pa.
Byer, Henry Edward ('19) (HJL), Vacuum Equip., 69 Mt. Airy Rd., Bernardsville, N.J.
Byerley, Thomas Evans (J'41) (BHS), Mech. Draftsman, Ga. Power Co., 75 Marietta St.; *for mail,* 1296 Oxford Rd., N.E., Atlanta, Ga.
Byers, Harry Richard ('19; '25) (EFHKS), Pres., Harry R. Byers, Inc., 707 U.S. Natl. Bank Bldg., Denver, Colo.
Bynum, Edwin A., Jr. ('20; '26; '35) (LBCSH), Ch. Engr., Ingleside Refinery, Humble Oil & Refining Co.; *for mail,* Box 512, Ingleside, Tex.
Byrne, Harry Ellis ('44) (BHNKE), Plant Engr., Taylor Forge & Pipe Works, 4735 W. 14th St., Cicero 50; *for mail,* 5317 Grand Ave., Downers Grove, Ill.
Byrne, Hugh Joseph ('42) (OMLD), Engr. with Leland S. Rosener, Engrs., 12th Floor, 233 Sansome St., San Francisco 4, Calif.
Byrne, James Joseph (J'38) (BJNZ), Assoc. Mech. Engr., Ord. Research & Devel. Center; *for mail,* Civilian War Housing Dormitories, Aberdeen Proving Ground, Md.
Byrne, Thomas Howard (J'44) (MJBA), Tool Engr., N. Am. Aviation, Inc., 5601 W. Imperial Blvd., Inglewood; *for mail,* 514 N. June St., Los Angeles 4, Calif.
Byrne, William Henry ('44), Cons. Engr., 80 Wall St., New York 5; *for mail,* 47-21—41st St., Long Island City, N.Y.
Byron, James L. ('23; '35) (CJLMNY), Engr., Underwood Corp., 581 Capitol Ave., Hartford 6, Conn.
Bystrom, John H. (J'44) (ACOMHB), Products Engr., Kellett Aircraft Corp., Lansdowne Ave.; *for mail,* 37-B Stephan Court, Millbourne Gardens, Upper Darby, Pa.
Bywater, L. Gordon (J'43) (SFCMZ), Lt. (j.g.), U.S.N.R., 833 N. 1 West St., Salt Lake City, Utah.

C

Cable, H. Edward (J'37) (CRS), Welding Engr., Lincoln Elec. Co., 926 Manchester Blvd., Pittsburgh 12, Pa.
Cabrera y Nevraumont, Jose Maria ('44) (OLJKSR), Owner, Mgr., Firm of Jose M. Cabrera, Engr., Cinco de Mayo 32-416, Mexico, D.F., Mex.

Cadwallader, Harry, Jr. ('13), Pres., Stand. Shop Equip. Co., 82nd & Tinicum Ave.; *for mail,* 1413 W. Somerset St., Philadelphia, Pa.

Cadwallader, Lewis W. ('34; '41) (SFE), Asst. Plant Engr., Potomac Elec. Power Co., 1st & V Sts., Washington, D.C.; *for mail,* 609 Gist Ave., Silver Spring, Md.

Cady, Cecil I. ('22) (DLO), Cons. Engr., 101 Park Ave., New York 17, N.Y.

Cady, George Henry ('29; '35) (TSFN), Plant Engr., Cranston Print Works, Inc., Cranston St., Cranston; *for mail,* 73 Roslyn Ave., Providence 8, R.I.

Cady, George Leonard (J'44) (OTS), Mech. Engr., Design & Field Supervision, Dauray & Bardsley Inc., 1 Front St., Woonsocket; *for mail,* 73 Roslyn Ave., Providence 8, R.I.

Cady, Harrison R. (J'16) (HFSDJK), Mech. Engr., Hackensack Water Co., 4100 Park Ave., Weehawken; *for mail,* 367 Maitland Ave., West Englewood, N.J.

Cadzow, Murray ('21; '26) (EFHKSZ), Mech. Engr., Pub. Bldgs. Admin., 731 Custom House, New York 4; *for mail,* 89-07—107th St., Richmond Hill 18, L.I., N.Y.

Cafiero, Dominick Francis ('40; '45) (BCD), Naval Arch., Naval Shipyard, Philadelphia; *for mail,* 222 Bridge St., Drexel Hill, Pa.

Cagin, Harry (J'36) (J), Tool & Die Engr., Superior Die Casting Co., 17325 Euclid Ave., Cleveland 12; *for mail,* 13994 Mont Ave., East Cleveland 18, Ohio.

Cahill, Edward Howard ('14; '18) (NBALOC), 24 Tenmore Rd., Haverford, Pa.

Cahill, John E. ('18; '35), Partner, John E. Cahill Co., 342 Madison Ave., New York, N.Y.

Cahn, Raphael David (J'44), 87-18 Rockaway Beach Blvd., Rockaway Beach, N.Y.

Cain, Basil S. ('35) (RBAEN), Asst. Engr., Loco. Div., Gen. Elec. Co., 2901 E. Lake Rd., Erie 2, Pa.

Cain, John Neal ('42) (RSGFEC), Loco. Eng. Man, Pa. R.R., Philadelphia, Pa.; *for mail,* Old Mill House, Kingsville, Ohio.

Caine, J. F. K. ('44) (CGTMDF), Managing Dir., Bennetts Laundry & Dry Cleaners, Grosvenor Rd.; *for mail,* 20 Sandringham Dr., Wallasey, Cheshire, England.

Caine, William Phillips ('99; '04; '18) (SFJ), Retired; 2215 Ave. H, Birmingham 8, Ala.

Cala, Chas. F. (J'34), Foreman, Charge Glass Making, Bausch & Lomb Optical Co., 635 St. Paul St.; *for mail,* 144 Hoover Rd., Rochester, N.Y.

Calabretta, Peter Trent (J'45) (NMCB), Mch. Designer, Am. Mch. & Fdy. Co., 5502—2nd Ave., Brooklyn; *for mail,* 2111 E. 3rd St., Brooklyn 23, N.Y.

Calder, Augustus W., Jr. ('32; '38) (CJM), Asst. Treas., New England Butt Co., 304 Pearl St., Providence 7; *for mail,* 434 Brook St., Providence 6, R.I.

Caldwell, Eugene ('28; '33) (CMJSEF), V.P., Gen. Mgr., Hyster Co., 2902 N.E. Clackamas St., Portland 8, Ore.

Caldwell, Eugene C. (J'41) (CWMY), Div. Mgr., Molded Plywood Div., Evans Products Co., 15310 Fullerton St., Detroit 27, Mich.

Caldwell, Harold Warren (J'44) (BHS), Lt., Army Air Forces, Base Weather Sta., Suisan-Fairfield Army Air Field, Fairfield, Calif.

Caldwell, James Ian MacLaren ('45) (APY), Field Engr., Mech. Div., U.S. Rubber Co., 317 S. Detroit, Tulsa; *for mail,* 2617 N.W. 29th St., Oklahoma City 7, Okla.

Caldwell, Wm. E. ('18; '22; '26) (EFS), *Melville Medallist,* '33; Mech. Plant Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York, N.Y.

Calhoun, Joel Newton ('45) (BJM), Air Conditioning Serv. Engr., Tech. Data Dept., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass.

Calkin, Edwin Daniel (J'33) (BM), Owner, Mgr., Elf Eng. Co., 206 S.W. Washington, Portland 4; *for mail,* Box 820, Route 10, Milwaukie 2, Ore.

Calkins, George B., Jr. (J'40), Mech. Engr., Fuels Sec., Automotive Div., Research & Devel. Labs., Socony-Vacuum Oil Co., Paulsboro; *for mail,* 73 N. Evergreen Ave., Woodbury, N.J.

Call, Leroy J. ('21) (CGMO), Mgr. Engrg., Carborundum Co.; *for mail,* 925 Maple Ave., Niagara Falls, N.Y.

Callahan, Joseph G. ('22; '26) Prod. Supt., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York, N.Y.; *for mail,* 887 Belle Ave., Teaneck, N.J.

Callahan, Vincent T. ('30) Reserve Power Plant Engr., Bell Tel. Labs., Inc., 463 West St., New York, N.Y.; *for mail,* Lauvir Court, 143 Tenafly Rd., Englewood, N.J.

Callahan, William John ('34; '42) (CFS), Maj., Inf., U.S.A., 527th Armd. Inf. Bn., Ft. Knox, Ky.; *for mail,* 343—101st St., Brooklyn, N.Y.

Callan, John ('18; '22; '35) (FSHD), Supt., Power Div., Tenn. Eastman Corp., Kingsport, Tenn.

Callaway, Clarence R. ('21) (BCDJN), 120 Cabrini Blvd., New York 33, N.Y.

Callouette, John Edward (J'45) (BOS), Asst. Ch. Power Plant Engr., Indus. Rayon Corp., Painesville, Ohio.

Calman, Joshua (J'45) (CJMYNT), Plant Engr., Protex Products Co., 44 Sherman Ave., Jersey City 7; *for mail,* 553 Devon St., Kearny, N.J.

Calnan, Edw. J. ('30; '38) (FKS), Power Engr., Ont. Paper Co., Ltd., Thorold, Ont., Can.

Calongne, Robert Joseph (J'46), Box 268, Lake Charles, La.

Calvet, Albert Marcel (J'40) (AKS), High St., Monroe, N.Y.

Cambou, Ernest J. (J'40), Engrg. Dept., Gen. Elec. Co., 212 N. Vignes St., Los Angeles; *for mail,* 6 Hill St., Mill Valley, Calif.

Camby, John Joseph (J'41) (KNBFOS), Technician, 5th Grade, Draftsman, G-3 Sec., TSFET, A.P.O. 757, c/o Postmaster, New York, N.Y.; *for mail,* 743 Orchard Ave., Avalon, Pittsburgh 2, Pa.

Cameron, Hugh S. ('25; '43) (EFSKOM), Asst. Prof. Mech. Engrg., Dept. Mech. Engrg., Rice Inst., Houston 1, Tex.

Cameron, John A. ('32) (CMDN), Retired; 68 Sumner St., Hartford 5, Conn.

Cameron, John Richardson (J'40) (SFEKCZ), Results Engr., E. I. du Pont de Nemours & Co., 626 Schuyler Ave., Arlington, N.J.

Cameron, William Charles (J'43), Avenal Div., Pacific Pipeline Constr. Co., Avenal; *for mail,* 119 Jackson St., Taft, Calif.

Cammann, Oswald ('37; '43) (NSMCL), Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge; *for mail,* 235 Conant Rd., Weston 93, Mass.

Cammen, Matthew Michel (J'38) (NEB), Engr., Ingersoll-Rand Co., Painted Post; *for mail,* 129 Pritchard Ave., Corning, N.Y.

Cammer, Moses (J'43) (CJMZNA), Partner, Gen. Engrg. Works, 260 West St., New York; *for mail,* 320 E. 53rd St., New York 22, N.Y.

Camp, E. V. ('17; '35), Pres., E. V. Camp & Associates, Inc., 215 Moreland Ave., N.E., Box 62, Sta. E, Atlanta, Ga.

Camp, George Dashiell ('30) (ORSWCL), Cons. Engr., Ejido 7-507; *for mail,* Apartado Postal 1005, Mexico, D.F., Mex.

Camp, Louis Foster, Jr. (J'36) (OCJMGB), Asst. Gear Engr., Camera Works, Eastman Kodak Co., 333 State St., Rochester 4; *for mail,* 106 Edgewood Ave., Rochester 10, N.Y.

Camp, Wilmer E. ('22), Owner, Mgr., W. E. Camp Co., 1219—28th St., Sacramento, Calif.

Campana, James A. (J'41) (FJS), 801 Cowan Ave., Jeannette, Pa.

Campanale, Anthony Daniel (J'44), Pvt., 128 Charles St., Akron 4, Ohio.

Campbell, Alan McCleave (J'41) (CDL), Mech. Designer, Television Div., Raytheon Mfg. Co., Foundry Ave., Waltham 54; *for mail,* 29 Hidden Rd., Weston, Mass.

Campbell, Alexander B. (J'44) (BHM), 119 Wellington Rd., Upper Darby, Pa.

Campbell, Alexander L. ('21) (DKL), Cons. Chem. Engr., H. K. Ferguson Co., Cleveland; *for mail,* 2128 Rossmoor Rd., Cleveland Heights, Ohio.

Campbell, Bruce W. (J'44) (BYA), Design Engr., Lord Mfg. Co., 1635 W. 12th St.; *for mail,* 163 W. 6th St., Erie, Pa.

Campbell, Charles Langdon ('43), Asst. Ch. Engr., Diamond Alkali Co.; *for mail,* Box 563, Painesville, Ohio.

Campbell, Clarence B. ('29; '35) (SBNJCE), Mgr. Engrg., Steam Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 8 College Ave., Swarthmore, Pa.

Campbell, Colin Gordon (J'36) (CJDSM), Ch. Engr., Fahralloy Can. Ltd., 95 Barrie Rd.; *for mail,* 20 Neywash St., Orillia, Ont., Can.

Campbell, David S. ('32; '37) (LCDKMN), Mech. Engr., Campbell Soup Co., 100 Market St., Camden; *for mail,* 201 Elm Ave., Haddonfield, N.J.

Campbell, Donald ('14) (BHJZYG), Mem., Morrison, Kennedy & Campbell, 2300 Graybar Bldg., 420 Lexington Ave., New York 17, N.Y.

Campbell, Donald Matthew (J'46) (ER), Lt. Comdr., 880 N. 22nd St., Philadelphia 30, Pa.

Campbell, E. Clarke (J'41) (AEP), Capt., 546th Sqd., 384th Bomb Gp., A.P.O. 557, New York, N.Y.

Campbell, Edmund Dana ('16) (RM), V.P., Charge Engrg., Am. Car & Fdy. Co., 30 Church St., New York 7, N.Y.

Campbell, F. Glenn (J'45), 22 Suttle Ave., Lowell, Mass.

Campbell, Frank Dale ('44), Cons. Engr., 8951 Lowe Ave., Chicago 20, Ill.

Campbell, Frank Roger (J'42), Asst. Prof. Mech. Engrg., Colo. Sch. of Mines; *for mail,* Box 88, Golden, Colo.

Campbell, George Eugene ('22) (HWMBNC), Ch. Engr., Wheland Co., Broad St., Chattanooga 2; *for mail,* 306 Sunnyside Dr., Chattanooga 4, Tenn.

Campbell, George Willard (J'36) (MFECK), Engr., with Thomas H. Urdahl, Cons. Engr., Washington; *for mail,* 325 A St., S.E., Washington 3, D.C.

Campbell, Gordon ('91; '14), Box 293, York, Pa.

Campbell, Gordon M. ('06) (BMS), Dir., Charge of Mfg., British Thomson-Houston Co. Ltd.; *for mail,* Bosworth Old Hall, Husbands Bosworth, Rugby, Warwicks, England.

Campbell, Howard Eslie (J'40) (ACMNH), Tool Engr., John Deere Tractor Co.; *for mail,* 432 Belmont Ave., Waterloo, Iowa.

Campbell, Hugh Thurston (J'43), 205 Ohio St., Baytown, Tex.

Campbell, James Alan ('37; '43) (BNRHZ), Asst. Dist. Engr., Westinghouse Air Brake Co., 1101 Matson Bldg., San Francisco 5, Calif.

Campbell, James H. (J'35) (BJM), Research & Devel. Dept., Engrg. Sec., Naval Ord. Test Sta., Inyokern, Calif.

Campbell, James P. (J'40) (AE), Lt., U.S.S. *Melville,* F.P.O., New York, N.Y.; *for mail,* 3229 Orange Ave., Long Beach, Calif.

Campbell, James R. ('32) (EBHJAK), Design Engr., Lombard Governor Corp., Ashland; *for mail,* 16 Priest Rd., Watertown 72, Mass.

Campbell, James Samuel, Jr. (J'34) (JMNCL), Instr., Mech. Engrg. Dept., Rensselaer Poly. Inst., Troy, N.Y.

Campbell, Jason L. (J'40), Capt., Student Officers Sec., Class 44-4-A, Army Air Base, Hendricks Field, Fla.

Campbell, Jesse Gordon (J'41) (ACJ), 1411 S. Chilton, Tyler, Tex.

Campbell, Jesse Marcus ('29; '45) (FKSZ), Assoc. Prof. Mech. Engrg., Mich. State College, East Lansing, Mich.

Campbell, John, Jr. (J'41) (SFKHZE), Tech. Specialist, Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 667 E. 232nd St., Bronx 66, N.Y.

Campbell, John Malcolm ('45), Research Engr., Gen. Motors Research Div., 485 W. Milwaukee Ave., Detroit 2, Mich.

Campbell, L. Barrett ('21) (NHAC), Cons. & Design Engr., Am. Chain & Cable Co., Inc., Connecticut Ave., Bridgeport; *for mail,* 186 Hillside Ave., Waterbury 20, Conn.

Campbell, Lawrence George (J'45) (ANMBCH), Asst. Ch. Engr., Lion Match Co., 22-15—43rd Ave., Long Island City; *for mail,* 23 Wall St., Farmingdale, N.Y.

Campbell, Lester ('19) (TBMC), Research Engr., Foster Mch. Co., S. Broad St.; *for mail,* 74 Franklin St., Westfield 4, Mass.

Campbell, Levin Hicks, Jr. ('44; H'44) (CDNSHM), V.P., Internatl. Harvester Co., 180 N. Michigan Ave., Chicago, Ill.

Campbell, Marvin Ray (J'42), Maxton Oil & Fertilizer Co., Maxton, N.C.

Campbell, Oliver F. ('29; '35), Combustion Engr., Sinclair Refining Co., East Chicago, Ind.

Campbell, Peter F. (J'41) (FJK), 433 Lawrie St., Perth Amboy, N.J.

Campbell, Robert Douglas ('35) (ESRH), Maj., Ord. Dept., Army Serv. Forces, Officer in Charge, Dallas Office, St. Louis Ord. Dist.; *for mail,* 4801 Riverview St., St. Louis 20, Mo.

Campbell, Robert James (J'43) (ABCMNO), Lt., U.S.N.R., Engr., Naval Air Facility, Naval Air Matl. Center, Bldg. 537, Naval Base Center; *for mail,* 2324 Geary Terrace, Philadelphia 12, Pa.

Campbell, Roger P. (J'39) (CBMFJK), Ensign, U.S.N.R.; *for mail,* 181 Commonwealth Ave., Boston, Mass.

Campbell, Thomas D. ('40), Col., University Club, Washington, D.C.

Campbell, Tristram Joseph ('22; '26; '35) (CMOAL), Gray Gables, Odana Ave., Old Orchard Beach, Me.

Campion, William Kingsley (J'44) (CHJKO), Lt., Corps of Royal Canadian Elec. & Mech. Engrs., Officer in Comd., Light Aid Det. (Vehicles), 1st (Hdq.) Co., Hdq., Military Dist. 1, London; *for mail,* 102 Queen St., St. Catharines, Ont., Can.

Canan, William Dean ('21; '26) (EFLS), Project Mgr., Rust Engrg. Co., Clark Bldg., Pittsburgh 22, Pa.

Canavan, Harold Mathew (J'44) (SJHLB), Supvg. Inspr., Mutual Boiler Ins. Co. of Boston, 431 Main St., Cincinnati 2; *for mail,* 2978 Mapleleaf Ave., Cincinnati 13, Ohio.

Canavan, James Edward (J'38) (CMDJ), Matls. Engr., Supvr. Shipbldg., U.S.N., East Howard St., Quincy; *for mail,* 65 Aberdeen Rd., Milton 87, Mass.

Canavan, William F. ('20) (LR), Pres., Leader Iron Works, Decatur, Ill.

Canby, Harry Beaver ('04; '17), Retired; 528 Belmont Park N., Dayton 5, Ohio.

Candee, Allan H. ('20; '28) (BN), Mech. Engr., Gleason Works, 1000 University Ave., Rochester 3; *for mail,* 404 Hillside Ave,, Rochester 10, N.Y.

Candee, Andrew Haley ('43) (RE), Trans. Engr., Westinghouse Elec. Corp., East Pittsburgh, Pa.

Candee, Frank W. ('26; '35) (BEF), Asst. Prof. Mech. Engrg., Mech. Engrg. Dept., State College of Wash., Pullman, Wash.

Candlish, Fairlie (J'39) (FJ), Flight Lt., Royal Can. Air Force, Royal Air Force; *for mail,* 102 Kenmore Blvd., Hamilton Beach, Hamilton, Ont., Can.

Caney, F. Wheeler (J'41), 327 Casey Ave., Richland, Wash.

Canfield, Bailey L. ('46), Asst. Ch. Engr., C. G. Conn, Ltd., Elkhart, Ind.

Cangialosi, John Camillo (J'43) (SRL), Supvr., Elec. Drafting Dept., Chem. Constr. Co., 350—5th Ave., New York 1, N.Y.; *for mail,* 3460 Hudson Blvd., Jersey City, N.J.

Cannady, Earl Edgar (J'43) (KBA), 714—5th St., Monett, Mo.

Canning, Wylie Ewart (J'45) (JYECSL), Inspc. Engr., St. Clair Processing Corp., Ltd., Sarnia, Ont., Can.

Cannizzaro, Salvatore (J'39) (JLM), Assoc. Matls. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 1249—65th St., Brooklyn 19, N.Y.

Cannon, Arthur Hildreth ('39) (AFKS). Supt. Power, Carbide & Carbon Chem. Corp., South Charleston; *for mail,* 334 Hawthorne Dr., Charleston, W. Va.
Cannon, C. Newton ('36; '45) (SA), Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 6 Stedman St., Wakefield, Mass.
Cannon, James P. (J'28) (CLW), 61 Ellen Dr., Buffalo 21, N.Y.
Cannon, John Bernard, Jr. ('41) (CNMJW), Design Engr., Stapling Mchs. Co., Rockaway, N.J.
Cant, David Arthur ('42) (CDJ), Partner, Mid-Continent Transmission Co., 3012 E. 15th St., Tulsa 14, Okla.
Cantley, William Irvine ('40) (RFEBNC), Retired; 433 Moran Rd., Grosse Pointe Farms, Detroit 30, Mich.
Cantrell, C. M. (J'43), Apt. 21, 2724 E. Magnolia Ave., Knoxville, Tenn.
Cantwell, John William (J'46) (ACL), Capt., Army Air Forces; 155 S. Broad St., Trenton, N.J.
Canty, Daniel Joseph, Jr. (J'45) (CDJ), Works Indus. Engr., Columbia Steel Co.; *for mail,* 1350 Birch St., Pittsburg, Calif.
Canvasser, Marvin A. (J'44) (AEJ), 1st Lt., U.S.A.; *for mail,* 18231 San Juan Dr., Detroit, Mich.
Capa, Germi Salih (J'45), Besiktas, Capamarka, Istanbul, Turkey.
Capelle, Ernest Andrew ('44) (CZL), Gen. Sales Mgr., Taylor Instrument Cos. of Can. Ltd., 110 Church St., Toronto 1; *for mail,* 48 Princeton Rd., Toronto 9, Ont., Can.
Capo, Joseph James (J'36) (JM), Mar. Engr., Design Sec., N.Y. Naval Shipyard, Brooklyn; *for mail,* 1922 Greene Ave., Brooklyn 27, N.Y.
Caporaso, Joseph Robert (J'44), 41 Hudson St., Dover, N.J.
Capper, Michael Arthur (J'43) (DHSRME), Engr., Meakins & Sons Ltd., 125 Wellington St. N.; *for mail,* 314 Main St. E., Hamilton, Ont., Can.
Capps, Frank Russell ('44) (BJM), V.P., C. I. Capps Co., Inc., Box 5477; *for mail,* 1838 Brookwood Rd., Jacksonville 7, Fla.
Capps, Parker Charles ('43) (CDJMNO), Pres., Gen. Mgr., C. I. Capps Co., Inc., 1727 Bennett St., Jacksonville 6, Fla.
Capron, John Dodge ('26; '35) (C), Pres., Glamorgan Pipe & Fdy. Co., Upper Basin, Lynchburg, Va.
Capstack, Eugene Joseph (J'45) (NMBDCL), Design Engr., Joseph E. Seagram & Son. Inc., 7th St. Rd.; *for mail,* 2328 Gaulbert Ave., Louisville 10, Ky.
Caraco, Isaac Ralph (J'45) (CHS), Prin. Engr., Calif. Shipbldg. Corp., Box 966, Wilmington; *for mail,* 2930½ W. Santa Barbara Ave., Los Angeles 43, Calif.
Carbone, Joseph Enrico (J'45) (BNKHMA), Research Engr., Republic Aviation Corp., Farmingdale, L.I., N.Y.
Card, Frederic M. ('19), 116 Edna Ave., Bridgeport, Conn.
Card, Lawrence Baker ('43) (BHJ), Mech. Engr., Transmission & Sta. Engrg. Dept., Pub. Serv. Co. of Colo., 900—15th St., Denver 1; *for mail,* 1071 S. Emerson St., Denver 9, Colo.
Cardwell, Franklin D. (J'45) (BLMTZ), Indus. Engr., 110 W. 43rd St., New York 18, N.Y.; *for mail,* 2436 Jonquil St., New Orleans 17, La.
Carell, Walter Sigurd (J'31) (SCEBFL), Power Consultant, Fabrics & Finishes Div., E. I. du Pont de Nemours & Co., Parlin, N.J.; *for mail,* 72 Boyd St., Stapleton 4, S.I., N.Y.
Carey, Paul C. ('30) (O), Mem. Firm, Runyon & Carey, 33 Fulton St., Newark 2, N.J.
Carey, Robert Henderson (J'43) (BYWZ), Research & Devel. Engr., Bakelite Corp.; *for mail,* 729 Hawthorne Ave., Bound Brook, N.J.
Cargill, Walter N. ('12) (FS), 185 Devonshire St., Boston; *for mail,* 7 Woodland St., Arlington, Mass.
Carhart, Frank Milton ('28) (OCSHAN), Partner, Jackson & Moreland, Park Sq. Bldg., Boston 16, Mass.
Cariss, Carington Carysfort ('11; '18) (BWSOYN), Ch. Engr., Waterous Ltd., Market St.; *for mail,* 95 William St., Brantford, Ont., Can.
Carito, William Antonio (J'43) (ABSKH), Seaman, 2nd Class, Naval Repair Base, Trans. Office, Bldg. 255, New Orleans, La.
Carl, Robert Arthur (J'40) (ABRZCS), Lt., U.S.N.R., Head, Struc. Flight Test Sec., Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail,* 1407 N. Kenilworth St., Arlington, Va.
Carl, Wellington C. ('43) (DMOLC), Industry Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 603 Hampton Ave., Pittsburgh 21, Pa.
Carle, Edward J. (J'32) (CKSZDF), Secy., Carle-Boebling Co., Inc., 1641 W. Broad St., Richmond 20; *for mail,* 3309 Suffolk Rd., Richmond 22, Va.
Carleton, William Perkins (J'44), Test Engr., Ranger Aircraft Engines Div., Fairchild Eng. & Airplane Corp., Farmingdale, L.I.; *for mail,* 203 Bay Ave., Patchogue, N.Y.
Carlin, Joseph Alexander ('32) (CHMN), Box 186, Bala-Cynwyd, Pa.
Carlisle, Francis L. (J'39) (BNAR), Mech. Engr. P5, Naval Ord. Test Sta., 1030 Green, Pasadena 1; *for mail,* 5040 Stratford Rd., Los Angeles 42, Calif.
Carlisle, Morten ('29), Retired; 71 E. Hollister St., Cincinnati 19, Ohio.
Carliss, Oswald Sheldon ('44) (NMZDB), Ch. Engr., Scale Sec., Yale & Towne Mfg. Co., 4530 Tacony St., Philadelphia 24; *for mail,* 724 Charles St., Glenside, Pa.
Carlson, Albert R. (J'33), Engrg. Dept. (Metal Shop), Metro. Life Ins. Co., 1 Madison Ave., New York; *for mail,* Doris Ave., Northport, L.I., N.Y.
Carlson, Arnold Evert (J'45) (KMCNS), Ensign, U.S.N.R., U.S.S. *Princeton,* CV-37, F.P.O. New York, N.Y.; *for mail,* 1410 Boulevard, West Hartford, Conn.
Carlson, Axel F. ('32) (CMT), Pres., Gen. Mgr., A.B.A.C. Gustafson, Kungsgatan 18, Stockholm, Sweden.
Carlson, Bernard (J'38), Moline Tractor Works, Deere & Co., Moline, Ill.
Carlson, Bernhard M. (J'38) (J), Indus. Engr., Columbia Steel Co.; *for mail,* Box 50, Route 2, Pittsburg, Calif.
Carlson, Charles A. ('14; '35) (BNJL), Mech. Engr., John Deere Plow Works, 13th St. & 3rd Ave.; *for mail,* 2727—11th Ave. C, Moline, Ill.
Carlson, David (J'42) (HJM), Lt., U.S.N.R., Air Tech. Serv. Comd., Regional Office, 7100 Lonyo Ave., Detroit 32, Mich.
Carlson, Gordon Verner ('39) (SKF), 806 S. Lincoln Ave., Urbana, Ill.
Carlson, H. Maurice (J'45) (FBHZSO), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Carlson, Harold C. R. ('34; '41) (NMCJBD), Ch. Engr., Charles Fischer Spring Co., 749 Atlantic Ave., Brooklyn 17, N.Y.
Carlson, Harold W. (J'40), Tiskilwa, Ill.
Carlson, Harry ('21; '35) (YW), Tech. Rep., Bakelite Corp., 30 E. 42nd St., New York 17, N.Y.
Carlson, Harry N. (J'42) (NBJMZC), Ch. Draftsman, Stand. Gage Co., Inc., 70 Parker Ave.; *for mail,* 162 N. Clinton St., Poughkeepsie, N.Y.
Carlson, Hjalmar G. (Non-Member), *A.S.M.E. Medallist,* '21, *Holley Medallist,* '21; Retired; 15 Water St., Shrewsbury, Mass.
Carlson, John R. ('43) (SKEFHB), Cent. Sta. Steam Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia, Pa.
Carlson, Kenneth James (J'42) (AMJR), Ensign, U.S.N.R., Electronics Engr., Pilotless Aircraft Sec., Rm. 1W68, Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail,* Box 390, Riceville, Iowa.
Carlson, Paul Gustave (J'39) (EABJM), Mech. Design Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 42 Jefferson Rd., Wakefield, Mass.
Carlsrud, Reidar ('27; '35) (CLM), Reid-Avery Co., Baltimore 22, Md.
Carlsson, Carl Axel Victor ('05) (NS), Retired; Lanham, Md.
Carlsson, Ernest R., Jr. (J'40) (AS), 148 Nassau Ave., Huntington, L.I., N.Y.
Carlsten, Sigurd Vallentine ('45) (NYHLMJ), Staff Engr., Firestone Tire & Rubber Co., 1200 Firestone Pkwy., Akron 17; *for mail,* 1535 Overlook Dr., Akron 7, Ohio.
Carlton, J. R. ('18) (SFHZBK), Engr., Pub. Serv. Elec. & Gas Co., 200 Boyden Ave., Maplewood, N.J.
Carlzen, Carl F. (J'39) (NBMJ), Mech. Devel. Engr., Colonial Radio Corp., 1280 Main St., Buffalo 9; *for mail,* 29 Macamley, Buffalo 20, N.Y.
Carman, Charles Holland, Jr. (J'41) (SLCYZE), Sales Engr., Elliott Co., 225 Broadway, New York 7, N.Y.
Carman, Edwin S. ('17; F'37) (OCD), *President,* '21; Pres., Edwin S. Carman, Inc., 1643 Lee Rd., Cleveland 18, Ohio.
Carman, Everett Perry ('45) (FEK), Fuel Engr., Bur. of Mines, U.S. Dept. of Interior, Washington 25, D.C.
Carman, Thornton Salisbury ('44) (DLOMCZ), V.P., Ch. Engr., Edwin S. Carman, Inc., 1643 Lee Rd., Cleveland 18, Ohio.
Carmichael, Colin ('30; '35; '35) (NBMEYZ), Assoc. Editor, *Machine Design,* Penton Bldg., Cleveland 13, Ohio.
Carmichael, James Thomas (J'44) (NMBJ), Ch. Engr., Brewster Co., 50 Fannin St.; *for mail,* 927 Boulevard, Shreveport 21, La.
Carmody, John Vincent (J'38) (ABNMJY), Design Layout Engr., Hamilton Stand. Propellers Div., United Aircraft Corp., 400 S. Main St.; *for mail,* 525 Tolland St., East Hartford 8, Conn.
Carmoega, Enrique Rafael ('27) (RECAF), Supt., Trans. Div., Am. R.R. Co. of Porto Rico, Box 2552, San Juan 11, P.R.
Carnegie, Andrew ('21; '35) (CFS), Supt. Prod., Ohio Edison Co., 47 N. Main St., Akron 8, Ohio.
Carner, Frank (J'43) (MBCN), Spec. Engr., Inland Steel Co., Indiana Harbor; *for mail,* 4322 Ivy St., East Chicago, Ind.
Carnes, Herman W. ('29; 43) (EFLCSY), Area Supt., Carbide & Carbon Chemicals Corp., Box P, Oak Ridge, Tenn.
Carnes, Paul S. ('40; '43) (CEFHS), D-Z Hide Rd., Kidderpore, Calcutta, India.
Carney, Henry Raymond (A'46) (CWY), Mgr., W. & J. Sloane, c/o N.C. Shipbldg. Co.; *for mail,* 114 Mimosa Pl., Wilmington, N.C.
Carney, Joseph F. ('15) (ESBYZO), Supvg. Engr., Hotel Waldorf-Astoria Corp., 301-09 Park Ave., New York 22; *for mail,* 1749 Grand Concourse, New York 53, N.Y.
Carney, Kreigh, Jr. (J'39) (JMB), Research Engrg. Dept., Jones & Lamson Mch. Co., Springfield; *for mail,* 47 Atkinson St., Bellows Falls, Vt.
Carney, Wm. H. ('23; '33) (CDM), C.P.A., 420 Lexington Ave., New York; *for mail,* 4134 Case St., Elmhurst, L.I., N.Y.
Carpenter, Allan O. (A'09), Ch. Engr., Painted Post Plant, Ingersoll-Rand Co., Painted Post; *for mail,* 32 E. 3rd St., Corning, N.Y.
Carpenter, Benjamin Sears ('39) (CMS), Plant Engr., Willard Storage Battery Co. of Calif., Box 6850, East Los Angeles Sta., Los Angeles 22, Calif.
Carpenter, Donald F. (J'40) (SKM), Mech. Engr., Drydock & Repair Div., Drydock Dept., Tampa Shipbuilding Co., Tampa; *for mail,* 2920 Coachman Ave., Tampa 6, Fla.
Carpenter, Edward L. ('40), Asst. Prof. Mech. Engrg., R.I. State College, Bliss Hall, Kingston; *for mail,* 11 Sweet Fern Lane, Peace Dale, R.I.
Carpenter, George Dewey ('19) (CSHEKZ), Supt., Water & Sewer Depts., City of Ithaca, Tioga St.; *for mail,* E. State St., Ithaca, N.Y.
Carpenter, Howard Baylis ('25; '27) (WHRA), Engrg. Div., Sales Dept., Stand. Oil Co. of N.J., 26 Broadway, New York 4, N.Y.
Carpenter, James W. (J'41), 19930 Lake Shore Blvd., Cleveland 19, Ohio.
Carpenter, Kenneth C. (J'42) (ACM), 1133 Sheridan, Saginaw, Mich.
Carpenter, Miles Stahl (J'37) (SJCELM), Spotwelding Supvr., Buick, Oldsmobile & Pontiac Assembly Div., Gen. Motors Corp., Edgar Rd., Linden; *for mail,* 624 Harrison St., Rahway, N.J.
Carpenter, R. C. ('44) (CES), Prof. Emeritus, Mech. Engrg., Miss. State College, State College; Cons. Engr., Starkville, Miss.
Carr, Arthur A. (J'36) (CDL), Plant Engr., Quaker Maid Co., Inc., 45 Washington St., Brooklyn; *for mail,* 26 Montgomery Ave., St. George, S.I. 1, N.Y.
Carr, Guy Rufus (J'42) (RMFJSB), Shop Inspr., Norfolk & West. Ry.; *for mail;* 418 Otterview Ave., Roanoke 15, Va.
Carr, Henry Richard (J'36) (KHSDEC), Mech. Engr., Reecon Engrg. Co., 615 Jackson Bldg., Buffalo 2; *for mail,* 120 Highland Ave., Buffalo 9, N.Y.
Carr, Hugh H. ('31; '35) (BLS), c/o Distillation Products, Inc., 755 Ridge Rd. W., Rochester, N.Y.
Carr, Hugh R. ('35) (ST), Treas., Ch. Cons. Engr., Mech-Chem. Engrg., Inc., 1180 Raymond Blvd., Newark; *for mail,* 27 Elmora Ave., Cranford, N.J.
Carr, John H. (J'41) (BZNJ), Mech. Engr., Naval Ord. Test Sta., Inyokern; *for mail,* 1109 S. Garfield Ave., Alhambra, Calif.
Carr, Laurence Hamilton ('42) (JMS), Dir. Research, Edward Valve & Mfg. Co., Inc., 1200 W. 145th St., East Chicago, Ind.
Carr, Leonard (J'45), 319 Leroy Ave., Cedarhurst, N.Y.
Carr, Louis B. (J'39) (ZLY), Design Engr., Instrument Dept., Dow Magnesium Corp., Velasco; *for mail,* 1418 W. 4th, Freeport, Tex.
Carr, Robert Adams ('44) (CRS), Pres., Dearborn Chem. Co., 310 S. Michigan Ave., Chicago 4, Ill.
Carr, Robert Edgar (J'38) (NBLHZY), Mech. Engr., Designer, Bur. of Ord., Navy Dept., 1030 E. Green, Pasadena 1; *for mail,* 1111½ S. Garfield Ave., Alhambra, Calif.
Carr, Ronald H. (J'43) (MZC), Mech. Engr., Emulsol Corp., 59 E. Madison St., Chicago 3; *for mail,* 4750 Malden St., Chicago 40, Ill.
Carreau, Gerald (J'43) (NMZBJG), Mech. Engr., Columbia Broadcasting System, Inc., 485 Madison Ave., New York; *for mail,* 402 Foster Ave., Brooklyn 30, N.Y.
Carr-Harris, Gordon Grant ('43) (SRYB), 315 Somerset St. W., Ottawa, Ont., Can.
Carrick, Gerald S. ('19) (CKS), Box 456, Pleasantville, N.Y.
Carrier, George Francis (J'39) (BK), Brown Univ., Providence, R.I.
Carrier, Willis H. ('05; '12; H'42) (KHBSZA), *A.S.M.E. Medallist,* '34; Chmn. of Bd., Carrier Corp., 300 S. Geddes St., Syracuse 1, N.Y.
Carriere, John Gerand ('39) (C), R. T. Collier Corp., 439 Petroleum Bldg., 714 W. Olympic, Los Angeles 15, Calif.
Carriere, Murray F. (J'38), 299 Evelyn Ave., Toronto, Ont., Can.
Carrig, James Augustine (J'43) (OEFCMD), Constr. Engr., J. J. Powers Constr. Co., Kendal Sq., Cambridge; *for mail,* 215 Orchard St., Watertown, Mass.
Carrigg, Thomas Francis, Jr. (J'46), 41 N. Cary St., Brockton 17, Mass.
Carrington, Frank G. ('42) (CDJKMN), Gen. Mgr., Glamorgan Pipe & Fdy. Co., Lynchburg, Va.
Carroll, Charles David (J'45) (AGH), Apprentice Seaman, U.S.N.R.; *for mail,* 124 Heather St., Salt Lake City 2, Utah.
Carroll, Elbert Henry ('00) (J), Pres., Treas., Carroll Pressed Metal Co., 133 Dewey St., Worcester 2, Mass.
Carroll, Elmore John ('43), Designer & Engr., Francisco & Jacobus, 511—5th Ave., New York; *for mail,* 146-35—15th Ave., Whitestone, L.I., N.Y.
Carroll, Emil J. ('18) (CMS), 128 Linden Ave., Glencoe, Ill.

Carroll, Frank Theodore, Jr. (J'43) (ABMG), Box 8005, University Sta., Baton Rouge 3, La.
Carroll, Harry Clarence ('27) (SFKO), Head, Mech. Engrg. Dept., Commercial Testing & Engrg. Co., 307 N. Michigan Ave., Chicago 1, Ill.
Carroll, James David ('30; '41) (SMCBAH), Mech. Engr., Bur. of Budget of City of N.Y., Municipal Bldg., New York 7; *for mail,* 112-41—204th St., St. Albans 11, L.I., N.Y.
Carroll, John Bruce, Jr. (J'40) (YGC), Sales Engr., J. B. Carroll Co., 319 N. Albany St., Chicago 12; *for mail,* 502 Washington, Oak Park, Ill.
Carroll, Lafayette D. ('88), Mech. Engr., Humphreys & Glasgow, Ltd., Humglas House, Carlisle Pl. & St. Francis St., Victoria, London, S.W. 1, England.
Carroll, Phil, Jr. ('43) (DMJ), 6 Crestwood Dr., Maplewood, N.J.
Carroll, Robert P. (J'40), Pvt., U.S.A., 1417 N. 31st St., Omaha, Neb.
Carse, Donald Rede ('42) (RJA), Pullman-Stand. Car Mfg. Co., 52 Vanderbilt Ave., New York 17, N.Y.
Carson, Gordon Bloom ('33; '41) (CMDTYS), Mgr. Engrg., Selby Shoe Co.; *for mail,* 2839 Ridgeway Rd., Portsmouth, Ohio.
Carson, Harold Vernon ('44) (BJS), West Chester Rd., R.D. 1, Coatesville, Pa.
Carson, John Maurice (J'40) (CFJS), Asst. Mech. Engr., Arabian Am. Oil Co., 200 Bush St., San Francisco, Calif.; *for mail,* c/o Arabian Am. Oil Co., via Boatmail, A.P.O. 816, c/o Postmaster, New York, N.Y.
Carson, Knight S. (J'40) (AE), 1st Lt., Tank Automotive Center, Detroit; *for mail,* 552 Cadieux Rd., Grosse Point, Mich.
Carson, William Harry (J'40) (FMCEHR), Sales Engr., Goodman Mfg. Co., 831—2nd Ave., Huntington 16; *for mail,* Box 3, Bluefield, W. Va.
Carson, William Henry ('31) (CYP), Dean, College of Engrg., Univ. of Okla., Norman, Okla.
Carspecken, Henry L., Jr. ('34; '43) (LNCZJH), Mech. Engr., Carbide & Carbon Chemicals Corp., 427 McCorkle Ave., South Charleston 3, W.Va.
Carstarphen, Charles Frederick (J'42), 3767 Stearnlee Ave., Long Beach 8, Calif.
Carswell, John Stokes ('45) (DCMNH), Pres., Carswell Mar. Associates, Inc., 15 Park Row, New York 7, N.Y.
Carten, Leo A. (J'34) (BNMJ), Lt. Col., Chief, Mch. Gun Branch, Small Arms Div., Office of Ch. of Ord., War Dept., Washington, D.C.; *for mail,* 724 S. St. Asaph St., Alexandria, Va.
Carter, Charles Williams (J'46) (ABL), Geo. S. Armstrong & Co., Inc., 52 Wall St., New York, N.Y.
Carter, Donald C. (J'38) (ACMZ), Works Mgr., Chefford Master Mfg. Co., Inc., W. Main St., Fairfield, Ill.
Carter, Douglas S. (A'23), Sales Engr., Superheater Co., 60 E. 42nd St., New York; *for mail,* 6 Ridgecrest E., Scarsdale, N. Y.
Carter, Emmett B. ('12) (COMDS), Am. Iron & Steel Inst., 350—5th Ave., New York 1, N.Y.; *for mail,* 19 Prospect Terrace, Tenafly, N.J.
Carter, Frederic W. ('20), 65 Bonair Ave., Waterbury 47, Conn.
Carter, George William (J'33) (FKLSEO), Asst. Prof. Mech. Engrg., Univ. of Utah, Salt Lake City 1, Utah.
Carter, Henry W. ('92; '03), Counsel, Legal & Pat. Dept., Owens-Ill. Glass Co., Ohio Bldg., Toledo, Ohio.
Carter, James Hinman ('40) (CEFRS), Commr., Smoke Regulation, City of St. Louis; *for mail,* 6054 Pershing Ave., St. Louis 12, Mo.
Carter, Joseph Coleman ('45) (KBFAHJ), Mech. Engr., De Laval Steam Turbine Co., Trenton 2, N.J.; *for mail,* 330 E. 43rd St., New York 17, N.Y.
Carter, Keith L. (J'40) (SZCFL), Design Engr., Stand. Oil Co. (Ind.), Whiting, Ind.
Carter, Louis Ewen ('41; '43) (CHS), Asst. Tech. Engr., Consltd. Gas, Elec. Light & Power Co. of Baltimore, 2016 Lexington Bldg., Baltimore 1; *for mail,* 4814 Laurel Ave., Baltimore 15, Md.
Carter, Preston Day ('45) (CDO), Maj., Ord. Dept., U.S.A., A.P.O. 757, c/o Postmaster, New York, N.Y.; *for mail,* 8741 Arcadia Ave., Detroit 4, Mich.
Carter, Robert Jefferson ('40) (L), Petroleum Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 110 Sewall Ave., Brookline, Mass.
Carter, Terry L., Jr. ('44) (DLM), Matls. Handling Engr., Am. Cyanamid & Chem. Corp., 30 Rockefeller Plaza, New York 20, N.Y.
Carter, Wayne Warren (J'42) (ZNFEBS), Assoc. Engr., Research Dept., Elliott Co.; *for mail,* 125 N. 2nd St., Jeannette, Pa.
Carter, Wilber Albert ('20; '35) (SFHKZO), Tech. Engr., Power Plants, Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Carter, William Richardson ('44) (SCEJKM), Asst. Supt., Engrg. Dept., Swett & Crawford, Underwriting Mgrs., Pacific Indemnity Co., 621 S. Hope St., Los Angeles 14; *for mail,* 1265 Leighton Ave., Los Angeles 37, Calif.
Cartinhour, John Wilkerson (J'44), Engr., Foster Wheeler Corp., 165 Broadway, New York, N.Y.
Cartwright, Kenneth ('43) (ERS), 55 Brookside Dr., Hamden, Conn.
Carty, Maurice Walter ('21) (DEFKLS), Sr. Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 77 Fairway Rd., Brookline 67, Mass.
Caruthers, Elmo, Jr. (J'30) (ECM), Comdr., S(E), U.S.N.R.; 21 Aberdeen Rd., Elizabeth 3, N.J.
Carver, Fred S. ('16; '21), Propr., Firm of Fred S. Carver, 345 Hudson St., New York 14, N.Y.
Carver, Vinton DeVere (J'44) (NM), Supt., Lab. Services, Tenn. Eastman Corp., Y-12 Area; *for mail,* 364 Louisiana Ave., Oak Ridge, Tenn.
Carvey, Thomas Banks, Jr. (J'43), Lt., U.S.N.R.; 843 Decotin St., N.W., Washington 11, D.C.
Carvin, Frank Dana ('30) (SAEKFO), Chmn., Mech. Engrg. Dept., Newark College of Engrg., 367 High St., Newark 2, N.J.
Casamento, Joseph Robert (J'43) (NBHZAC), Pvt., 1st Class, U.S.M.C.R., Arch. & Mech. Draftsman, Hdq. & Serv. Co., 2nd Engr. Bn., 2nd Div.; *for mail,* 231—72nd St., North Bergen, N.J.
Casberg, Carl Herbert ('21; '28) (CDJLN), Prof. Mech. Engrg., Univ. of Ill., Urbana, Ill.
Case, George S. ('17) (BCJ), Chmn. of Bd., Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, Ohio.
Case, Lynn B. (J'18) (CGWTY), R.F.D. 1, Plank Rd., Fredericksburg, Va.
Case, Melville Cornell (J'34) (LYCS), Plant Engr., Rohm & Haas Co., 730 Dale Ave.; *for mail,* 2339 Ocoee Trail, Knoxville, Tenn.
Case, Robert Charles ('24; '35) (RWJMYG), "Courtlands," 3 Court Rd., Tunbridge Wells, Kent, England.
Case, Willard Lacy ('38), Assoc., Geo. S. Armstrong Co., Inc., 52 Wall St., New York 5, N.Y.
Case, William Erwin (J'37) (MCA), Supvr. Inspc., Scovill Mfg. Co., Waterville; *for mail,* 44 Grace Ave., Waterbury 55, Conn.
Caserza, Louis Manuel (J'41) (HNFMA), Asst. Naval Arch., Naval Shipyard, San Francisco 24; *for mail,* Box 87, Route 1, Colma 25, Calif.
Casey, Albert A. ('45) (SFCDEO), Gen. Supt. Plants, Cleveland Elec. Illum. Co., 75 Pub. Sq., Cleveland 1, Ohio.
Casey, Eugene ('44), The White House, Washington, D.C.
Casey, George Rhoads ('44), Pres., Treadwell Engrg. Co., Easton, Pa.
Casey, John Edward ('24; '35) (SFCKZB), Ch. Engr., Jersey Cent. Power & Light Co., South Amboy, N.J.
Casey, John Schuyler ('27; F'43) (CO), Pres., M. H. Treadwell Co., Inc., 140 Cedar St., New York 6, N.Y.
Casey, Robert Joseph (J'44), 5525 Whitby Ave., Philadelphia 43, Pa.
Cash, Stuart Albert (J'45) (EMY), 821 Bergen Ave., Jersey City, N.J.
Caskey, Kenneth Harry (J'25) (BJFLM), Ch. Engr., Harrisburg Steel Corp.; *for mail,* 2257 Rudy Rd., Harrisburg, Pa.
Casler, William A. (J'40) (NECZB), Engr., Research Dept., Caterpillar Tractor Co., East Peoria; *for mail,* 7242 Euclid Ave., Chicago 49, Ill.
Caspell, Edwin Ernest (J'32) (CJDBM), Works Supt., Am. Steel & Wire Co., 238 Fairmount Ave., New Haven 7; *for mail,* 26 Pennsylvania Ave., East Haven 12, Conn.
Casper, Harlan W. (J'41) (EKS), Part Owner, Casper Implement & Hardware Co., Clifton, Kan.
Cass, Laurence John (J'41) (EFHKMS), Capt., Army Air Forces, Engrg. Officer, Ground Crew; *for mail,* 115—77th St., North Bergen, N.J.
Cassedy, William Fraser, Jr. ('39) (CMZN), V.P., Charge Prod., Foote, Pierson & Co., Inc., 75 Hudson St., Newark 4, N.J.
Cassel, Harrison Henry (J'41) (NB), Engr., Roessler Mch. Co., 5201 Magnolia St., Philadelphia 44; *for mail,* 4505 N. 18th St., Philadelphia 40, Pa.
Cassell, Charles William ('29; '41) (CDGLOS), Plant Engr., Bryant Paper Co., 2030 Portage St., Kalamazoo 29F; *for mail,* 1240 Royce Ave., Kalamazoo 28, Mich.
Cassidy, Francis Robert (J'43) (AEFBK), Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* 53-47—212th St., Bayside, L.I., N.Y.
Cassidy, Herbert Wesley Dwight ('38; '45) (ACEMN), Designer, Glenn L. Martin Co., Middle River, Baltimore; *for mail,* 40-A Westway N., Baltimore 21, Md.
Cassidy, Perry R. ('13; '26) (BJS), Exec. Asst., Babcock & Wilcox Co., Barberton; *for mail,* Mayflower Hotel, Akron, Ohio.
Cassidy, Thomas Frank, Jr. ('29; '45) (BCM), Plant Mgr., Gray Mfg. Co., 16-30 Arbor St., Hartford 1; *for mail,* 31 Wells Farm Dr., Wethersfield, Conn.
Cassin, William (J'40), Jr. Draftsman, Humble Oil & Refining Co., Ingleside; *for mail,* Box 27, South San Antonio, Tex.
Cassola, Charles Alfred (J'43) (AWCJD), Aircraft Inspr., Naval Air Matl. Center; *for mail,* 500 S. 46th St., Philadelphia, Pa.
Casson, Ernest Howard ('44) (EFKOS), Cons. Engr., 53 W. Jackson Blvd., Chicago 4, Ill.
Casson, Kenneth Hodgson ('41) (CMNB), Supt., Barnes Drill Co., 814 Chestnut St.; *for mail,* 1317 Ridge Ave., Rockford, Ill.
Cassotti, Mario ('32; '44) (SKEBFH), Ventilation Design Engr., Gibbs & Cox, Inc., 21 West St., New York 6; *for mail,* 318 W. 107th St., New York 25, N.Y.
Castagne, Americo Joseph (J'43), 2nd Lt., Army Air Forces, Bachelor Officers' Quarters, 3084th Base Unit, Gardner Field, Taft, Calif.
Castellano, Charles (J'41) (ABECSY), Assoc. Mech. Engr., Air Tech. Serv. Comd., Wright Field, Dayton; *for mail,* 305 Oakwood Ave., Dayton 9, Ohio.
Castellini, Darius Luigi (J'39) (BAECNY), Mech. Engr., Watson Labs., Air Tech. Serv. Comd., Red Bank; *for mail,* 57 Belshaw Ave., Eatontown, N.J.
Castiglione, Charles B. (J'43) (SJKHLM), Roots-Connersville Blower Corp., Connersville, Ind.
Castillo, Charles A. (J'37) (CD), Ch. Engr., Rochester Ropes, Inc.; *for mail,* 402 Macoy Ave., Culpeper, Va.
Castle, Drew W. ('23; '32) (ACM), Vocational Dir., Joliet Township High Sch. & Jr. College, Joliet, Ill.
Castle, Kendall B. (J'25), Capt., Ord. Dept., Army Serv. Forces, Rochester Ord. Dist., Rochester; *for mail,* 67 Council Rock Ave., Rochester 10, N.Y.
Castle, S. N. ('09; '11) (CSHLEB), 2425 University Ave., Honolulu 5, T.H.
Castleman, John Hamilton (J'43) (HKLZ), Mech. Engr., Design, Wood River Refinery, Shell Oil Co., Inc., Wood River, Ill.; *for mail,* 5923 Lotus Ave., St. Louis 12, Mo.
Castro-Amy, Rafael (J'45) (BMO), Box 2926, San Juan, P.R.
Caswell, Thomas B. ('42) (HC), Contrg. Engr., McCarthy Well Co., 670 Eustis St., St. Paul 4; *for mail,* Maplewood, Wayzata, Minn.
Caswell, Vincent Earl (J'41) (AMDBJN), Stress Analyst, Aircraft Div., Universal Moulded Products Corp., Commonwealth Ave.; *for mail,* Gen. Delivery, Bristol, Va.
Cate, Eugene Russel (J'37) (ABESYN), Lt. Comdr., U.S.N.R., Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail,* 510 Anne St., Falls Church, Va.
Cather, Harold Malcolm ('37) (SKEBNC), Head, Mech. Engrg. Dept., College of Engrg., W.Va. Univ., Morgantown, W.Va.
Cather, Jay Howard ('19; '24) (EFKS), Supt., Power Dept., Kodak Park Works, Eastman Kodak Co., Rochester; *for mail,* 285 San Gabriel Dr., Rochester 10, N.Y.
Catlin, Harold Milo ('44) (CDSMOL), Mech. Engr., Day & Zimmerman, Inc., Packard Bldg., Philadelphia 2; *for mail,* 44 Hilaire Rd., St. Davids, Pa.
Catlin, John (J'40) (BJN), Research Engr., Remington Arms Co., 939 Barnum Ave., Bridgeport 2, Conn.
Catlin, Welles G. ('40) (CDJ), 32 Grant St., Utica 3, N.Y.
Cattaneo, Alfred Giovanni ('44) (EACFSK), Research Engr., Dept. Head, Shell Devel. Co., 4560 Horton St., Emeryville, Calif.
Cattermole, Lester Gehman ('17; '19; '30) (COMJWL), 1294 Bay St., Springfield 9, Mass.
Cauble, Gordon B. (J'40), Lt., Sig. Corps, U.S.A., Dir., Trng. Div., East. Sig. Corps Trng. Center, Russel Hall; *for mail,* 10 Russel Ave., Ft. Monmouth, N.J.
Caughey, Clarence Harold ('45) (MCJKNS), Ch. Engr., Min. Div., Cardox Corp., 307 N. Michigan Ave. Chicago 1, Ill.
Cauvel, Howard Lyman (J'46), Box 784, Route 1, Bellevue, Wash.
Cavanaugh, James Pierce ('22; '35), Pres., Treas., J. E. Lonergan Co., 2nd & Race Sts., Philadelphia 6, Pa.
Cave, John Richard (J'36), Mech. Engr., N.Y. Staff, U.S. Rubber Co., 1230—6th Ave., New York 20; *for mail,* 212-16—38th Ave., Bayside, N.Y.
Cavin, Gustave Charles ('16) (RAB), 154 Stuart St., Kingston, Ont., Can.
Cawley, George ('21; '35) (JYMR), Designer, Flintkote Co., Oak St., East Rutherford; *for mail,* 382 Highland Ave., Upper Montclair, N.J.
Caywood, Charles Melick ('45) (DKS), Plant Engr., Gerber Products Co.; *for mail,* Kimbark Inn, Fremont, Mich.
Cecil, Robert E. ('21) (CJ), V.P., Sales, Scaife Co., Oakmont; *for mail,* 1402 N. Negley Ave., Pittsburgh 6, Pa.
Cederborg, Gibson Ames (J'44) (AHBNEM), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail,* Apt. 4, 260 Regent St., Hampton, Va.
Ceely, Frederick Jesse (J'43) (KBSHCL), Design Engr., Steam Div., Foster Wheeler Corp., 165 Broadway, New York 6; *for mail,* 146-30 Huxley St., Rosedale, N.Y.
Cehrs, Charles H. (J'40) (AEM), Engrg. Draftsman, Boeing Aircraft Co., Georgetown Sta., Seattle; *for mail,* 1225 E. Northlake, Seattle 5, Wash.
Cejka, Joseph B. (J'41) (MOAJBC), Asst. Prof. Mech. Engrg., Rutgers Univ., New Brunswick, N.J.
Cella, John Thomas (J'44), 37 Washington Sq. W., New York, N.Y.
Cernea, Roland Emanuel ('41) (BCDJMN), Plant Engr., Pa. Forge Corp., Milnor & Bleigh Sts., Philadelphia 35; *for mail,* Apt. 10, 5051 Oxford Ave., Philadelphia 24, Pa.
Cerny, Walter J. ('30; '37) (ABCMNO), Dir. Engrg., Northrop Aircraft, Inc., Northrop Field, Hawthorne; *for mail,* 5960 Tuxedo Terrace, Hollywood 28, Calif.
Cerreto, Sebastian Salvatore (J'44), 22—10th St., New London, Conn.

Cesarz, William John, Jr. (J'45) (FJS), U.S. Navy; 757—1st St., Westfield, N.J.
Chadwick, Jean Stuart (J'39) (FK), Engrg. Cartographer, Aircooled Motors Corp., Liverpool Rd., Syracuse; *for mail,* 320 Wellesley Rd., Syracuse 7, N.Y.
Chadwick, Lee Sherman ('99; '09) (BCEFHJ), Pres., Chmn. of Bd., Perfection Stove Co., 7609 Platt Ave., Cleveland 4, Ohio.
Chadwick, Norman Howard (J'43) (CHND), John Robertson Co., Inc., 133 Water St., Brooklyn 1, N.Y.; *for mail,* 1231 Cambridge Ave., Plainfield, N.J.
Chafee, John S. ('26; '35) (CMNT), V.P., Saco-Lowell Shops, 60 Batterymarch St., Boston 10, Mass.; *for mail,* 245 President Ave., Providence 6, R.I.
Chaffe, William H. ('22) (CDMN), Works Mgr., Simmons Co., Brunswick Ave., Elizabeth F; *for mail,* Box 122, Meridith Rd., Colonia, N.J.
Chaffee, Rupert A. (J'38) (BEP), Serv. Engr., Clark Bros. Co., Inc., Olean, N.Y.; *for mail,* 125 W. 1st St., Tulsa, Okla.
Chaffey, Edmund Kilbourne, Jr. (J'42) (NOESG), Mech. Engr., Pur. Dept., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 3027 Mt. Allister Rd., Pittsburgh 14, Pa.
Chaffin, Warren Leland ('29; '35) (CMSTDA), Exec. V.P., J. L. Stifel & Sons, Inc., Main & 4th Sts., Wheeling, W. Va.
Chaharyn, Orist D. (J'45), Capt., Co. C, 1st Trng. Bn., Camp Ritchie, Md.
Chalfant, Arthur I. (J'41) (BHKE), Research Engr., Baker Mfg. Co.; *for mail,* 12 N. 2nd St., Evansville, Wis.
Chalkley, Curtis Rathbone ('16; '24) (BJKMNR), Mech. Engr., Navy Dept., 35 E. 7th St., Cincinnati 2, Ohio; *for mail,* 40 Beechwood Rd., Ft. Mitchell, Ky.
Chalkley, Henry G. ('08), Sweet Lake Land Oil Co., Lake Charles, La.
Challies, John Bow ('43) (SCH), V.P., Exec. Engr., Shawinigan Water & Power Co., 107 Craig St. W., Montreal 1, Que., Can.
Challman, Arnold Telfer (J'44) (ZCLBEJ), Sales Engr., Minneapolis-Honeywell Regulator Co., 1136 Howard St., San Francisco 3, Calif.
Chambellan, Rene Edmond (J'46) (BCL), 537 St. Pauls Ave., Cliffside Park, N.J.
Chamberlain, Gordon Lyle (J'31) (RJMNEB), Engr., N.Y. Air Brake Co., Starbuck Ave.; *for mail,* 315 California Ave., Watertown, N.Y.
Chamberlain, Leon Hoyt ('25; '35) (HZYLJA), East. Mgr., Am. Well Works, 475—5th Ave., New York 17; *for mail,* 766 Palmer Rd., Bronxville 8, N.Y.
Chamberlin, John Maurer (J'43) (YLJMAB), Tool Engr., Plastics Div., Monsanto Chem. Co., Monsanto Ave.; *for mail,* 61 Oak St., Indian Orchard, Mass.
Chamberlin, John Windsor ('43) (TKEJO), Pres., Appliance Corp. of Am., 430 E. Silver Spring Rd., Milwaukee 11, Wis.
Chamberlin, William T. ('17) (EFKS), Mech. Engr., Dept. of Pub. Works, City of New York, 125 Worth St., New York 7, N.Y.
Chambers, Edwin Graydon ('32; '42) (CMDLJ), Engr., Mfg. Processes, West. Elec. Co., Inc., 2600 Broening Highway, Baltimore; *for mail,* 7312 Yorktown Dr., Towson 4, Md.
Chambers, Henry Edward, Jr. ('31) (ESC), Lt. Col., Corps of Engrs., U.S.A., c/o A.M. Lockett & Co., 401 Magnolia Bldg., Dallas, Tex.
Chambers, James Wesley (J'43) (JRDELO), Inspr., La. Div., Stand. Oil Co. of N.J., Box 551, Baton Rouge 1; *for mail,* 3739 Perkins Rd., Baton Rouge 15, La.
Chambers, Norman Campbell ('05; '14) (EJMN), Retired; 40 E. 54th St., New York 22, N.Y.
Chambers, William Henry (J'38) (NCJMBY), Mech. Design Draftsman, Submarine Sig. Co., 160 State St., Boston; *for mail,* 154 Cedar St., East Dedham, Mass.
Chambers, William R. ('21; F'45) (CDFHOS), Cons. Engr., Clinton Engrg. Works, Tenn. Eastman Corp.; *for mail,* R.D. 6, Holston Hills, Knoxville, Tenn.
Champion, Albert R. (J'40) (NKEAZS), Scripps Inst. of Oceanography, Univ. of Calif., La Jolla, Calif.
Champion, C. H. ('33), Managing Dir., Charles H. Champion & Co. Ltd., 99 Howard's Lane, Putney, London, S.W. 15, England.
Champion, Edward Lees (J'44) (SHK), V.P., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.
Champion, Noel Martin (J'43), Aviation Radio Technician, 3rd Class, Carrier Aircraft Serv. Unit, F9, F.P.O., San Francisco, Calif.; *for mail,* 1302—1st Ave. E., Spencer, Iowa.
Champlin, Donald Webb ('42) (MJC), V.P., Gen. Mgr., Defiance Mch. Works, Inc., Defiance, Ohio.
Champney, Ralph Preston ('16; '35) (CDLMY), Pres., Mitchell & Smith, Inc., 9501 Copland Ave., Detroit 17, Mich.
Chan, Albert So (J'43) (KSMNHA), Design Draftsman, Mar. & Struc. Engr., Bethlehem Steel Corp., 20th & Illinois St., San Francisco; *for mail,* 241½ N. Humboldt Rd., San Mateo, Calif.
Chan, Wellington K. (J'41) (BCMAZG), Lab. Engr., Ward Leonard Elec. Co., 31 South St., Mount Vernon; *for mail,* 37-38—65th St., Woodside, N.Y.
Chandler, Arthur Hall (J'43) (MCA), 40 Howard St., Reading, Mass.
Chandler, Edward Farrington ('45) (AOS), Engr. (Consultant), 148 Richmond St., Brooklyn 8, N.Y.
Chandler, Harry Stuart, Jr. (J'40) (HLM), Engr., Defence Indus., Ltd., Box 10, Montreal; *for mail,* 65B Cedar Ave., Shawinigan Falls, Que., Can.
Chandler, Howard McCune ('13) (ABR), 109-28—212th St., Bellaire, Queens Village 9, N.Y.
Chandler, Lawrence Fowler (J'43) (MCJ), 2010½ Chamberlain Ave., Chattanooga 4, Tenn.
Chandler, Marland Hewes (J'40) (SOFCL), Pacific Gas & Elec. Co., 245 Market St., San Francisco, Calif.
Chandler, Orville Woodrow (J'42) (NABCJM), Mech. Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 2112 Hugh St., Schenectady, N.Y.
Chandler, Robert (J'34) (CDM), Apt. 3 N, 175 Linwood, Buffalo 9, N.Y.
Chandler, Thompson ('46) (CJK), Chem. Engr., Carbide & Carbon Chemicals Corp., Box N, S. Charleston 3, W. Va.
Chandler, William Charles (J'41) (CES), Cadet Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark, N.J.; *for mail,* 2193 E. 21st St., Brooklyn 29, N.Y.
Chaney, Alex Fred (J'41) (HLMBJA), Design Engr., Simmons Mfg. Co.; *for mail,* 241 Pearl St., Ashland, Ohio.
Chaney, George Hall, Jr. (J'45) (ACW), c/o Mrs. C. J. Greene, 759 Middlefield Rd., Palo Alto, Calif.
Chankalian, Robert H. (J'39) (CMG), Electronic Technician's Mate, 3rd Class, U.S.N., Naval Research Lab., Washington 20, D.C.; *for mail,* 505 Post Ave., Lyndhurst, N.J.
Channappa, B. K. ('45), Mysore Iron & Steel Works, Bhadravati, India.
Chanoux, Theodore (J'43) (COMA), 826 E. 234th St., Bronx, N. Y.
Chao, Eugene (J'45) (SEFKOB), Asst. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 38 Divinity Hall, Harvard Univ., Cambridge 38, Mass.
Chao, Pei Hua (J'43) (FKS), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Chao, Wu Wai (J'45) (BJEMNT), 290 Massachusetts Ave., Cambridge 39, Mass.
Chapin, B. R. (J'42) (ASC), Field Serv. Rep., United Aircraft Serv. Corp., East Hartford, Conn.; *for mail,* 243 S. Crestway, Wichita, Kan.
Chapin, George W. (J'36) (CPS), Lt. Comdr.; 213 North C, McAlester, Okla.
Chapin, Warren Winthrop ('17) (WJ), Cons. Engr., 2 W. 45th St., New York 19, N.Y.; *for mail,* 354 Highland Ave., South Norwalk, Conn.
Chaplin, Frank Sprague ('42) (BRJA), Research Engr., The Franklin Inst., 20th St. & Parkway, Philadelphia 3; *for mail,* 822 Glendalough Rd., Philadelphia 18, Pa.
Chaplin, John H. ('20; '26; '27), Veeder-Root, Inc., Hartford, Conn.
Chapman, Edmund Earle ('44), Mech. Asst., Atchison, Topeka & Santa Fe Ry. Co., 1033 Ry. Exchange Bldg., 80 E. Jackson Blvd., Chicago; *for mail,* 229 N. Humphrey, Oak Park, Ill.
Chapman, Edward Corbin ('45) (JLNSFB), Ch. Metallurgist, Combustion Engrg. Co., Inc., 1032 W. Main St., Chattanooga 1, Tenn.
Chapman, Everett ('43) (JARBKY), Consultant, Box 387, Coatesville; *for mail,* R.F.D. 4, West Chester, Pa.
Chapman, John Barton (J'45) (BCJMZ), Process Engr., Remington Arms Co., Inc., 939 Barnum Ave., Bridgeport; *for mail,* 7 Lalley Blvd., Fairfield, Conn.
Chapman, Paul Allen ('43), Acting Gen. Plants Mgr., Am. Bemberg Corp., N. A. Rayon Corp.; *for mail,* 235 F St., Elizabethton, Tenn.
Chapman, Reginald Ernest (J'44) (CD), Factory Mgr., Internatl. Business Mchs. Co., Ltd., 300 Campbell Ave. W., Toronto 9, Ont., Can.
Chapman, Robert Gorham ('36; '44) (NMC), Chmn., Dept. Mech. Engrg., Univ. of Vt., Burlington, Vt.
Chapman, Robert Hett ('19; '24; '27) (TCS), V.P., Asst. Treas., Riverdale Mills & Inman Mills, Box 1714, Spartanburg, S.C.
Chapman, Semmes, Jr. (J'42) Lt. (j.g.), U.S.N.R., 4117 Gosnold Ave., Norfolk, Va.
Chapman, W. Lewis ('41) (LS), Exec. Engr., Mich. Sugar Co., 2nd Natl. Bank Bldg.; *for mail,* 116 Garden Lane, Saginaw, Mich.
Chapman, William Cullen (J'46), Lt., Army Air Forces, 1504th Base Unit, Fairfield, Calif.
Chappell, Henry David ('43) (SBKCFE), Mech.-Elec. Engr., Power & Maint. Div., Burroughs Adding Mch. Co., 6071—2nd Ave., Detroit 32; *for mail,* 15493 Whitcomb Ave., Detroit 27, Mich.
Chappell, William G. (J'38) (C), 3334 Donald Ave., Cincinnati, Ohio.
Chaput, Arthur J. ('30; '35) (BHJ), Prod. Mgr., Schmitt Steel Co., 2407 N.W. 28th St.; *for mail,* 1955 N.W. Hoyt St., Portland, Ore.
Charignon, Michael J. (J'38) (BFJ), Asst. Supt., Steam Generation, Republic Steel Corp., Market St.; *for mail,* 856 Woodford, Youngstown, Ohio.
Charlamb, Alexander (J'43) (BAYK), Asst. Engr., Replacement Group, Natl. Aniline Div., Allied Chem. & Dye Corp., 1051 S. Park Ave., Buffalo; *for mail,* c/o Mrs. Wright, 73 Pawnee Pkwy., Buffalo 10, N.Y.
Charlesworth, Richard Lawrence (J'45) (CMBJNF), Student Engr., Internatl. Plainfield Motor Co., Front St., Plainfield; *for mail,* 358 Hartford Rd., South Orange, N.J.
Charlesworth, Robert Everett (J'43) (GOH), Design Engr., Norden Labs. Corp., 35 Varick St., New York, N.Y.; *for mail,* 358 Hartford Rd., South Orange, N.J.
Charlesworth, Roger Barnes (J'40) (CO), Supvr., Wage Incentives, West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 358 Hartford Rd., South Orange, N.J.
Charlewood, Charles Benjamin ('45) (SKC), Mech. Engr., Foster Wheeler Ltd., Aldwych House, London, W.C.2; *for mail,* c/o British Columbia House, 1-3 Regent St., London, S.W.1, England.
Charlton, Edward James ('44) (HJKR), Mgr. Devel. Engrg., Lukens Steel Co., Coatesville, Pa.
Chase, Arthur Leonard (J'45) (SRNTEF), 19 Kilton St., Taunton, Mass.
Chase, Charles Harris ('02) (NSEFTO), Prof. Emeritus, Engrg. Sch., Tufts College, Medford 57; *for mail,* 39 Lincoln St., Stoneham 80, Mass.
Chase, Julian Dwight ('36) (COMLSD), Indus. Engr., Spec. Asst., Office of Scien. Research & Devel., 1530 P St., N.W., Washington 25; *for mail,* Wardman Park Hotel, Washington 8, D.C.
Chase, Loren Alvin ('37; '45) (ZHB), Methods Engr., Bristol Co., Waterbury 91; *for mail,* 36 Superior Ave., Waterbury 64, Conn.
Chase, Philip Hartley ('29) (S), Ch. Engr., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Chasman, Bernard (J'40) (NBAYZC), Matls. Engr., Engrg. Div., Air Tech. Serv. Comd., Army Air Forces, Wright Field, Dayton; *for mail,* 18 Parran Dr., Dayton 10, Ohio.
Chason, Daniel Henry ('21; '29) (CJLMN), Devel. Engr., Singer Mfg. Co., Elizabethport; *for mail,* 843 Park Ave., Elizabeth 3, N.J.
Chasteen, Jack Sanford (J'44), Lt., Corps of Engrs., Det. E-201, 3rd Mil. Govt. Regt., A.P.O. 403, c/o Postmaster, New York, N.Y.
Chatard, William M. ('03; '09) (EFKS), 5506 Normandy Pl., Baltimore 10, Md.
Chater, John A. ('27; '32) (BDL), Ch. Engr., Niagara Sprayer & Chem. Co., Inc., Middleport; *for mail,* 504 Park Ave., Medina, N.Y.
Chatfield, Howard ('27; '35) (AC), Supvg. Engr., Minneapolis Territory, Fidelity & Casualty Co. of N.Y., 918 McKnight Bldg., Minneapolis 1, Minn.
Chatterjee, Basudeb ('40) (ACJNZ), Capt., Deputy Dir. of Trng., Dept. of Labor, Govt. of India; *for mail,* Benares Hindu Univ., Patna, India.
Chatterton, Herbert Irving (J'41) (EBC), Ch. Planner, Charge Prod. Dept., Todd Pacific Shipyards, Inc., Ft. Alexander Ave.; *for mail,* Box 1497, Tacoma 1, Wash.
Chattey, John Karl ('35; '42) (BCO), Capt., Sanitary Corps., A.U.S., A.P.O. 839, c/o Postmaster, New Orleans, La.
Chattler, Leo Morse (J'42) (AHBN), Lt., U.S.N.R., Asst. Head, Hyds. Sec., Bur. of Aero., Navy Dept., 18th & Constitution, Washington 25, D.C.
Chave, Charles Trudeau ('29; '38) (KLSCEZ), Process Engr., Stone & Webster Engrg. Corp., 49 Federal St.; *for mail,* 37 Beacon St., Boston 8, Mass.
Chavez, Charles Edward (J'42), Ensign, 1624 N. 52nd St., Milwaukee, Wis.
Chawner, Rupert William ('25) (HSCEKO), Gen. Mgr., Temescal Water Co., 707 Main, Corona, Calif.
Cheasley, Thomas C. ('39) (FSREK), Asst. to Pres., Sinclair Coal Co., 114 W. 11th St., Kansas City 6, Mo.
Cheek, Edward Earl (J'42) (BEJ), Ensign, U.S.N.R., U.S.S. *Cable,* ARS-19, F.P.O., San Francisco, Calif.
Cheeseman, Herbert Larcombe (J'41), Lt., Signal Corps Officers Replacement Pool, Camp Murphy, Fla.
Chelgren, William Judd (J'38) (CMDLZ), Prod. Supvr., Plastics Dept., E. I. du Pont de Nemours Co., Arlington; *for mail,* 25 Park St., Montclair, N.J.
Chen, Chia-Tseng (J'45) (A), Apt. 12F, 400 W. 119th St., New York, N.Y.
Chen, Chie-Yuan (J'43) (JKS), 12 Steele's Rd., London, N.W.3, England.
Cheney, Frank, Jr. ('15), Retired; 20 Hartford Rd., Manchester, Conn.
Cheney, Lloyd Theodore (J'44) (BAJ), Asst. Prof. Civ. Engrg., Cornell Univ., Ithaca, N.Y.
Chenoweth, Dale Murray (J'37) (YNL), Plant Engr., S. Branch Factory, Armstrong Cork Co., South Braintree; *for mail,* 41 Lantern Lane, East Braintree, Mass.
Chenoweth, George Milton, Jr. ('39) (CMNJD), Factory Supt., Am. Sawmill Mchy. Co., Main St., Hackettstown; *for mail,* 5 Crane Rd., Mountain Lakes, N.J.
Cherdantzeff, Peter ('22; '25; '35) (SHFBEK), Asst. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 3647 Broadway, New York 31, N.Y.
Cherniak, George Sadwin (J'45) (BEJNZO), Engr., J. M. Lessells & Associates, 108 Massachusetts Ave., Boston 15, Mass.
Cherry, Lawrence Hildreth ('37; '46) (ANMH), Sr. Mech. Engr., Manhattan Project, Los Alamos Lab., Box 1663, Santa Fe, New Mex.

Chesler, Isidor ('21; '35) (BWYZML), Dir., Research & Devel., Eagle Pencil Co., 710 E. 14th St., New York 9, N.Y.
Chesney, Malcolm M. ('20; '25), Secy., Suggestion Com., Gen. Elec. Co., 100 Woodlawn Ave.; *for mail,* 7 Kenilworth St., Pittsfield, Mass.
Chess, Gerald E. (J'35) (AKL), Indus. Sales Engr., Johnson Bros., 16 Yosemite Ave., Fresno 1, Calif.
Chester, Harold Dean (J'32) (BEFNSJ), Design Engr., Stand. Oil Co. of Calif., Richmond; *for mail,* 1249 Garfield Ave., Albany 6, Calif.
Chester, John N. ('05), Sr. Partner, Chester Engrs., 210 E. Park Way, N.S., Pittsburgh, Pa.
Chester, Ray Guthrie ('27; '35) (EFS), Supt. Diesel Generation, So. Calif. Edison Co., Ltd., Box 351, 5th & Grand Sts., Los Angeles; *for mail,* 10248 Kauffman Ave., South Gate, Calif.
Chester, Thomas ('15) (OLBHK), 700 Seward Ave., Detroit 2, Mich.
Chesters, Frank Chyczewski (J'37) (CMOAJD), Indus. Engrg. Dept., Thompson Aircraft Products Co., 23555 Euclid Ave., Euclid 17, Ohio.
Chestnut, Robert Glenn (J'46), 119—4th St. W., Cornwall, Ont., Can.
Chevrolet, Arthur J. (J'37) (FMP), Mech. Engrg. Draftsman, Silent Hoist Co., 841—63rd St., Brooklyn; *for mail,* 2020 Camp St., Bronx, N.Y.
Chew, Bernard Briscoll (J'41) (NBAE), Ch. Analyst, Lycoming Div., Aviation Corp.; *for mail,* 1804 Memorial Ave., Williamsport 38, Pa.
Chiavetta, Virgil V. (J'39) (BMNOD), Lt. (j.g.), U.S.N.R., Matl. Redistribution & Disposal Office, Navy Dept., 219 N. Broad St., Philadelphia 7, Pa.; *for mail,* 1256 E. 13th St., Brooklyn 30, N.Y.
Chick, Alton Charles ('21; '34) (BHLY), *Manager,* '43-'45, *Vice-President,* '45-'46; Asst. V.P., Engr., Mfrs. Mutual Fire Ins. Co., 815 Grosvenor Bldg., Providence 1, R.I.
Chicoine, Harvey D. (J'39) (CMJ), Prod. Engr., Charge Mech. Engrg., Strong Mfg. Co., Sebring; *for mail,* 1492 W. State St., Alliance, Ohio.
Chiger, Arthur (J'39) (NOESYL), Mech. Devel. Engr., Hazeltine Electronics Corp., 58-25 Little Neck Pkwy., Little Neck, L.I.; *for mail,* 1082 Morris Park Ave., New York 61, N.Y.
Child, Ralph Dexter, Jr. (J'42), 906 Trestle Glen Rd., Oakland 10, Calif.
Childerhose, Edward MacCabe (J'45) (CLMNO), Indus. Engr., Paint & Varnish Div., Canadian Industries, Ltd., Foot of Laughton Ave., Toronto 9, Ont., Can.
Childers, Harry Frederick ('44) (ACS), Plant Engr., Colgate-Palmolive-Peet Co., 17th & Kansas Ave., Kansas City, Kan.
Childs, C. Walter ('22) (DLMOJ), Plant Engr., Scovill Mfg. Co., 99 Mill St., Waterbury 91; *for mail,* 25 Steuben St., Waterbury 61, Conn.
Childs, Geo. D. (J'38), 6317 Broadway Terrace, Oakland, Calif.
Childs, Jack William ('44) (NCOJAZ), 1st Lt., U.S.M.C.; *for mail,* 303 Peachtree Battle Ave., Altanta, Ga.
Childs, John Norris ('43) (BJY), V.P., Metalweld, Inc., 2617 Hunting Park Ave., Philadelphia, Pa.
Chilton, Thomas Hamilton ('41) (KLBHZJ), Mgr., Devel. Engrg. Div., Engrg. Dept., E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Chin, Thomas Wing-Fu (J'45), Draftsman, Polaroid Corp., 12 Ames St., Cambridge; *for mail,* 475 Broadway, Chelsea 50, Mass.
Chinn, George I. (J'46) (BHK), Gerotor Pump Div., Gerotor May Corp., Baltimore 3, Md.
Chiodo, Chester H. (J'41) (AEH), 540 Thompson Pl., San Antonio, Tex.
Chipman, Fred W. ('19) (CMS), Treas., Internatl. Engrg. Works, Inc., 553 Waverly St., Framingham, Mass.
Chiras, David ('35; '42) (BCNLJ), Borematic Engrg. Dept., Heald Mch. Co., Worcester 6; *for mail,* 915 Providence St., Whitinsville, Mass.
Chirgwin, Allan Bixby (J'44), Mech. Engr., Farrel-Birmingham Co., Inc., Ansonia; *for mail,* 290 Yale Ave., New Haven, Conn.
Chisholm, Charles Rounds ('34) (HJMS), Ch., Repair Cost Appraisal Sec., War Shipping Admin., 39 Broadway, New York 6, N.Y.; *for mail,* 918 Foulkrod St., Philadelphia 24, Pa.
Chisholm, James (J'41) (BFJ), Lt., Co. D, 540th Engr. Regt., A.P.O. 758, c/o Postmaster, New York, N.Y.
Chittenden, George Iden ('24; '35) (DFOS), Mech. Engr., George S. Rider Co., 3800 Terminal Tower, Cleveland 13, Ohio.
Chivens, Clyde Cuthbertson (J'35), Partner, Boller & Chivens, 111-B Mission St., South Pasadena; *for mail,* 2274 Queensberry Rd., Pasadena 7, Calif.
Chizmarik, Joseph Henry (J'37), Indus. Engr., Wigton-Abbott Corp., South St., Plainfield; *for mail,* 215 Madison Ave., Dunellen, N.J.
Chodorowski, Emil Gregory (J'45) (ACO), Ch. Inspr., Masell Mfg. Corp. (Cole Steel Equip. Co., Inc.), 195 Front St., Brooklyn; *for mail,* 12-33—154th St., Beechhurst, N.Y.
Chont, Daniel G. (J'37), Excello Corp.; *for mail,* 8334 Vanderbilt Ave., Detroit, Mich.
Choporis, Peter Nicholas ('44), Mech. Engr., Birk & Choporis, 30 N. La Salle St., Chicago 2, Ill.
Chow, Ming Cheng ('45) (CML), 629 W. 115th St., New York 25, N.Y.
Chownes, William H. P. (J'45), Lt., U.S.N.R., Naval Air Sta., Box 34, Jacksonville, Fla.
Chowning, James Reid ('43; '45) (HLCSG), Design Engr., Vessels & Piping, Atlas Powder Co., Wilmington 99, Del.
Choyce, Nicholas B., Sr. ('44) (CDMSY), Charge Engrg. Maint., Loft Candy Corp., 38-38—9th St., Long Island City; *for mail,* 41-66—77th St., Jackson Heights, N.Y.
Chrisman, John Lyle (J'35), Devel. & Research Engr., Plastic & Rubber Prod. Co., 2100 Hyde Park Blvd., Los Angeles 44; *for mail,* 421 N. Gregory St., Whittier, Calif.
Christ, Walter O. (J'42) (DCMJ), Prod. Mgr., Silent Hoist & Crane Co., 841—63rd St., Brooklyn 20; *for mail,* 7123—4th Ave., Brooklyn 9, N.Y.
Christensen, Boyd Allison (J'44) (CNMLBR), 1624 Linden, Oklahoma City 8, Okla.
Christensen, Harvey Devon (J'44), 830 E. 66th, Seattle 5, Wash.
Christensen, Melford Carl (J'44), 346 Lake St., Manistique, Mich.
Christensen, Roy Hodge (J'43) (ABJMS), Stress Analyst, Douglas Aircraft Co., Inc., Ocean Park Blvd., Santa Monica; *for mail,* 587½ S. Ogden Dr., Los Angeles 36, Calif.
Christensen, Sabinus Hoegsbro (J'41) (BK), Research Physicist, Carrier Corp., 300 S. Geddes St.; *for mail,* 257 Furman St., Syracuse 1, N.Y.
Christerson, Paul David (J'46) (CMW), Jr. Engr., Willamette Iron & Steel Corp., 3050 N.W. Front St.; *for mail,* 3506 N.E. 70th St., Portland, Ore.
Christian, Chas. G. (J'38), 401 Lathrop St., Lansing, Mich.
Christie, Alexander Graham ('07; '16; F'36) (SFCKE), *Manager,* '22-'25, *Vice-President,* '25-'27, *President,* '39; Prof. Mech. Engrg., Johns Hopkins Univ., Charles & 34th St., Baltimore 18, Md.
Christie, Alexander Sidney Hugh (J'44) (NJWB), Design Engr., Lawrence Mfg. Co., Ltd., 595 W. 8th Ave.; *for mail,* 1025 W. 9th Ave., Vancouver, B.C., Can.
Christie, Robert W. (J'44) (JKT), 722 Hollywood Ave., Hohokus, N.J.
Christie, William Donald ('29; '37) (SFEO), Cons. Engr., 156 E. 42nd St., New York 17, N.Y.; *for mail,* 107 Southern Pkwy., Ridgewood, N.J.
Christman, Matthias Vincense (J'44) (ACMBDN), Math. Instr. Aero. Engrg., Northrop Aero. Inst., Hawthorne; *for mail,* 8211½ Crenshaw Dr., Inglewood, Calif.
Christmann, John Louis (J'37) (BDNZL), Mech. Engr., Product Devel. & Design, Merrick Scale Mfg. Co., 180 Autumn St.; *for mail,* 14 Temple Pl., Passaic, N.J.
Christoffersen, William Lawrence (J'37) (ZAOHL), Instrument Engr., Socony-Vacuum Oil Co., Inc., 400 Kingsland Ave., Brooklyn 22; *for mail,* 105-25—93rd St., Ozone Park 17, N.Y.
Christoph, Otto Karl (J'39) (CEFMTS), Dir., Paul J. Christoph Co., Caixa Postal 687, Rio de Janeiro, Brazil, S.A.
Christy, Howard A. (J'39) (AOZ), Engr., Sperry Gyroscope Co., Brooklyn, N.Y.; *for mail,* 31 S. Calvert St., Baltimore 2, Md.
Christy, William Gaston ('19) (FORSEC), *Manager,* '41-'44; Smoke Abatement Engr., Charge of Dept. of Smoke Regulation, Hudson County, Court House, Jersey City 8, N.J.
Chrsconovic, Nickolai (J'43) (BKFNAO), Stress Analyst, N. Am. Aviation, Inc., Inglewood, Calif.; *for mail,* 1325 Noble Ave., New York 60, N.Y.
Chryst, William Albert ('17) (MAEJ), Cons. Engr., Delco Products Div., Gen. Motors Corp., 329 E. 1st St., Dayton 2, Ohio.
Chung, Jackson Chia-Shan (J'42) (MCDJ), Prod. Engr., Studebaker Corp., Main St., South Bend 27, Ind.
Chun-Ming, William (J'43), 1120 Palama St., Honolulu 7, T.H.
Churan, James Frank (J'43) (MLC), U.S.N., 4727—39th Pl., Long Island City 4, N.Y.
Church, Austin H. ('28; '34; '36) (BHS), Assoc. Prof. Mch. Design, N.Y. Univ., Univ. Heights, New York 53, N.Y.
Church, Burton Arthur (J'41) (YSKHC), Engr., Croll-Reynolds Co., 17 John St., New York 7; *for mail,* 167 Radford St., Yonkers 5, N.Y.
Church, Edwin F., Jr. ('12) (BEKS), Prof. Mech. Engrg., Head of Dept., Poly. Inst. of Brooklyn, 85 Livingston St., Brooklyn 2, N.Y.
Church, H. D. ('13) (ERA), Retired; Route 1, Siesta Key, Sarasota, Fla.
Church, Hollis Fayette, Jr. (J'42) (MKFCYT), Products Engr., Miller Co., 99 Center St., Meriden; *for mail,* South Windsor, Conn.
Church, Maynard Dickson ('12) (SCJMNB), V.P., Worthington Pump & Mchy. Corp.; *for mail,* 392 N. Main St., Wellsville, N.Y.
Churchill, Alan W. (J'45) (MCZJNB), Engr., Charge Valve Engrg. Dept., Automatic Switch Co., 41 E. 11th St., New York 3, N.Y.
Churchill, Arnold J. (J'36) (BSLZ), 23 Princeton Ave., Swarthmore, Pa.
Churchill, Lawrence S. (J'44) (BSKFZ), Instr. Mech. Engrg., Speed Scien. Sch., Univ. of Louisville, Louisville 8, Ky.
Churchill, Warren Roy, Jr. (J'44) (CDHMLZ), Mgr., New Constr. Dept., Porter Bldg. Equip Co., 6505 Euclid Ave., Cleveland 3; *for mail,* 1456 E. 112 St., Cleveland 6, Ohio.
Churchman, Henry Nelson (J'45) (JRS), Sr. Draftsman, Glenn L. Martin Co., Middle River, Baltimore; *for mail,* 849 Brunswick Rd., Baltimore 21, Md.
Churgin, Leopold ('44), 203 W. 94th St., New York 25, N.Y.
Chute, Robert Eugene (J'42) (CMJKWS), Asst. to M.M., Am. Can Co., 184 Read St.; *for mail,* 100 Read St., Portland 5, Me.
Chvosta, Jerry Frank (J'42) (ANSKOC), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Mun. Airport, Cleveland; *for mail,* 16721 Lonmond Blvd., Shaker Heights 20, Ohio.
Ciba, John Lawrence (J'44) (MCJYNZ), Mech. Engr., Aireon Mfg. Corp., Fairfax & Funston Rd., Kansas City 15, Kan.
Cibulka, Alois ('42) (O), Pres., Internatl. "RST" Engrg. Co., Highlands, Tex.
Cichelli, Andrew Edward (J'43) (JLSE), Lubricating Foreman, Gen. Mech. Dept., Bethlehem Steel Co.; *for mail,* 802 E St., Sparrows Point 19, Md.
Cieslik, Walter J. (J'41) (EMS), 3018 W. 15th St., Cleveland, Ohio.
Cimolino, Gerald Natale (J'45) (EJL), Ensign, U.S.N.R., 56th Naval Constr. Bn., F.P.O., San Francisco; *for mail,* 629 N. Serrano Ave., Los Angeles 4, Calif.
Ciricillo, Samuel Francis (J'42) (NEGKF), Mech. Design Engr., Gen. Elec. Co., Lawrence St., Bloomfield; *for mail,* 23—6th Ave., Newark 4, N.J.
Cirrito, Anthony J. (J'39) (ABK), Mech. Engr., Drever Co., 748 E. Venango St., Philadelphia; *for mail,* 932 Lancaster Ave., Bryn Mawr, Pa.
Cisler, Walker Lee ('23; '36) (SCFHE), Ch. Engr., Power Plants, Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Citron, Elmer Henry ('43) (DFHJKL), Mech. Research Engr., Research Div., Eagle Picher Lead Co., Smelter Hill; *for mail,* 1902 Harlem, Joplin, Mo.
Cizek, Albert W., Jr. (J'35) (WYKFRS), Assoc. Matl. Engr., Bur. of Ships, Navy Dept., Matl. Lab., N.Y. Naval Shipyard, Brooklyn; *for mail,* 27-42 Curtis St., East Elmhurst, N.Y.
Clade, Robert ('19; '35) (BCL), Works Mgr., Am. Car & Fdy. Co., 1501 E. Ferry Ave., Detroit 11; *for mail,* 100 Burlingame St., Detroit 2, Mich.
Claessens, Frank A. ('43; '45) (BHL), Mech. Engr., Charge of Design & Mech. Lab., DeBell & Richardson, Box 240, Springfield; *for mail,* 69 Hale St., West Springfield, Mass.
Claffey, Charles John (J'44) (NKJOH), Engr. Draftsman, Tenn. Eastman Corp., Y-12 Area; *for mail,* 219 Greeley Hall, Oak Ridge, Tenn.
Clain, Charles Edward (J'41), Sr. Gen. Mechanic, Post Engr., Air Base, Mitchel Field; *for mail,* Mitchel Gardens, 16 Kenwood Pl., Hempstead, N.Y.
Clancy, Jas. R. ('24; '32; '35), Commercial Engr., N.Y. & Queens Elec. Light & Power Co., 28-19 Bridge Plaza N., Long Island City; *for mail,* 147-37—9th Ave., Whitestone, L.I., N.Y.
Clapp, William Howard ('14) (BJMNY), Prof. Emeritus, Mch. Design, Calif. Inst. of Tech., Pasadena; *for mail,* 95 S. Mentor Ave., Pasadena 5, Calif.
Clapper, James Howard (J'44) (ACESF), Lt. (j.g.), U.S.N.R., Amphibious Forces; Exec. & Engrg. Officer, LSM-367, F.P.O., New York, N.Y.; *for mail,* 5103 Cass St., Omaha 3, Neb.
Clarage, Harry L. (J'35), R.R. 3, Wall Lake, Delton, Mich.
Clark, Addison L. ('15), Pres., Am. Brake Shoe & Fdy. Co., 1010 Russ Bldg., San Francisco, Calif.
Clark, Albert Bernard ('15) (FKS), Partner, Sargent & Lundy, 140 S. Dearborn St., Chicago, Ill.
Clark, Albert George (J'43) (CKS), Devel. Engr., Engrg. Gen. Dept., Rm. 333, Bldg. 22, Gen. Elec. Co., 1 River Rd.; *for mail,* 1417 Union St., Schenectady, N.Y.
Clark, Albert Leslie (J'36) (COMKFS), Mgr., Specialty Dept., U.S. Radiator Corp., United Artist Bldg., Detroit, Mich.
Clark, Arthur Joseph, Jr. (J'43) (BAS), Jr. Stress Analyst, Dept. 81, Ranger Aircraft Engs. Div., Fairchild Eng. & Airplane Corp., Farmingdale, L.I., N.Y.
Clark, Cantwell, IV (J'42) (CMNTZ), Research Engr., E. I. du Pont de Nemours & Co.; *for mail,* 665 Walnut St., Waynesboro, Va.
Clark, Charles Gordon ('30; '39) (LSFHJK), Ch. Engr., Atlantic Sugar Refineries, Ltd.; *for mail,* 201 Duke St., St. John, N.B., Can.
Clark, Charles James ('44) (HKLSOY), Mgr., Pump & Condenser Dept., Ingersoll-Rand Co., 400 W. Madison St., Chicago 6, Ill.
Clark, Claude Lester ('45) (JOLEBS), Research Metal. Engr., Timken Roller Bearing Co., Canton 6, Ohio.
Clark, D. S. ('40) (CRS), Asst. Prof. Mech. Engrg. & Gen. Engrg., Purdue Univ.; *for mail,* 511 Russell St., West Lafayette, Ind.
Clark, David L., Jr. (J'41) (ACP), 255—26th St., Hermosa Beach, Calif.
Clark, Donald Sherman ('30; 37) (JMB), Assoc. Prof. Mech. Engrg., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.
Clark, Edward Ogden (J'41), Stenton & Abington Aves., Chestnut Hill, Philadelphia 18, Pa.
Clark, Ellery Darracott (J'39) (BCJLMY), 1st Lt., Ord. Dept., U.S.A., Asst. Shop Officer, Watertown Arsenal; *for mail,* 75 Pearl St., Watertown 72, Mass.
Clark, Ernest Ellsworth ('29; '35) (CSMEFL), Pres., Ch. Engr., Clark Indus. Co. of N.Y., 60 E. 42nd St., New York 17; *for mail,* Apt. J-420, 105 Carlton Ave., Brooklyn, N.Y.

Clark, Frank G. ('27), Gen. Foreman, Boilermakers & Welders Dept., Stand. Oil Co. of La.; *for mail,* 828 North Blvd., Baton Rouge, La.
Clark, Frank Henry ('10) (RJCFM), Retired; Box 485, Chalmette, La.
Clark, Frank Shaw ('09; '12; F'38) (FOS), Cons. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Clark, Frederic Baylis, Jr. (J'39) (CBR), Trackwork Engr., Ramapo Ajax Div., Am. Brake Shoe Co., 3355 E. Slauson Ave., Los Angeles 11; *for mail,* 1931 E. Washington St., Pasadena 7, Calif.
Clark, George C., Jr. (J'41) (HM), Devel. Engr., Wallace & Tiernan Co., Inc., 11 Mill St., Belleville, N.J.; *for mail,* 9 Hilltop Rd., Bronxville, N.Y.
Clark, George Marvin, Jr. (J'45) (CLY), Ensign, U.S.N.R., 238 Rusk St., Pittsburg, Tex.
Clark, George Stuart ('28; '40) (CLMED), *Student Award,* '24; Project Engr., Bechtel Bros. McCone Co., 220 Bush St., San Francisco 4; *for mail,* 2519 Hawthorne Terrace, Berkeley 8, Calif.
Clark, Harry K. (J'38), Pres., Automatic Combustion Equip. Co., Inc., 49 South Ave., Rochester 4, N.Y.
Clark, Henry Livingston, Jr. (J'38) (YMCND), Lt. (j.g.), U.S.N.R., Asst. Design Supt. (Ord.), Naval Shipyard, Boston 29; *for mail,* 550 Washington St., Winchester, Mass.
Clark, Henry Lurkins ('41) (DLCNJ), Mill Mech. Foreman, U.S. Vanadium Corp., Bishop, Calif.
Clark, James McAlister (J'44) (ASEMFC), Corps of Engrs., U.S.A., Geodetic Computer, 2766th Engr. Photomapping Co., 29th Engr. Base Topograhic Bn., A.P.O. 75, c/o Postmaster, San Francisco, Calif.; *for mail,* 232 Rusk St., Pittsburg, Tex.
Clark, James Montgomery ('28; '41) (ABMCYH), Asst. Gen. Mgr., Mfrs. Aircraft Assn., Inc., 30 Rockefeller Plaza, New York 20, N.Y.
Clark, John Jay (J'42) (ABE), Jr. Exper. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St.; *for mail,* 19 Scott Circle, East Hartford 8, Conn.
Clark, Joseph Peter ('37; '46) (CJM), Foreman, Cable Dept., Crescent Insulated Wire & Cable Co., Trenton, N.J.; *for mail,* 1 Barnesley Ave., Morrisville, Pa.
Clark, Leo Edward ('43; '45) (DERSYM), Application Engr., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.
Clark, Myron Henry ('13) (CDJ), Propr., Myron Clark Associates; *for mail,* Davis Rd., Concord, Mass.
Clark, Peter J. ('32), Draftsman, Fire Dept., City of N.Y., 3202 Queens Blvd., Long Island City; *for mail,* 155 Willoughby Ave., Brooklyn, N.Y.
Clark, Ralph William (J'46), Duquesne Light Co., Pittsburgh; *for mail,* 109 E. 6th St., Oil City, Pa.
Clark, Raymond Edward ('27; '40) (ZCMSFH), Mgr., Contract Engrg., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail,* 956 Nela View Rd., Cleveland Heights 12, Ohio.
Clark, Robert Allan (J'42) (HKJM), Ensign, U.S.N.R., North St., Middlebury, Conn.
Clark, Robert Brock (J'38) (SHBM), Mech. Engr., Gen. Elec. Co., 166 Broad St., Fitchburg, Mass.
Clark, Samuel Welsh ('33) (CFKSZ), Ch. Engr., Monterrey Ry., Light & Power Co., Apartado 917, Monterrey, N.L., Mex.
Clark, Theobald Forstall ('23) (LSBFKZ), Plant Supt., Philadelphia Gas Works Co., 31st & Passyunk Ave., Philadelphia 45, Pa.
Clark, Thomas Charles, Jr. (J'44) (ABC), Controls Group Engr., Curtiss-Wright Corp., Robertson; *for mail,* 7612 Maryland, Clayton, Mo.
Clark, Volney O. ('38) (CM), 1205 Thayer St., Los Angeles 24, Calif.
Clark, Wallace ('19; '21; F'45) (CM), Sr. Partner, Wallace Clark & Co., 521—5th Ave., New York 17, N.Y.
Clark, Walter R. ('08; '14) (CDJMHF), V.P., Charge Engrg., Bridgeport Brass Co., 30 Grand St., Bridgeport 2, Conn.
Clark, Walter William ('30; '35) (CMY), Product Engr., Sprague Elec. Co., Beaver St., North Adams, Mass.
Clark, Wilford Harold (J'43) (AFS), c/o John Ingalls, R.F.D. 1, Shelby, Neb.
Clark, William Gibson ('45) (EMA), Ch. Lub. Engr., Pure Oil Co., 35 E. Wacker Dr., Chicago 1, Ill.
Clark, William H. D. (J'43), 540 Dominion Sq. Bldg., Montreal, Que., Can.
Clark, William Lilburn, Jr. (J'46) (HMO), U.S.N.R., 1931 Sussex Pl., Toledo 7, Ohio.
Clark, William Sackett (J'42) (NOFCEJ), Mech. Engr., Office of Chief of Engrs., War Dept.; *for mail,* Apt. 43, 1322—15th St., N.W., Washington, D.C.
Clark, William Thomas (J'42) (EFA), U.S. Maritime Serv., 107 E. Ridley Ave., Ridley Park, Pa.
Clarke, Allen W. ('19) (RBJ), Asst. Gen. Mech. Engr., Am. Car & Fdy. Co., St. Charles, Mo.
Clarke, C. Edwin (J'12) (CFM), Asst. Gen. Mgr., Bethlehem Steel Co., Sparrows Point; *for mail,* 222 St. Dunstans Rd., Baltimore 12, Md.
Clarke, Charles Michael ('29; '35) (ES), Mchy. Engr., Fidelity & Casualty Co. of N.Y., 80 Maiden Lane, New York; *for mail,* 90 New York Ave., Baldwin, N.Y.
Clarke, Charles W. E. ('17; '13; F'38) (FKS), V.P., Cons. Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Clarke, Charles William (J'38) (MCAJ), Ch. Prod. Engr., Airesearch Mfg. Co., 9951 Sepulveda Blvd.; *for mail,* 8117 Croydon Ave., Los Angeles 45, Calif.
Clarke, Eugene C. ('21; '35) (ABCN), Pres., Gen. Mgr., Chambersburg Engrg. Co., Derbyshire St., Chambersburg, Pa.
Clarke, Harry Gidley ('45), Dawson & Dobson, Ltd.; *for mail,* Box 7764, Johannesburg, S. Africa.
Clarke, Philip C. ('38; '44) (BMNZ), Asst. Gen. Mgr., Hunter Pressed Steel Co., Lansdale, Pa.
Clarke, Philip L. ('07; '20) (CSL), Engr., So. Calif. Gas Co., 810 S. Flower St., Los Angeles 14; *for mail,* 1530 Cleveland Rd., Glendale 2, Calif.
Clarke, Sidney George ('22; '26) (BNMJCG), Chief, Engrg. & Estimating, Hamilton Gear & Mch. Co., 76 Van Horne St., Toronto 4; *for mail,* 245 Arlington Ave., Toronto 10, Ont., Can.
Clarke, Walter B. (J'40) (SOKFE), Capt., Trans. Corps, U.S.A., Commanding Officer, 860th Army Mar. Ship Repair Co., A.P.O. 980, Seattle, Wash.; *for mail,* 1085 Jefferson Ave., Akron 2, Ohio.
Clarke, Walter J. ('14), Dist. Mgr., Anchor Packing Co., 100—6th Ave., New York, N.Y.; *for mail,* 398 Thornden St., South Orange, N.J.
Clarke, Warren Hines ('21; '37) (CAD), Propr., Warren H. Clarke & Co., Box 28, Grand Cent. Annex, New York 17, N.Y.
Clarke, Wm. E. (J'37), Lt., U.S.N.R.; *for mail,* 29-05—39th Ave., Long Island City, N.Y.
Clary, Edward L. (J'41) (MLJZAD), Process Engr., Bldg. 85, Buick Motor Div., Gen. Motors Corp., Flint 2; *for mail,* 2701 Walter St., Flint 4, Mich.
Clary, Frank Alfred, Jr. (J'33) (NMTZJD), V.P., Arthur G. Russell Co., Inc., 16 Jeannette St., Forestville, Conn.
Clausen, Arthur W. ('22), Mech. Engr., 517 Hurlingham Ave., San Mateo, Calif.
Clausen, Henry Kristian (J'43) (CONHWL), Engr. & Cannery Foreman, Salmon Dept., Libby McNeill & Libby, 88 Hamlin St., Seattle 2; *for mail,* 7535 Jones Ave., N.W., Seattle 7, Wash.
Clausen, Jens ('21), Asst. Gen. Mgr., Harrisburg Pipe & Pipe Bending Co., Harrisburg; *for mail,* R.D. 1, Camp Hill, Pa.
Clauss, Julius A. ('28) (BDFJKO), V.P., Charge Engrg., Great Lakes Steel Corp., Ecorse, Detroit; *for mail,* 17587 Birchcrest Dr., Detroit 21, Mich.
Clautice, William Edward (J'43), Ensign, U.S.N.R., 1409 Geary St., San Francisco, Calif.
Clavin, C. Grattan (J'39) (FSMCOR), Capt., Solid Fuels Sec., G-4, Supreme Hdq., Allied Expeditionary Forces, A.P.O. 757, c/o Postmaster, New York, N.Y.
Clay, James Ashton, Jr. (J'45) (JOP), V.P., Serv. Engrs., Inc., 1st Natl. Bank Bldg., Ft. Worth 2, Tex.
Clayshulte, Nelson, Jr. (J'41) (MDABJE), Mesilla, New Mex.
Clayson, Gordon Silver ('44) (CBZNJ), V.P., Gen. Mgr., Wm. Ainsworth & Sons, Inc., 2151 Lawrence St., Denver 2, Colo.
Clayton, Jean Paul ('11; '26) (SRFHEA), Ch. System Officer, Commonwealth Edison Co., 72 W. Adams St., Chicago 45; *for mail,* 1153 Pine St., Winnetka, Ill.
Clayton, Lewis John ('27) (YNHMCG), Mgr., Engrg. Div., Viceroy Mfg. Co. Ltd., 345 Royce Ave., Toronto; *for mail,* 88 Clendenan Ave., Toronto 9, Ont., Can.
Clayton, Mark Manning ('41) (SOKFMC), Sr. Indus. Engr., Bonneville Power Admin., 811 Oregon; *for mail,* 1440 N.E. 59th Ave., Portland 13, Ore.
Clearwaters, Walter Lowell (J'43) (KAHSB), Project Engr., Research & Devel., U.S.N. Underwater Sound Lab., Ft. Trumbull, New London, Conn.
Cleary, George Francis (J'45) (EHS), Mar. Engr., U.S. Maritime Comm., Financial Center Bldg., 14th & Franklin St., Oakland; *for mail,* 42 Dorchester, San Leandro, Calif.
Cleary, James Warren (J'42) (FSL), Engr., Hercules Powder Co., Parlin, N.J.
Cleaver, William Henry ('42) (BJM), Asst. Ch. Engr., Samuel M. Langston Co., 6th & Jefferson St., Camden; *for mail,* 237 W. Pine St., Audubon, N.J.
Cleeves, William D. (J'35) (BHL), Draftsman, Tar & Chem. Div., Koppers Co., Inc., 200 Flannery Bldg., Pittsburgh 13; *for mail,* 1120 Milton Ave., Pittsburgh 18, Pa.
Clegg, Clarence Merrill (J'43) (AM), Lt., Army Air Forces, Aircraft Maint. Officer, Ferrying Div., Air Trans. Comd., Officers' Mail Rm., Army Air Base, Charleston, S.C.; *for mail,* 101 Euclid Ave., Haddonfield, N.J.
Clegg, Douglas (J'40) (ABM), 2 Duck Pond Rd., Glen Cove, N.Y.
Cleghorn, Mark Perkins ('14) (SEF), Prof. Mech. Engrg., Iowa State College; *for mail,* 513 Ash Ave., Ames, Iowa.
Clem, John M. (J'42) (EFBCJ), Engr., Fairbanks, Morse & Co.; *for mail,* 841 Elm St., Beloit, Wis.
Clemens, Abraham B. ('04) (BNEF), Dean Emeritus, Schools of Tech., Internatl. Correspondence Schools, 1000 Wyoming Ave., Scranton; *for mail,* 63 N. Franklin St., Wilkes-Barre, Pa.
Clemens, Alonzo W. ('20) (KBDEFM), Designing Engr., Bethlehem Steel Co.; *for mail,* 322 E. Goepp St., Bethlehem, Pa.
Clemens, Joseph D. (J'39), Major, Box 311, R.R. 1, Lincoln Ave., St. Joseph, Mich.
Clement, Edward J. ('45) (ACD), Tech. Editor, *Mill & Factory,* Conover Mast Corp., 205 E. 42nd St., New York 17; *for mail,* 10 Washington St., Hempstead, N.Y.
Clement, George Herbert (J'41) (ABC), 421—4th St., Manhattan Beach, Calif.
Clement, Geo. P. (J'38) (SHKR), Exper. Test Engr., Worthington Pump & Mchy. Corp., Harrison, N.J.; *for mail,* 155 Audubon Ave., New York, N.Y.
Clement, Richard Thomas (J'41) (YNBAGM), 129 Sycamore St., Pittsburgh 11, Pa.
Clement, Richard W. (J'37) (FJS), Student Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* Apt. 3D, 191 W. 102nd St., New York 25, N.Y.
Clement, Walter J. ('19; '32) (CJL), Ch. Engr., Pulverizing Div., Bossert Co., Inc., Box 358, Utica, N.Y.
Clement, William David (J'42) (AES), 13 Reservoir St., Nashua, N.H.
Clements, Roger William (J'46) (BER), Lt., U.S.N.R.; 1915 Hennepin, Minneapolis, Minn.
Clemow, George A. (J'40) (KBNSJL), Jr. Engr., Research Lab., Linde Air Products Co., East Park Dr. & Woodward Ave.; *for mail,* 431-B East Park Dr., Tonawanda, N.Y.
Clennon, John Patrick (J'44) (BLM), 538 S. Humphrey Ave., Oak Park, Ill.
Cleveland, Robert Linsted, Jr. (J'45) (SNMEJC), Mech. Engr., J. L. Lucas & Son, Inc., Post Rd.; *for mail,* 171 Rockland Rd., Bridgeport, Conn.
Clevenson, Sherman A. (J'46), 313 Chapel St., Hampton, Va.
Cliff, Richard Henry, Jr. (J'43) (BZKFEG), Sales Engr., Orr & Sembower, Inc., Morgantown Rd.; *for mail,* 812 N. 3rd St., Reading, Pa.
Cliffe, Edwin Lee ('41) (LMC), West. Div. Engr., Pa. Salt Mfg. Co. of Wash., 2901 Taylor Way, Tacoma 2, Wash.
Clifford, John James (J'45) (BEHKNA), Designer, Foster Wheeler Corp., 117 Liberty St., New York; *for mail,* 3536 Cambridge Ave., New York 63, N.Y.
Cline, Charles Aaron (J'43) (ABJHY), Pvt., Hdq. Btry., 213th Field Arty. Bn., c/o Postmaster, Harpers Ferry, W.Va.
Cline, William Everette (J'41) (EFHKSZ), Capt., U.S.A., Maint. Officer, Armd. Force; *for mail,* S. Main Ave., Newton, N.C.
Clinedinst, Wendel Waters ('37) (SKHLC), Engr., C. H. Wheeler Mfg. Co., 233 Broadway, New York 7, N.Y.
Clinedinst, William Ott ('33; '41) (BJHANE), Dir. of Research, Natl. Tube Co., 1702 Frick Bldg., Pittsburgh 30, Pa.
Clingerman, Robert Louis ('30) (ACOY), Cons. Engr., William Bayley Co., Springfield, Ohio; *for mail,* 3701 Massachusetts Ave., N.W., Washington 16, D.C.
Clokey, Allison Walter (J'43) (BJM), Engr. Piece Part Prod., West. Elec. Co., Inc., Locust St., Haverhill; *for mail,* 62 Upland Ave., Bradford, Mass.
Cloran, Everett Michael (J'42), Research & Cons. Engr., College of Engrg., Univ. of So. Calif., University Park, Los Angeles 7; *for mail,* 629 S. Grand Ave., Pasadena 2, Calif.
Close, Ralph G., Jr. (J'38), 552 Cane Run St., Harrodsburg, Ky.
Clough, Alvah B. ('41) (BHJ), Ch. Draftsman, Cuno Engrg. Corp., Vine St.; *for mail,* 267 Liberty St., Meriden, Conn.
Clough, Robert Charles (J'39) (NBMH), Mch. Designer, Ill. Tool Works, 2501 N. Keeler Ave., Chicago 39, Ill.; *for mail,* 309 E. Hanover St., Marshall, Mich.
Clough, Roy Eugene (J'40) (MLNJCY), Engr., G. C. Demetrak Co., 700 Minnesota St., San Francisco; *for mail,* Box 787, Route 2, Walnut Creek, Calif.
Clouse, John Henry ('30; A'31) (BZAHKW), Prof. Physics, Univ. of Miami, Coral Gables, Fla.
Clousing, Lawrence Adrian ('33; '36) (ABKHEZ), Engr., Test Pilot, Head of Flight Research, Natl. Advis. Com. for Aero., Moffett Field, Calif.
Clower, Marion Grove ('39; '45) (CFOS), New Constr. Engr., Joseph E. Seagram & Sons, Inc., Louisville; *for mail,* 2024 Village Dr., Louisville 5, Ky.
Clubbs, Bennett A. (J'41) (LWSHBC), Plant Engr., So. Kraft Div., Internatl. Paper Co., Panama City, Fla.
Clucas, George Worden ('15; '35) (CH), Pres., Stannard Power Equip. Co., 53 W. Jackson Blvd., Chicago 4, Ill.
Cluett, Albert Edmund ('00; '03) (RFT), Exec. V.P., Troy Savings Bank, Box 666, Troy, N.Y.
Cluett, Sanford Lockwood ('03; '11) (TBMJ), V.P., Cluett, Peabody & Co., Inc., 433 River St., Troy, N.Y.
Clune, Jos. P. ('18), Asst. to V.P., Solvay Process Co.; *for mail,* 171 Robineau Rd., Syracuse, N.Y.
Clutter, Carl E. (J'39) (JE), 368 Ivy St., Glendale, Calif.
Clutz, Claude S. (J'45) (ABNKJZ), Exper. Test Equip. Engr., Wright Aero. Corp., Wood-Ridge; *for mail,* F-36, Abbott Court, Fairlawn, N.J.

Cluverius, Wat Tyler ('40) (CLDHR), Rear Admiral, U.S.N., Retired; Pres., Worcester Poly. Inst., Worcester 2, Mass.
Clyman, Harold James (J'42) (CBA), Asst. to Mgr. of Engrg., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail*, 71 Forest Rd., Springfield, Del. Co., Pa.
Coakley, William E. ('24; '35) (CDM), Prod. Mgr., Pressed Steel Tank Co., 1445 S. 66th St., Milwaukee 14, Wis.
Coar, Richard John (J'42) (ZHNABK), Asst. Project Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail*, 29 Townley St., Hartford 5, Conn.
Coates, Henry Troth ('27) (FBMSK), Pur. Agt. & Engr., Dairymen's League Co-operative Assn., Inc., 11 W. 42nd St., New York 18, N.Y.
Coates, Robert Edwin (J'43) (ADM), Seaman, 1st Class, Radio Technician, U.S.S. *Diamond Head*, AE-19, F.P.O., New York, N.Y.
Coates, William ('42) (FS), Pres., Wil-Co-Lene Chem. Co., Inc., 744 Broad St., Newark; *for mail*, Apt. 401, 139 N. Arlington Ave., East Orange, N.J.
Cobb, Arnold Colvin ('41) (REF), Serv. Engr., Socony-Vacuum Oil Co., Inc., 59 E. Van Buren St., Chicago 5; *for mail*, 135 N. Wright St., Naperville, Ill.
Cobb, Dedrick Bieman, Jr. (A'45), Inspr., Hartford Steam Boiler Inspc. & Ins. Co., Box 1401; *for mail*, Azalea Dr., Pinecroft, Greensboro, N.C.
Cobb, Elton Thomas ('30) (BCM), Ch., Veterans Unit, War Assets Admin., Shrine Bldg.; *for mail*, 640 Keel Ave., Memphis 7, Tenn.
Cobb, James Stanley, Jr. (J'43) (KSAN), Ensign, (E)-L, U.S.N.R., Engr. Officer, F.P.O., San Francisco, Calif.; *for mail*, 514 W. Foster Ave., State College, Pa.
Cobb, Leland Davison ('44) (BCJMNT), Gen. Mgr. Staff, Cost Engr., New Departure Div., Gen. Motors Corp., N. Main St., Bristol; *for mail*, 88 Garden St., Forestville, Conn.
Cobb, Leonard A. ('29) (CEFKSZ), Gen. Supt. Steam Plants, No. States Power Co., 15 S. 5th St., Minneapolis, Minn.
Cobb, Willard Halsey (J'12) (BCH), Gen. Mgr., Mech. Goods Div., U.S. Rubber Co., 1230—6th Ave., New York 20, N.Y.
Coberly, Clarence James ('18; '35) (HBJMCE), Pres., Gen. Mgr., Kobe, Inc., 3040 E. Slauson Ave., Huntington Park, Calif.
Cobleigh, Henry Rice ('46) (EMC), Staff Mem., A.S.M.E., 29 W. 39th St., New York 18; *for mail*, 27 Crane Ave., White Plains, N.Y.
Cocagne, Arthur Charles (J'42), Designer, Douglas Aircraft Co., Inc., El Segundo; *for mail*, 8727 Glider Ave., Los Angeles 45, Calif.
Cochran, Alex. R. ('22; '35) (CDR), V.P., Gen. Mgr., Loiza Sugar Co., Central Canovanas, Canovanas, Humacao, P.R.
Cochran, Alexander Russell, Jr. (J'38) (LCKHZE), Indus. Engr., Shell Oil Co., Inc., Box 2527, Houston 1, Tex.
Cochran, Chas. B. (J'39) (SFZLC), Asst. Supt., Power Div., Westvaco Chlorine Products Corp., South Charleston 3, W.Va.
Cochran, David (J'38) (ZCABKE), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 1, N.Y.
Cochran, Edward Oswald (J'46) (CJS), 509 Gilmour St., Peterborough, Ont., Can.
Cochran, John Richard (J'44) (BYM), Engr., Research & Devel. Dept. 29, Maytag Co., Newton, Iowa.
Cochran, W. Keith (J'38) (MHA), Asst. Ch. Engr., Vendorlator Mfg. Co., 4000 Railroad Ave.; *for mail*, Box 32, Route 1, Fresno, Calif.
Cockburn, Robert Ellsworth, Jr. (J'41) (AKNO), Capt., Pilot, Army Air Forces; *for mail*, c/o C. M. Todd, Box 6, Route 2, Orland, Calif.
Cockram, William ('40) (CFKJ), Owner, Gen. Mgr., W. Cockram & Co., Walnut St., Paris, Ont., Can.
Coco, Rodney Glenn (J'43) (AMW), 3545 Maywood Ave., Baton Rouge 9, La.
Codd, Leo Antoine (A'44) (AGMSCR), Exec. V.P., Army Ord. Assn., 705 Mills Bldg., Washington 6, D.C.
Codding, E. Hale ('25) (EPS), Capt., U.S.N.R., 26 Bethune St., New York, N.Y.
Codrington, George W. ('29) (CEF), V.P., Gen. Motors Corp., Gen. Mgr., Cleveland Diesel Eng. Div., 2160 W. 106th St., Cleveland 11, Ohio.
Cody, Clifford S. ('37) (AKEFBS), Mech. Design Engr., Westinghouse Elec. Corp.; *for mail*, 1293 Bradley Rd., Springfield, Mass.
Coe, Frank Campbell ('19; '24) (FJKLOS), Firm of Frank Campbell Coe, Equip. Engrs., Commercial Trust Bldg., Philadelphia 2, Pa.
Coe, Robert Gustav (J'43) (MDYFA), Test Lab. Asst., York Corp., Roosevelt Ave.; *for mail*, 229 Carlisle Ave., York, Pa.
Coes, Harold Vinton ('07; '13; F'36) (COMLD), *Vice-President*, '27 & '32-'34, *Manager*, '29-'32, *President*, '43; V.P., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Coffey, Edward John, Jr. (J'42) (ACMEJO), 1130 N. New St., Bethlehem, Pa.
Coffin, Charles William ('43) (YLN), Mech. Engr., Hood Rubber Co. Div. of B. F. Goodrich Co., Nichols Ave., Watertown 72; *for mail*, 750 East St., Dedham, Mass.
Coffin, G. S. ('18; '35) (SEFHCK), 2301 Saymoure Rd., Wilmington 56, Del.
Coffin, Louis Fussell ('26) (JSNDCE), Supt., Mech. Dept., Bethlehem Steel Co.; *for mail*, 820 C St., Sparrows Point 19, Md.
Coffin, Louis Fussell, Jr. (J'39) (BJN), Instr., Mech. Engrg. Dept., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Coffin, Robert Samuel (A'44) (CLS), Pres., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio.
Cofoid, Wayne Edward (J'46), Tonica, Ill.
Cogan, Myles Honohan Resuggan (J'40) (OKYSRZ), Office Engr., Shell Chem. Corp., Deer Park; *for mail*, 515 Richmond Rd., Houston 6, Tex.
Coggeshall, Clayton S. ('40) (CKFS), Mgr., Turbine Div., Gen. Elec. Co., Schenectady 5, N.Y.
Coghlan, Sherman F. ('23; '35) (MCSHNO), Pres., J. M. Montgomery & Co., Inc., 306 W. 3rd St., Los Angeles 13; *for mail*, 414—9th St., Santa Monica, Calif.
Cohan, Alvin M. (J'41) (EANTS), Project Engr., Breeze Corp., 41 S. 6th St., Newark; *for mail*, 320—77th St., North Bergen, N.J.
Cohen, Abraham Saul ('15; '35) (ESOBNF), Mgr., Mech. Engr., N.L.C. Engrg. & Equip. Co., 101 Tremont St., Boston 8, Mass.
Cohen, Edgar Fabian ('42), 539 Webster Ave., New Rochelle, N.Y.
Cohen, Herman ('37), 209 Westminster Rd., Brooklyn 18, N.Y.
Cohen, Leo (J'45) (AES), Westinghouse Elec. Corp., South Philadelphia; *for mail*, 226 Dickinson Ave., Swarthmore, Pa.
Cohen, Leonard (J'44) (CAMOZ), Pvt., 1st Class, Army Air Forces, Sqd. A, 1503rd Base Unit, Air Trans. Comd., Hamilton Field, Calif.
Cohen, Lester (J'41) (GMS), 345 Hamilton Pl., Hackensack, N.J.
Cohen, Peter (J'44) (BEM), Mech. Engr., Elec. Tamper & Equip. Co. Ltd., Rm. 417, 1440 St. Catherine St. W.; *for mail*, 5215 Jeanne Mance St., Montreal, Que., Can.
Cohen, Robert (J'42), Capt., Ord. Dept., Office, Combined Maint. Shops, Camp Phillips, Kan.
Cohn, Gunther (J'45) (NFYBJ), Research Asst., The Pa. State College; *for mail*, 236 E. Foster Ave., State College, Pa.
Cohn, Ira A. (J'44) (AJM), Cpl., Hdq. Btry., 379th Field Arty. Bn., A.P.O. 102, c/o Postmaster, New York, N.Y.
Coifman, Lee Gregory (J'44) (BJAZNH), Engr., J. M. Lessells & Associates, 108 Massachusetts Ave., Boston 15, Mass.
Cokelet, William V. W. (J'41) (BJW), 298 Academy St., Jersey City 6, N.J.
Coker, Lawrence T. (J'42) (DMC), Fire Controlman, 3rd Class, U.S.N., U.S.S. *Bowie*, APA-137, F.P.O., San Francisco, Calif.; *for mail*, 1358 N. Clover Leaf Circle, Mobile 19, Ala.
Coker, Robert Howard (J'45) (SECFA), Power House Gen. Foreman, Plant 5, Allison Div., Gen. Motors Corp., Indianapolis; *for mail*, 4312 Norwaldo, Indianapolis 5, Ind.
Colburn, Allan Philip ('44) (KBYAFH), Prof. Chem. Engrg., Univ. of Del., Newark, Del.
Colburn, George, Jr. (J'45), Ensign, U.S.N., 7838—17th Ave., Kenosha, Wis.
Colby, Burton E. (J'39), 1022 S. 85th St., West Allis, Wis.
Colby, Clarence Edwin (J'43) (AEJ), 130th Naval Constr. Bn., F.P.O., San Francisco, Calif.; *for mail*, 25 Oak St., Penacook, N.H.
Colby, Elmer Wilford (J'35) (BEFNSZ), Ch. Power Engr., Keeler Brass Co., 955 Godfrey, S.W., Grand Rapids; *for mail*, 1327 Hope St., S.E., Grand Rapids 6, Mich.
Colby, Emanuel M. ('21; '26; '30) (CSO), Asst. Gen. Mgr., Park Cent. Hotel, 870—7th Ave., New York 19, N.Y.
Colby, Haldwell Selden ('27) (SFCJM), Exec. V.P., Malleable Founders Society, 1800 Union Commerce Bldg., Cleveland 14, Ohio.
Colclough, Otho T. (J'37) (CFK), Custodian, Am. Legation, Ottawa, Ont., Can.
Coldwell, Everett Sharples ('20; '26; F'38) (CMOA), V.P., Dir., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Cole, Arthur W. ('04; '11) (EFS), Retired; Barley Neck Rd., Orleans, Mass.
Cole, Carter Stanard ('30; '35) (RJK), Asst. Tech. Secy., American Society for Testing Materials, 260 S. Broad St., Philadelphia 2; *for mail*, 206 Greenwood Ave., Jenkintown, Pa.
Cole, Charles Bradford ('19; '26) (ACNMHJ), Pres., Tool Equip. Co., 24 S. Pulaski Rd., Chicago 24; *for mail*, Gary Mill Rd., Route 2, West Chicago, Ill.
Cole, Edward Smith ('97; '06; F'41) (CHZ), Pres., Pitometer Log Corp., 237 Lafayette St., New York 12, N.Y.; *for mail*, 133 Bellevue Ave., Upper Montclair, N.J.
Cole, Gilmoure Nichols ('31; '44) (ABCJMN), Ch., Design Operas., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford; *for mail*, 58 Stephen St., Manchester, Conn.
Cole, Harry (J'27) (REC), Supt. Mchy., Tenn. Coal, Iron & R.R. Co.; *for mail*, Box 316, Route 6, Bessemer, Ala.
Cole, Hugh Molloy Ormston ('44) (RMS), Capt., c/o H. O. Cole, 1060 Grosvenor Ave., Winnipeg, Man., Can.
Cole, James H. ('28; '34; '35) (CE), Stand. Oil Co. of Calif., Richmond; *for mail*, 7465 Curry Ave., El Cerrito, Calif.
Cole, Kenneth W. ('24; '35) (CFJ), Dist. Mgr., Pressed Steel Tank Co., 208 S. La Salle St., Chicago 4, Ill.
Cole, Lenord S. (J'35) (CMS), 705 N. Bishop, Dallas 8, Tex.
Cole, Lyndon C. ('27) (HJMNY), Ch. Engr., Cleveland H-P-M Engrg. Div., Hyd. Press Mfg. Co. (Mt. Gilead), 1012 Engrs. Bldg., Cleveland 14; *for mail*, 2676 St. James Pkwy., Cleveland Heights 6, Ohio.
Cole, Ralph Alfred (J'20) (CMBLDA), Factory Mgr., Motor Wheel Corp., Lansing; *for mail*, 505 Ardson Rd., East Lansing, Mich.
Cole, Robert Williams (J'36) (EFJRY), Mgr., Rocky Land Poultry Farm, Box 125; *for mail*, 39 Holden St., Attleboro, Mass.
Cole, Sidney I. ('28; '34; '37) (CDM), V.P., Treas., Charge of Engrg. & Sales, Indus. Erectors, Inc., 1316 W. Cermak Rd., Chicago 8, Ill.
Cole, William N. (J'41), Mech. Engr., Project Engrg. Gp., E. I. du Pont de Nemours Co., Buffalo; *for mail*, 73 Colonial Ave., Kenmore 17, N.Y.
Coleman, Ernest L. (A'22) (CDM), Mgr., Acctg. Systems Dept., Peat, Marwick, Mitchell & Co., 105 S. La Salle St., Chicago 3, Ill.
Coleman, Francis A. (J'44) (EMS), 45-27 Parsons Blvd., Flushing, L.I., N.Y.
Coleman, Glenn Alfred ('45) (AE), Plant Engr., Douglas Aircraft Co., Inc.; *for mail*, 3824 S. Victor, Tulsa, Okla.
Coleman, Harry Francis ('23; '25; '35) (WSLA), Contrg. Engr., Grinnell Co., Inc., 1017 Calliope St., New Orleans 13; *for mail*, 2123 Palmer Ave., New Orleans 15, La.
Coleman, Harry Shipp ('18) (CMZSR), Asst. Dir., Mellon Inst. of Indus. Research, 4400—5th Ave., Pittsburgh 13, Pa.
Coleman, John B. ('21), Ch. Engr., Charge Fire Protection Engrg., Grinnell Co., Inc., 260 W. Exchange St., Providence, R.I.
Coleman, Kenneth S. (J'41) (ACE); 17 William St., Kearny, N.J.
Coleman, Philip L. ('24; '32; '35), 850 Lake Shore Dr., Chicago, Ill.
Coleman, William Wheeler ('08), Chmn. of Bd., Bucyrus-Erie Co., South Milwaukee, Wis.
Coles, Victor L. (J'41) (MADCGB), 250 W. 103rd St., New York 25, N.Y.
Coles, Walter Francis, Jr. (J'42) (SLO), Jr. Engr., H. M. Wilson Co., 18th & Brandywine Sts., Philadelphia; *for mail*, 7420 Georgian Rd., Philadelphia 38, Pa.
Colesworthy, Joseph Bicknell ('45) (MDBHSL), Owner, Firm of J. B. Colesworthy, 1709 W. 8th St., Los Angeles 14; *for mail*, 1537 N. Orange Grove Ave., Los Angeles 46, Calif.
Colgin, Rufus Asbury ('41) (CBJM), V.P., Charge Plant Opera., W. C. Norris, Mfr., Inc., 10 N. Elwood Ave., Tulsa 1, Okla.
Coliz, James T. (J'43) (NAHWYO), Design Engr., Curtiss Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail*, 100 Spruce St., Newark 3, N.J.
Collar, Carl, Jr. (J'40) (HZS), Engr., Research & Devel. Dept., Worthington Pump & Mchy. Corp., Harrison; *for mail*, Apt. 1-C, A Bldg., S., 125 Prospect St., East Orange, N.J.
Collart, Kirk Street (J'42) (AJMNOY), Lt., Army Air Forces, 1951st Engr. Utility Co., A.P.O. 719, San Francisco, Calif.; *for mail*, 307 Mason St., Newark, N.Y.
Colles, George Wetmore ('95; '01) (DHM), Oper., Rosharon Waterworks, 1 Lochridge Rd., Rosharon, Tex.
Colley, Chas. J. ('24) (CFS), Mgr., Power Div., Monsanto Chem. Co., 1700 S. 2nd St., St. Louis, Mo.
Colley, Clay Thomas ('26; '32; '35) (DCOMLY), Engr., Link-Belt Co., 361 S. Anderson St., Los Angeles 33; *for mail*, 1846 W. 43rd St., Los Angeles 37, Calif.
Colliander, Carl Torsten (J'35) (JMNAY), Research Engr., Remington Arms Co., Inc., Ilion, N.Y.
Collier, John Howard ('44), Pres., Crane Co., 836 S. Michigan Ave.; *for mail*, 1260 N. Dearborn Pkwy., Chicago, Ill.
Collier, Robert Henry (J'41) (MDCW), Engr., Planning Dept., Internatl. Harvester Co., Lagonda Ave., Springfield, Ohio.
Collier, Thomas Austin, Jr. (J'46), c/o Pitzonka, Box 211, Bristol, Pa.
Collier, Wm. I. ('30), W. I. Collier & Co., 3414 Duvall Ave., Baltimore 16, Md.
Collignon, Mario (J'43) (CMLDZ), Sales Mgr., Grasas Vegetales S.A., Apartado 449; *for mail*, Ave. Union 57 Norte, Guadalajara, Jal., Mex.
Collins, Arthur Reynolds (J'43) (ABFJKM), Ch. Engr., S. Wind Div., Stewart-Warner Corp., 1514 Drover St., Indianapolis 7, Ind.
Collins, Bertrand Robson Torsey ('91; '01) (FS), Retired; Chebeague Island, Me.
Collins, Clifford Hiram (J'35) (BCEN), Asst. Ch. Engr., Gladden Products Corp., 635 W. Colorado Blvd., Glendale 4; *for mail*, 1634 W. 38th St., Los Angeles 37, Calif.
Collins, David Merrill (J'44) (JLW), Radio Technician, 3rd Class, U.S.N.R., Co. 15, Billet 119, Navy Pier, Chicago, Ill.
Collins, Dwight Robertson ('42) (R), Supt. Air Brakes, Denver & Rio Grande Western Ry., 1531 Stout, Denver 2; *for mail*, 3935 Chase St., Denver 14, Colo.

Collins, F. Alton ('10; '17; '35) (CMBN), Sales Engr., Hoover Ball & Bearing Co., 326 Hoover Ave.; *for mail,* 1705 Shadford Rd., Ann Arbor, Mich.
Collins, Frank Wyatt ('21) (BDHNY), Mech. Engr., Pan-Am. Engrg. Co., 820 Parker St., Berkeley 2, Calif.
Collins, Glenville Arthur ('38) (KO), Pres., Columbia Engrg. Co., 321 Platt Bldg., Portland 5, Ore.
Collins, Ivor Winter, Jr. (J'41) (MONC), Designer, Draftsman, Frederick Hart & Co., 837 Main St., Poughkeepsie, N.Y.
Collins, James Warner (J'45) (MHN), Jr. Engr., Guyan Mchy. Co., 751-53 Stratton St., Logan; *for mail,* Box 155, Stollings, W. Va.
Collins, John, Jr. ('21; '28) (LKFSC), Ch. Engr., Socony-Vacuum Oil Co., Inc., 400 Kingsland Ave., Brooklyn 22, N.Y.; *for mail,* 230—8th Ave., Paterson 4, N.J.
Collins, John Aloysius ('24) (FKS), V.P., Mutual Boiler Ins. Co. of Boston, 60 Batterymarch, Boston 10, Mass.
Collins, Laurence Joseph ('44) (BJO), Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 18 Brookhouse Dr., Marblehead, Mass.
Collins, M. R., Jr. (J'40) (APS), Crane Co., Nashville, Tenn.
Collins, Maurice James (J'44), Apt. 3, 1045 Wendell Ave., Schenectady, N.Y.
Collins, Winthrop Ingersoll ('43), Dist. Sales Mgr., Babcock & Wilcox Co., 2730 Koppers Bldg., Pittsburgh; *for mail,* 1029 Oldgate Rd., Wilkinsburg, Pa.
Collisson, Norman Harvey ('44) (CMJ), Capt., U.S.N.R., Office of Asst. Secy. of Navy, Rm. 2022, Navy Dept., Washington, D.C.
Collora, Nicholas Andrew ('33; '45) (CDFJKS), Engrg. Div., Commercial Solvents Corp.; *for mail,* 1028 S. Center St., Terre Haute, Ind.
Colomb, Clifford Francis ('41) (COMDZJ), Sales Engr., Truscon Steel Co., 1143 Canal Bldg., New Orleans 12, La.
Colony, C. Gordon (J'41) (CDJM), 2368 Victory Pkwy., Cincinnati, Ohio.
Colpitts, James Vandever ('32) (KL), Sales Engr., E. A. Kaestner Co., 516 N. Calvert St., Baltimore, Md.; *for mail,* Fair Brae Farm, R.D. 3, West Chester, Pa.
Colston, Robert ('21; '21; '35) (EFJ), Dist. Rep., Hagan Corp., Hall Labs., Inc., & Calgon, Inc., 53 W. Jackson Blvd., Chicago 4, Ill.
Colvin, Charles Herbert ('16; '25; F'45) (ACMZB), Engrg. & Admin. Consultant, 320 Central Park West, New York 25, N.Y.
Colvin, Fred Herbert ('95; '99; F'41) (ACM), *Worcester Reed Warner Medallist,* '42; 311 Lincoln Ave., Point Pleasant, N.J.
Colvin, James A. ('15; '25) (SKFECH), Mgr., Power Prod. & System Opera., No. States Power Co., 15 S. 5th St., Minneapolis 2, Minn.
Colvin, Timothy E. ('30; '44) (ACHM), Pres., Aerco Corp., 12024 Center St., Hollydale, Calif.
Colwell, Augustus Warren ('46) (AJL), Exper. Engr., A. O. Smith Corp., Milwaukee; *for mail,* 1207 N. 33rd St., Milwaukee 8, Wis.
Combe, F. A. ('20), Cons. Engr., 1188 Phillips Pl., Montreal, Que., Can.
Comber, William Robert (J'40) (CJM), Prod. Mgr., Bendix-Westinghouse Air Brake Co.; *for mail,* 398 N. Olive St., Elyria, Ohio.
Combes, C. L. (J'40) (ERS), Assoc. Editor, Simmons-Boardman Publ. Corp., 30 Church St., New York, N.Y.
Comly, G. Norwood ('05; '07), Pres., New England Concrete Pipe Corp., Newton Upper Falls, Mass.; *for mail,* Moylan-Rose Valley, Pa.
Compton, Fred Arthur ('39) (CFORSY), Pur. Agt., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 18427 Parkside Ave., Detroit 21, Mich.
Compton, Karl Taylor ('33), Pres., Mass. Inst. of Tech., Cambridge 39, Mass.
Compton, Robert B. ('22; '29) (CDGLMW), Chmn., Post War Planning Com., Gaylord Container Corp., 111 N. 4th St., St. Louis 2, Mo.
Compton, Wendell Cooper (J'37) (CMDL), 1432 Lawndale Rd., Elkhart, Ind.
Comroe, Irving Harold ('45) (JMNBCG), Prod. Engr., Frozen Bake Co., 1112 S. Akard St., Dallas 1, Tex.
Comstock, Charles Worthington ('08) (BHLOSZ), 3527—81st St., Jackson Heights, L.I., N.Y.
Comstock, James Floyd (J'41) (BSNKGC), Lt. (j.g.), U.S.N.R., Ch. Engr., U.S.S. *Caswell,* F.P.O., San Francisco, Calif.; *for mail,* Murray, Neb.
Comstock, Louis K. ('02; F'41), 3761 Social Security Bldg., Washington, D.C.
Conant, Wm. S. ('95; '04), Cons. Engr., 1402—31st St., N.W., Washington, D.C.
Concordia, Charles ('41) (BSAKEH), Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Condit, Kenneth Hamilton ('21; '28) (CM), *Manager,* '36-'39, *Vice-President,* '39-'41; Dean of Engrg., Princeton Univ., Princeton, N.J.
Condit, Merlyn Edwin (J'45) (OAYCLM), 534 N. Hoover Ave., Whittier, Calif.
Condon, John Edmund (J'45) (CSZNHB), U.S.N.R., 154 Oliver Rd., Newton 68, Mass.
Congdon, Donald Edgar ('44) (TSLOCF), Plant Engr., Textron Mills, Inc., Main St., Manville; *for mail,* 80 New Meadow Rd., Burrington, R.I.
Conger, Henry Lawton (J'43), Assoc. Engr., U.S. Engr. Dept., Punahou Campus, Dillingham Hall, Honolulu; *for mail,* 1550 K Young St., Honolulu 19, T.H.
Conhagen, Alfred ('22; '31; '35) (CY), Pres., Alfred Conhagen, Inc., 30 Church St., New York 7, N.Y.
Conklin, Robert M. (J'39) (CJM), Dept. 8451, West. Elec. Co., Inc., Haverhill; *for mail,* River Bend Farm, Lower River Rd., West Newbury, Mass.
Conkling, William C. (J'40) (BHK), Engrg. Devel. Dept., Wallace & Tiernan Co., Inc., Newark; *for mail,* 41 Johnson Ave., Star Route, Caldwell, N.J.
Conley, John W. (J'35) (CDJ), Engr., Westvaco Chlorine Products Corp., Newark; *for mail,* 566 Everett Ave., Palo Alto, Calif.
Conley, William Joseph ('45), Cons. Engr., Lincoln Elec. Co., 12818 Coit Rd., Cleveland, Ohio.
Conlon, William T. ('21; '25; '33), Dravo Corp., Pittsburgh, Pa; *for mail,* 10 N. Ridgewood Rd., S. Orange Ave., South Orange, N.J.
Conn, Thomas D. ('16; '19; '35) (BCNW), Deputy Harbor Engr., Bur. of Harbors, City of Baltimore, Recreation Pier, Baltimore 31; *for mail,* 3505 Lynchester Rd., Baltimore 15, Md.
Connell, Joseph T. (J'42) (DCMJ), Asst. Engr., Dravo Corp., Neville Island, Pittsburgh 25; *for mail,* 1080 Main St., Coraopolis, Pa.
Connelly, John Robert ('27; '37) (CNMJSD), Capt., Ord., Quarters 8, Springfield Armory, Springfield; *for mail,* 8 Armory Sq., Springfield 5, Mass.
Conner, John L. ('27), Asst. Gen. Supt., Charge Sta. Opera., Philadelphia Elec. Co., 900 Sansom St., Philadelphia; *for mail,* 241 Strathmore Rd., Brookline, Upper Darby P.O., Pa.
Conner, Kenneth B. (J'33) (BJOECL), Ch. Engr., Ace Mfg. Corp., K & Erie Ave., Philadelphia 24; *for mail,* 525 E. Leverington Ave., Philadelphia 28, Pa.
Conner, Norval White ('31; '42) (BHNOAZ), Prof. Fluid Mechanics, N.C. State College; *for mail,* 2719 Bedford Ave., Raleigh, N.C.
Conner, Ralph Norman (J'43) (CH), Sales Engr., Pelton Water Wheel Co., 2929—19th St., San Francisco 10, Calif.
Connolly, Walter Leland (J'44), Engr., Grove Regulator Co., 2532 Ridge Rd., Berkeley 4; *for mail,* Box 586, Route 3, Santa Cruz, Calif.
Connon, George W. ('07) (KLOSZ), Cons. Engr., Ave. 5 De Mayo 20, Mexico, D.F., Mex.
Connon, Jack Anthony (J'41) (BGL), 60 Berrian Rd., New Rochelle, N.Y.
Connor, Gerald Aaron (J'40) (ABSMFR), Design Engr., Douglas Aircraft Co., Inc., El Segundo, Calif.; *for mail,* 5466 Plover Ave., St. Louis, Mo.
Connor, Herschel W. (J'40) (AGM), Lt., Army Air Forces, Sheppard Field, Tex.
Connor, Nicholas John ('27; '35) (SKF), Sales Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 8 Barry Rd., Scarsdale, N.Y.
Connor, Willard Harvey (J'45) (CJMN), Staff Asst. Supt.'s Office, Gen. Elec. Co., 1635 Broadway, Ft. Wayne 2, Ind.
Connor, William Alexander (J'44), Serv. Engr., Page-Hersey Tubes, Ltd., 100 Church St., Toronto; *for mail,* 301 Kingswood Rd., Toronto 13, Ont., Can.
Connors, Robert Theodore ('46), Plant Engr., Jefferson Medical College & Hospital, 10th & Sansom Sts., Philadelphia 7; *for mail,* 5149 Cedar Ave., Philadelphia 43, Pa.
Conover, Frank H. ('26) (CMN), Mgr., Procurement Dept., Dorr Co., Inc., 570 Lexington Ave., New York 22, N.Y.
Conover, George W., Jr. (J'41), Box 55, Middlebush, N.J.
Conrad, Charles W. ('21) (OSCEFL), Pres., Conrad & Young, Inc., 33 Union St., East Walpole, Mass.
Conrad, Edward Kimpton (J'42), Capt., Army Air Forces, TSBPR4M6 (E); *for mail,* Box 418, Route 3, Dayton, Ohio.
Conrad, Joseph David (J'39) (BNSJ), Sec. Engr., Mech. Design Sec., Westinghouse Elec. Corp., Lester; *for mail,* 256 E. Leamy Ave., Springfield, Pa.
Conran, Fred M. ('14; '35), Designer, Builder, Spec. Mchy., 1275 Robert St., Hillside, Elizabeth, N.J.
Conran, Frederick M., Jr. (J'41) (BM), Draftsman, Conran Mch. Co., 107 Colden St., Newark; *for mail,* 1273 Robert St., Hillside, Elizabeth, N.J.
Consiglio, Jerome T. (J'41) (ASNMC), 2nd Lt., Trans. Corps, Army Serv. Forces, Asst. Ord. Officer, Trans. Corp. Sch., Army Air Base, New Orleans 12, La.
Consley, Julian Morton, Jr. ('43) (CSMFKB), Steam Serv. Supvr., Westinghouse Elec. Corp., 3451 E. Marginal Way, Seattle 4, Wash.
Constable, Thomas Alfred ('45) (CEHJLO), Supvr. Matl. Review, McDonnell Aircraft Corp., Lambert Field, St. Louis, Mo.
Constam, Alyn F. ('18; '35) (CDJ), Pres., Universal Industries Corp., 627 Union Commerce Bldg.; *for mail,* 10837 Hathaway Ave., Cleveland, Ohio.
Constance, John Dennis ('41; '44) (LKF), Gen. Engrg., Air Reduction Co., Inc., 60 E. 42nd St., New York 17, N.Y.; *for mail,* 506 Olympia Ave., Cliffside Park 7, N.J.
Constantinescu, V. ('24; '35) (CMNRZ), Propr., Mech. & Mar. Engr., Union Mchy. Repair Shop, 1238—3rd Ave. S., Lethbridge, Alta., Can.
Constantino, Constantine Stephen (J'39) (AKNFBE), Sr. Test Engr., Wright Aero. Corp., Paterson, N.J.; *for mail,* 305 E. 26th St., New York 10, N.Y.
Conta, Lewis D. ('36; '45) (BEJ), Asst. Prof. Engrg. Matls., College of Engrg., Cornell Univ., Ithaca, N.Y.
Contant, Peter Marinus ('32) (LODSZF), Tech. Rep., Dorr Co., 570 Lexington Ave., New York 22, N.Y.; *for mail,* Taj Mahal Hotel, Bombay, India.
Converse, Bernard T. ('09), Delaware Trust Bldg., Wilmington, Del.
Conway, Geo. R. G. ('13), Pres., Mex. Light & Power Co. Ltd., Apartado 124 Bis., Mexico, D.F., Mex.
Conway, John Joseph ('43) (HLWO), 53 Park Pl., New York 7, N.Y.
Conway, Martin John ('25) (PFJYO), Mgr., Petroleum Div., Lukens Steel Co., Coatesville; *for mail,* Hill Top Farm, Gap, Pa.
Coogan, Charles Henry, Jr. ('31; '41) (KSBE), Asst. Prof. Mech. Engrg., Mech. Engrg. Dept., Univ. of Conn., Storrs, Conn.
Cook, Benjamin F., Jr. (J'41) (ACL), Pewee Valley, Ky.
Cook, Charles ('43) (SKA), Mgr. Serv., Westinghouse Elec. Corp., Lester; *for mail,* 1138 W. Wilton St., Philadelphia 43, Pa.
Cook, Chas. B. (A'13), V.P., Charge Prod., Royal Typewriter Co., Inc., 150 New Park Ave., Hartford, Conn.
Cook, Dewey Delwin ('44) (AES), Pres., Cooktite Ring Sales Co., 1737 Howard St., Chicago 26, Ill.
Cook, Donald William (J'42), Cpl., Det. C, Bks. T-343, Proving Ground Det., Aberdeen Proving Ground, Md.
Cook, Earle Stanley ('25; '29) (AJYZ), Asst. to V.P., of Engrg., Westinghouse Air Brake Co., Wilmerding, Pa.
Cook, Edgar Charles (J'44), 3558 E. 105th St., Cleveland 5, Ohio.
Cook, Ernest Mellor (J'45) (AJT), 15 Albermarle Rd., Waltham 54, Mass.
Cook, Eugene Bertram (J'42) (NYALWB), Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 554 N. 29th St., Milwaukee 8, Wis.
Cook, Fay Vincent (M'45) (EBCF), M. M., Stanolind Pipe Line Co., Box 591, Tulsa 2, Okla.
Cook, Fred C. (J'34) (CJL), Outfitting Supvr., Ingalls Shipbldg. Corp.; *for mail,* 1112 Buena Vista St., Pascagoula, Miss.
Cook, Fred L. ('37) (BCH), Asst. Ch. Engr., John Robertson Co., Inc., 133 Water St., Brooklyn; *for mail,* 63 Buckingham Pl., Lynbrook, L.I., N.Y.
Cook, George, Jr. ('24; '26; '30) (SFEKD), Mech. Sqd. Ch., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2; *for mail,* 189 W. Champlost Ave., Philadelphia 20, Pa.
Cook, Geo. C. ('23) (SFK), Burns & Roe, Inc., 233 Broadway, New York, N.Y.; *for mail,* Millington, N.J.
Cook, Hardy M. ('23), Engr. of Tests, Brooklyn Edison Co., Inc., 380 Pearl St.; *for mail,* 8950 Colonial Rd., Brooklyn, N.Y.
Cook, Henry Edgerton, Jr. (J'45) (NBJCMS), Electronics Technician's Mate, 2nd Class, U.S.N., "C" Div., U.S.S. *Cumberland Sound,* AV-17, F.P.O., San Francisco, Calif.; *for mail,* 1182 Bushwick Ave., Brooklyn 21, N.Y.
Cook, Herbert E. ('29), Gen. Supt., Engrg. Div., B. F. Goodrich Co., Akron; *for mail,* 335 Broad Blvd., Cuyahoga Falls, Ohio.
Cook, Howard Lee ('32; '41) (HB), Prin. Drainage Engr., U.S. Dept. of Agric., Washington, D.C.; *for mail,* 5905 Wilson Lane, Bethesda 14, Md.
Cook, John Schippers (J'43) (MCDE), Lt. (j.g.), U.S.N., Naval Air Base 943, F.P.O., San Francisco, Calif.; *for mail,* 1223 Cobb Ave., Kalamazoo 54, Mich.
Cook, John W. (39) (CHF), Supt., Society for Establishing Useful Manufactures, 72 McBride Ave., Paterson, N.J.
Cook, Kenneth Franklin (J'44) (CDS), Methods Engr., Stockyards, Wilson & Co.; *for mail,* 541 N.W. 30th St., Oklahoma City, Okla.
Cook, Laurie ('23; '30) (SE), 1405 Beechview Ave., Pittsburgh, Pa.
Cook, Marsden Alfred (J'30) (BMLW), Asst. Prof. Mech. Engrg., Rensselaer Poly. Inst., Troy, N.Y.
Cook, Raymond W. ('27), V.P., Charge Works, Wallace Barnes Co., 18 Main St., Bristol, Conn.
Cook, Richard Louis (J'42) (MLN), 383 Linden Blvd., Brooklyn, N.Y.
Cook, Thomas J., Jr. (J'39) (CDM), Apt. 202, 174 Locust St., Akron, Ohio.
Cook, Thomas Johnson ('24) (BCD), Pres., Gen. Mgr., McKinnon Industries, Ltd., Subsidiary of Gen. Motors Corp., St. Catharines, Ont., Can.
Cook, Thos. Russell ('11) (CDJRST), Sr. Engr., Coverdale & Colpitts, 120 Wall St., New York 5, N.Y.
Cook, William Dodd ('21; '28; '35) (ACFW), Sr. Adviser to Suburban Propane Gas Corp., Phillips Petroleum Co., Rm. 8, 19 N. Harrison St., East Orange, N.J.
Cook, William Pierson, Jr. ('11; '13; '29) (W), Exec. V.P., Eppinger & Russell Co., 80—8th Ave., New York 11, N.Y.

Cook, William Pierson, III (J'43) (CYD), Lt. (j.g.), U.S.N.R., Radar Officer, U.S.S. *Vesole*, DD-878, F.P.O., San Francisco, Calif.; *for mail*, 35 Whitehall Blvd., Garden City, N.Y.
Cook, Willis Dyckman ('21) (E), Supt. Equip., Shop 3, Div. of Highways, State of Calif., 703 B St.; *for mail*, 928—8th St., Marysville, Calif.
Cooke, Bennett W. (A'24), Pres., Motor Inst. of Am., 333 Park Ave., Glencoe, Ill.
Cooke, Gerhart ('43) (CZSMFK), Mfg. Rep., 960 Mission St., South Pasadena, Calif.
Cooke, James Welch ('46), Field Engr., Gen. Steel Castings Corp., Eddystone; *for mail*, Spring-Mill & Gulph Rds., Villanova, Pa.
Cooke, John C. (J'43) (CMS), Indus. Engr., Mgmt. Consultant, E. I. du Pont de Nemours & Co., Market St., Wilmington, Del.; *for mail*, R.F.D. 1, Elkton, Md.
Cooke, Morris Llewellyn ('03; F'36) (CG), *Manager*, '14-'15; Cons. Engr., St. Georges Rd., Mt. Airy P.O., Philadelphia; also New Hope, Pa.
Cooke, Thomas C. ('44; '45) (KBESOF), Engr., Tomlinson Co., Inc., 1312-24 W. Marshall St., Richmond, Va.; *for mail*, 2209 Wilson St., Durham, N.C.
Cooke, W. G. (J'37), Capt., Royal Canadian Army; Natl. Gypsum (Can.), Ltd.; *for mail*, Dingwall, Victoria Co., N.S., Can.
Cookingham, Sterling H. (J'36) (CMJD), Asst. Foreman, Tractor Works, Internatl. Harvester Co., 2600 W. 31st Blvd., Chicago; *for mail*, 3841 Woodside Ave., Hollywood, Ill.
Cookson, Harold Wesley, Jr. (J'44) (MJNDC), Mgr., Cookson Co., 964 Harrison St., San Francisco 7, Calif.
Cooledge, Marshall M. (A'44) (RAGJCY), East. Sales Rep., Buckeye Steel Castings Co., 50 Church St., New York 7, N.Y.
Cooley, Martin Luther, Jr. (J'42) (ES), Box 2002, Ancon, C.Z.
Cooley, Sherman Junior (J'45), Heron Lake, Minn.
Cooley, William Crockett (J'44) (AKBFJN), Ensign, U.S.N.R., Rocket Engr., Propulsion Lab., Pilotless Aircraft Unit, Naval Air Sta., Mojave, Calif.
Coolidge, Donald James (J'44), House 4, N. Campus, Northwestern Univ., Evanston, Ill.
Coolidge, Richard N. ('27), Gen. Mgr., Cumberland River Sand Co., 10 Fatherland St.; *for mail*, Belle Meade Blvd., Nashville, Tenn.
Coon, Thurlow Emmett ('08; '14) (FSKZ), Pres., Coon-DeVisser Co., Inc., 2051 W. Lafayette Blvd., Detroit 16, Mich.
Cooney, Robert Lawrence (J'40) (DJN), Engr., Chase Fdy. & Mfg. Co., 2300 Parsons Ave., Columbus 7; *for mail*, 1228 Wilson Ave., Columbus 6, Ohio.
Coonley, Howard (A'17), Pres., Walworth Co., 60 E. 42nd St., New York, N.Y.
Coonradt, Arthur Chapin ('25; '35) (KSFE), Prof. Mech. Engrg., Chmn., Mech. Engrg. Dept., N.Y. Univ., 180th St. & University Ave.; *for mail*, 114 W. 183rd St., New York 53, N.Y.
Coop, Albert Bullock ('37) (EKFS), Clerk, Ch. Engr., Mass. Htg. Corp., 56 Rogers St., Cambridge 42, Mass.
Cooper, Albert Hudiburgh ('37; '40) (LKOYJZ), Head, Dept. Chem. Engrg., Univ. of Denver, Denver 10, Colo.
Cooper, Calvin ('42), 2938 Fairfields Ave., Baton Rouge, La.
Cooper, Charles Dolman ('45), Prof. Engrg. Drawing, Ohio State Univ., Columbus 10; *for mail*, 393 Mimring Rd., Columbus 2, Ohio.
Cooper, Delbert John (J'45) (BHKCNZ), Engr., Phillips Petroleum Co.; *for mail*, 115 S. Chickasaw, Bartlesville, Okla.
Cooper, Earl E. ('25) (CMN), Ch. Engr., Chamber-Bering-Quinlan Co., 700 N. Jasper St., Decatur 60; *for mail*, W. Macon St., Decatur 32, Ill.
Cooper, Eli George ('35; '42) (FEJKNC), Ch. Engr., Gen. Mgr., May Engrg. Co., 12 Adam St.; *for mail*, 46 Prospect St., Pittsfield, Mass.
Cooper, Francis Howland (J'37), Lt. Col., U.S.M.C., 2nd Armd. Amphibian Bn., F.P.O., San Francisco, Calif.
Cooper, Frank Scott, Jr. ('44), Asst. Mgr., Foreign Operas. Dept., Socony-Vacuum Oil Co., 26 Broadway, New York 4, N.Y.; *for mail*, 172 Fairview Ave., Stamford, Conn.
Cooper, Fred'k F. ('14), Pres., Olympian Dredging Co., Rm. 517, 525 Market St.; *for mail*, 316 Laurel St., San Francisco, Calif.
Cooper, George H. ('16; '35) (AMSBZ), Engr., Installation Lab., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 Main St., East Hartford 8; *for mail*, 209 Fairfield Ave., Hartford 6, Conn.
Cooper, Howard ('17; '45) (EAR), Ch. Lub. Engr., Sinclair Refining Co., 630—5th Ave., New York 20, N.Y.
Cooper, Howell C. ('12) (EF), Retired; 114 Centennial Ave., Sewickley, Pa.
Cooper, Kenneth K. (J'39) (NMBKCE), Design Engr., Design Engrg. Div., Air Conditioning Dept., Gen Elec. Co., 1605 Winter St., Ft. Wayne 2; *for mail*, 1524 Park Ave., Ft. Wayne 6, Ind.
Cooper, Lemuel Browning ('41; '46) (JNS), Asst. Prof. Mech. Engrg., Univ. of Wash., Seattle 5, Wash.
Cooper, Leo James ('43) (CDM), Sales Engr., Otis Elevator Co., 260—11th Ave., New York, N.Y.; *for mail*, Rensselaer Rd., Essex Fells, N.J.
Cooper, R. Dunham ('24; '35) (CFS), Gen. Mgr., Produce Terminal Corp., 305 Exchange Bldg., Union Stock Yards, Chicago, Ill.
Cooper, Roland Stevens ('24; '35) (EFKS), West. Dist. Mgr., West. Dist. Engr., Air Preheater Corp., 122 S. Michigan Ave., Chicago 3, Ill.
Cooper, Samuel J. ('26; '35) (ACF), c/o Canadian Pacific Airlines, Lac-a-la-Tortue, Que., Can.
Cooper, Warren Boughton (J'43), Seaman, 1st Class, U.S.N., Co. 113-206, Navy Pier, Chicago, Ill.
Cooper, Wilbur Stanley ('24; '30; '35) (SREZOK), Chief, Steam Power & R.R. Equip. Sec. (Subsidiary of Reconstruction Finance Corp.), 425—2nd St., N.W., Washington 25; *for mail*, 3930 Langley Court, N.W., Washington 16, D.C.
Cooper, Wilfrid Broxup (J'42) (LTC), Dir. of Research, Tubize Rayon Corp.; *for mail*, 113 Chatillon Rd., Rome, Ga.
Coors, Robert M. (J'42) (ABC), Lt., Engrg. Div. Aircraft Lab., Matl. Center, Wright Field, Dayton, Ohio.
Cope, Andrew P. (J'43) (ZGAN), Asst. Project Engr., Research, Sperry Gyroscope Co., Clinton Rd. & Stewart Ave., Garden City; *for mail*, Apt. 3-E, 299 Jackson St., Hempstead, N.Y.
Cope, Edge Taylor ('25; '31) (FKS), Sr. Research Engr., Detroit Edison Co., 2000—2nd Ave., Detroit; *for mail*, 12850 Lauder, Detroit 27, Mich.
Cope, William J. ('27; '32) (KSEFNB), Prof. & Head, Dept. of Mech. Engrg., Univ. of Utah, Salt Lake City 1, Utah.
Copeland, Harold Curtis (J'43) (ERFSA), 129 N. Sherwood St., Ft. Collins, Colo.
Copithorne, A. Russell (J'44), 54 Garfield Ave., Chelsea, Mass.
Coplen, Herman Leroy, Jr. (J'37) (ZCHA), Devel. Engr., Research Dept., Aerojet Engrg. Corp., Box 296, Azusa; *for mail*, 1579 Coolidge Ave., Pasadena 7, Calif.
Copley, Alden Gibson (J'45) (CO), Ensign, U.S.N.R., 52 Brookside Pl., New Rochelle, N.Y.
Copony, Alfred ('39), 300 Conestaga Rd., Wayne, Pa.
Copp, Earle M. ('28) (CL), Exec. Sugar Engr., Graver Tank & Mfg. Co., 424 Madison Ave., New York 17, N.Y.
Copp, Lowell J. (J'37) (MC), Indus. Engr., Cincinnati Milling Mch. Co., Marburg Ave., Cincinnati 9, Ohio.
Copp, Ralph ('26), Supvg. Engr., Pevely Dairy Co., 1001 S. Grand Blvd., St. Louis, Mo.; Cons. Engr., U.S. Dept. of Agric., Rm. 2617 S. Agric. Bldg., Washington, D.C.; *for mail*, 4440 Lindell Blvd., St. Louis, Mo.
Coppersmith, Charles William ('26) (GHM), Assoc. Prof. Engrg. Drawing, Case Sch. of Applied Sci., Univ. Circle, Cleveland 6, Ohio.
Coppersmith, Frederick M. ('33; '43) (MJC), Plant Supt., Natl. Lead Co., 85 Jay St., Brooklyn 1; *for mail*, 3950 Bronx Blvd., Bronx 66, N.Y.
Corapi, Vincent (J'45) (AJP), 69 Forest St., Watertown 72, Mass.
Corbett, Charles Hawley (J'45) (CGM), V.P., Am. Tag Co., 151 Cortlandt St., Belleville, N.J.
Corbett, Clifford William ('45) (MCO), Chief, Engrg. Dept. 9784, West. Elec. Co., Inc., 300 Central Ave., Kearny, N.J.; *for mail*, 333 E. 53rd St., New York 22, N.Y.
Corbett, Lawrence Bowes ('43) (GJYZ), Devel. Engr., Internatl. Business Mchs. Corp., Endicott; *for mail*, 55 Walnut St., Binghamton, N.Y.
Corbin, Edwin Marion (J'39) (BNKFS), Design Engr., Foster Wheeler Corp., 165 Broadway, New York; *for mail*, 281 Wadsworth Ave., New York 33, N.Y.
Corbin, Spotswood Wellford ('45) (CDEHNO), Mgr., Indus. Resale Div., Gen. Elec. Co., 1 River Rd., Schenectady 1; *for mail*, R.D. 1, Ballston Lane, Charlton, N.Y.
Corby, J. B. (A'21), Pres., Treas., Corby Supply Co., 3942-44-46 W. Pine Blvd., St. Louis, Mo.
Corby, John Clarkson, Jr. (J'42) (NZY), Clerk, Inf., Co. A, 15th Bn., 4th Regt., AGFRD-1, Camp Pickett, Va.; *for mail*, 75 W. Catawissa St., Nesquehoning, Pa.
Corcoran, James Lucien ('42) (SCMJ), Ch. Engr., Consltd. Safety Valve Div., Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport; *for mail*, 290 Booth Hill Rd., Bridgeport 18, Conn.
Cordell, Philip M. ('31; '35) (SEHMCW), Indus. Sales Engr., Tex. Elec. Serv. Co., Ft. Worth 1, Tex.
Cordes, Fred Kamehameha (J'40) (CLNG), Sec. Supt., Calif. Packing Corp., Wahiawa, Oahu, T.H.
Corey, Donald Holbrook ('30; '34; '35) (JSM), Welding Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Corey, Fred B. ('94; '00), Retired; "Stony Knoll," R.D. 1, Barberton, Ohio.
Corey, Roger L. (J'40) (AES), Ensign, U.S.N.R., Turnpike St., Canton, Mass.
Corl, Harry E. ('22; '35) (SEFOB), Cons. Engr., 1505 Race St., Philadelphia 2; *for mail*, 130 Overhill Rd., Upper Darby, Pa.
Corl, James Gray (J'40) (SETZFH), 431 Oakwood Dr., Fullerton, Pa.
Cornelius, Charles Taylor ('26; '37) (BCJ), Works Mgr., Aluminum Co. of Can., Ltd.; *for mail*, 58 Maple Ave., Shawinigan Falls, Que., Can.
Cornelius, L. A. ('31), Pres., Wolverine Brass Works, Grand Rapids, Mich.
Cornell, Dana Robert ('18; '21; '35) (CBRMF), Works Mgr., Stand. Forgings Corp., 3444 Dickey Rd., East Chicago, Ind.
Cornell, Donald H. (J'39) (YNZB), Physical Research Lab., B. F. Goodrich Co., Akron 18, Ohio.
Cornell, Edward Shelton, Jr. ('32) (BCJKNY), Pres., Shelton Co., 303 Main St.; *for mail*, 56 Valley Rd., Stamford, Conn.
Cornell, Harry Leroy (J'42) (A), Lt., Army Air Forces, Pilot, 366th Bomb Sqd., 305th Bomb Gp., A.P.O. 634, c/o Postmaster, New York, N.Y.; *for mail*, Peterson Plan, McKeesport, Pa.
Cornell, Robert L. ('27) (CDLK), Pres., Fla. Precooling Co., Box 110, Sanford, Fla.
Cornell, Robert L., Jr. (J'39) (BHK), Lt. Ord. Sec., A.P.O. 846, c/o Postmaster, New York, N.Y.
Cornell, Sidney ('44; '45) (ABC), Research Assoc., Woods Hole Oceanographic Inst.; *for mail*, Box 233, Woods Hole, Mass.
Cornell, Walter Rodney ('25) (BH), Prof. Mechanics of Engrg., Cornell Univ.; *for mail*, 507 Hanshaw Rd., Ithaca, N.Y.
Cornell, Wellington C. (J'41) (ABJ), 1st Lt., 11th Co., 1st STR, TIS, Ft. Benning, Ga.
Cornell, William Bouck ('22) (CDMLOY), Prof. Mgmt. & Chmn., Dept. of Mgmt. & Indus. Relations, N.Y. Univ., Washington Sq., New York 3, N.Y.; *for mail*, 197 Grove St., Montclair, N.J.
Cornell, William George (J'41) (BHKEA), Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 799 Albany St., Schenectady 7, N.Y.
Cornish, Donald Frederick (J'34) (J), Mech. Engr., Sales, Anaconda Am. Brass, Ltd., New Toronto, Ont., Can.
Cornman, Guy Leslie, Jr. (J'45) (CJL), 1010 Knox St., Durham, N.C.
Cornog, Robert (J'40) (KSABLM), 2007 Carnegie Lane, Redondo Beach, Calif.
Cornwell, David R. L. (J'32) (NGKE), Asst. Ch. Engr., C. F. Pease Co., 2601 W. Irving Park Rd., Chicago 18, Ill.
Cornwell, Edward Stevens (J'44) (ABS), Engr., Draftsman B, Engrg. Div., Douglas Aircraft Co., 5651 Manchester Ave., El Segundo; *for mail*, 143 N. Alexandria Ave., Los Angeles 4, Calif.
Cornwell, Hobart V. ('14; '24) (FKS), Cons. Engr., 11 Park Pl., New York; *for mail*, 117 Marvin Ave., Hempstead, L.I., N.Y.
Cornwell, William A. (J'39) (HA), Asst. Hyd. Engr., U.S. Bur. of Reclamation; *for mail*, Box 992, Visalia, Calif.
Corr, John Edward (J'43) (BNKS), Test Engr., Gen. Elec. Co., River Works, Lynn; *for mail*, 505 Pleasant St., Malden 48, Mass.
Correll, Harold Evans (J'43) (NZABEM), 2nd Army Air Force, Photographic Intelligence Det., Advance Det. Hdq., Far East Air Force (A-2), A.P.O. 925, c/o Postmaster, San Francisco, Calif.; *for mail*, Box 154, 104 Hickok St., Christiansburg, Va.
Corrigan, Brian (J'40) (EADCLJ), Mech. Engr., Tech. & Research Div., Texas Co., 135 E. 42nd St., New York 17; *for mail*, 472 Gramatan Ave., Mt. Vernon, N.Y.
Corrough, Hugh M. ('40) (CEK), Dir., Alco Products Div., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Corry, Robert Thurton (J'44) (NMBH), Engrg. Asst., Carbide & Carbon Chemicals Corp., Oak Ridge, Tenn.; *for mail*, 58 Main St., Box 907, New Canaan, Conn.
Cortes, Jos. M. ('23; '31; '35) (ABW), Gen. Mgr., Cia. Colombiana de Sacos Bates, P.O. Box 1007, Bogotá, Colombia, S.A.
Corwin, Bernard (J'35) (NMCJ), Engr., Hilton Tool & Mch. Co., 1548 Southern Blvd., Bronx 60, N.Y.; *for mail*, 241 S. Arlington Ave., East Orange, N.J.
Corwin, Lloyd A. ('39), S. 38th & W. Grant St., Milwaukee, Wis.
Cory, David Cleveland (J'34) (CDGL), Asst. to Pres., Peter Cailler Kohler Swiss Chocolates Co., Inc., 60 Hudson St., New York 13, N.Y.
Cosentini, William Randolph (J'42) (SKEFHB), Design Engr., Jaros, Baum & Bolles, 415 Lexington Ave., New York; *for mail*, 144-03—71st Ave., Flushing, N.Y.
Cosper, William Roy, Jr. (J'40) (FKS), 2832 Edgehill Rd., Cleveland Heights 18, Ohio.
Cosser, Charles T. (J'40) (AEK), Engrg. Div., Dist. Office, Gen. Elec. Co., Los Angeles; *for mail*, Apt. 213 Dracker Apts., 10824 Lindbrook Dr., West Los Angeles 24, Calif.
Costello, Richard John (J'45) (ESHKNF), 307 Sedgwick Dr., Syracuse 3, N. Y.
Costigan, James Thomas ('45) (LEC), Dist. Mgr., Sharples Corp., 501—5th Ave., New York 17, N.Y.
Coston, Charles Leslie (J'39) (BSZ), Assoc. Ord. Engr. (P-3), Design & Drafting Div., Sec. 8 (Pamphlets), Naval Shipyard, 11th & N St., S.E., Washington 25; *for mail*, 2814 Q St., S.E., Washington 20, D.C.
Cotter, Chester ('37) (OSEF), Engr., Black & Veatch, Cons. Engrs., 4706 Broadway, Kansas City 2; *for mail*, 2019 Erie St., North Kansas City 16, Mo.
Cotter, George Louis ('24; '45) (RCHA), Asst. West. Mgr., Westinghouse Air Brake Co., 80 E. Jackson Blvd., Chicago 4, Ill.
Cotterell, Wm. J. ('24) (FKS), Managing Dir., Internatl. Combustion, Ltd., P.O. Box 5981, Johannesburg, S. Africa.
Cotterman, Frank David (J'36) (BNZJ), Research Engr., Crane Co., 836 S. Michigan Ave., Chicago 5, Ill.

Cottle, Arthur P. ('35) (AEJ), Retired; 632 Home Ave., Oak Park, Ill.
Cottle, Harry Norman, Jr. (J'41) (ABZJ), Project Engr., Exper. Sec., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 31 E. Highland Ave., Orange, N.J.
Cottle, Richard Arnold (J'40) (DEL), Sales Engr., Independent Rubber Co., 665 W. Washington Blvd.; *for mail,* 632 Home Ave., Oak Park, Ill.
Cotton, Edwin Ray ('38; '40) (S), Sales Engr., Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Cotton, Howard W. ('42) (BMN), H. W. Cotton Co., 535—5th Ave., New York 17, N.Y.
Cottrell, Frederick G. (Non-Member), *Holley Medallist,* '37; Consultant, Research Corp., 405 Lexington Ave., New York 17, N.Y.; *for mail,* 1557 Waverly St., Palo Alto, Calif.
Cottrell, Nicholas (J'40), 6911 Beverly Rd., Bethesda, Md.
Cottrell, Robert Boyd, Jr. (J'41) (RMJCAD), Capt., Ord. Dept., Instr., Officer Candidate Sch., Aberdeen Proving Ground, Md.
Cottrell, Robert Boyd, Sr. ('41) (RCJMB), Ch. Mech. Engr., Am. Steel Fdys., 410 N. Michigan St., Chicago 11, Ill.
Couch, Charles William (J'38) (KB), Assoc. Mech. Engr., Puget Sound Naval Shipyard; *for mail,* 1136 Evans Ave., Bremerton, Wash.
Couch, David H. ('19) (SFK), Mech. Engr., Ebasco Services, Inc., 2 Rector St., New York 6; *for mail,* R.F.D. 3, Douglas Rd., Beachcroft, Huntington, L.I., N.Y.
Couch, Harold Kennan (J'41), Serv. Engr., Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3, Ill.
Coulson, Armand Leroy (J'43) (CMZAHS), Lt. (j.g.), U.S.N.; *for mail,* 712 Lorentz Ave., Morgantown, W. Va.
Coulson, Harry Glen ('28) (NCDEAZ), Design Engr., Noble Co., 1860—7th St., Oakland 7; *for mail,* 961 Tulare Ave., Berkeley 6, Calif.
Coulter, Earl Eugene (J'43) (ABE), 17735 Parkmount Ave., Cleveland 11, Ohio.
Coulter, Warren Raymond ('44), Ch. Engr., Coulter Copper & Brass Co., Ltd., 115 Sumach St.; *for mail,* 21 Donlea Ave., Toronto, Ont., Can.
Coupal, Eugene A. ('46) (CMO), 30 Norwich Rd., Needham, Mass.
Coupland, Bertram Russell (J'41), Engr., Charge Design & Tool & Equipment, Super-Cold Corp., 1020 E. 59th St.; *for mail,* 9611 S. LaSalle St., Los Angeles, Calif.
Courtenay, Mortimer Hasell ('36) (TC), Field Engr., SKF Industries, Inc., 591 Peachtree St., N.E.; *for mail,* 2590 Forrest Way, N.E., Atlanta, Ga.
Courtney, A. W. Jr. (J'39) (ABS), Capt., Hdq., 5th Army Air Force, A.P.O. 925, c/o Postmaster, San Francisco, Calif.
Courtney, Albert Lee (J'43) (MBLZDO), Prod. Engr., Studebaker Corp., Sample St.; *for mail,* 519 Gladstone Ave., South Bend 19, Ind.
Coutinho, John DeSiqueira (J'42) (ABM), Graduate Aero. Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* 10-A Bennett Gardens, Hempstead, N.Y.
Couture, Joseph Walter ('44), Ch. Engr., Diltj Mch. Works, North 1st; *for mail,* R.F.D. 1, Fulton, N.Y.
Couzins, Newton W. (J'39) (CJM), Ch. Engr., Berkeley Steel Constr. Co., Inc., 2nd & Camelia Sts.; *for mail,* 249 Yale Ave., Berkeley, Calif.
Covey, Kenneth S. ('43), 509 Anderson St., Pittsburgh, Pa.
Covino, Adolph O. ('40), 2431 Dean St., Brooklyn, N.Y.
Cowan, Alfred Vanterhoit (J'41) (LYKZEF), Chem. Engr., Research, Koppers Co., Inc., Koppers Bldg., Pittsburgh 19; *for mail,* 2 Olympia Pl., Pittsburgh 17, Pa.
Cowan, Ben (J'32) (LWHNB), Mech. Engr., Price Bros. & Co., Ltd., Kenogami; *for mail,* 33-204 Notre Dame St. W., Montreal, Que., Can.
Cowan, Edw. L. (J'38) (CFS), Asst. Ch. Engr., Gaylord Container Corp., Bogalusa, La.
Cowan, Frank ('21) (ACDHKM), Ch. Engr., Arnold Bros. Inc., Perry, Iowa.
Cowan, James H. (J'40) (EOCLSF), Mgr., Utilities, City of Spur, Spur, Tex.
Cowan, Percy J. ('37; F'38), Upcott, Cranes Park, Surbiton, Surrey, England.
Cowell, Allan Tyler (J'43) (FAKJE), Asst. Automotive Engr., Atlantic Refining Co., 3144 Passyunk Ave.; *for mail,* 5855 N. 6th St., Philadelphia 20, Pa.
Cowell, Warner T. ('25) (NMJABL), Gear Engrg. Dept., Gleason Works, 1000 University Ave., Rochester; *for mail,* 32 Hartsdale Rd., Rochester 9, N.Y.
Cowgill, William Warwick ('33) (YTACMO), Project Mgr., U.S. Rubber Co., 1230—6th Ave., New York 20, N.Y.; *for mail,* Greenfield Hill, Fairfield, Conn.
Cowie, Alexander ('37) (BJNAH), Research Engr., Foote Bros. Gear & Mch. Corp., 5225 S. Western Ave., Chicago 9; *for mail,* 3017 E. 78th St., Chicago 49, Ill.
Cowles, Clifford Atwood ('17; '25) (SZLEFK), Owner, Cowles & Co., 1112 Texas Bank Bldg., Dallas 2, Tex.
Cowles, William Eddy (J'42) (ACD), Test Engr., Gen. Elec. Co.; *for mail,* 1038 Dean St., Schenectady, N.Y.
Cox, Abraham Beekman ('14) (SBA), Maj., Ord., Field Serv., Retired; Glensfoot Farm, Cherry Valley, N.Y.
Cox, Clarence Edward ('29) (BEK), Ch. Engr., Engrg. Dept., Chicago Pneumatic Tool Co., Howard St.; for mail, 1324 Chestnut St., Franklin, Pa.
Cox, Edward Lorenz ('23; '45) (BEHKYW), Specification & Design Engr., Navy Dept., 17th St. & Constitution Ave., Washington; *for mail,* 1620 Ft. Davis St., S.E., Washington 20, D.C.
Cox, Felix Otterbine, Jr. (J'42) (FZM), Instrument Mechanic, Bethlehem Steel Co., Sparrows Point; *for mail,* 2903 Dunmore Rd., Baltimore 22, Md.
Cox, Frank G. ('05; A'08), 2415 W. 17th St., Wilmington 73, Del.
Cox, Harold Newton, Jr. (J'37), Tech. Sgt., Ord., U.S.A., Hdq. & Supply Co., 140th Ord. Bn., A.P.O. 505, c/o Postmaster, New York; *for mail,* Southold, L.I., N.Y.
Cox, James Cleo ('38), Foreman, Welding-Copper Shop, Tenn. Eastman Corp.; *for mail,* Route 3, Cliffside, Kingsport, Tenn.
Cox, James S. (J'39) (BCJ), 728 Delaware Ave., Bethlehem, Pa.
Cox, James William ('16; '21) (CDT), Textile Engr., 320 Broadway, New York 7, N.Y.
Cox, John Lyman ('27) (JBHKL), Ch. Engr., Midvale Co., 4320 Wissahickon Ave., Philadelphia 40, Pa.
Cox, John Thomas, Jr. (J'43), 3632 Meadowlake, Houston, Tex.
Cox, Len (J'44) (MJNB), Test Engr., MacDonald & MacDonald, 718 Granville St.; *for mail,* 3436 W. 36th Ave., Vancouver, B.C., Can.
Cox, Osborne S. (J'38) (ZMJNY), Gage Designer, A.C. Spark Plug Div., Gen. Motors Corp., Flint 2; *for mail,* 635 N. Ball St., Owosso, Mich.
Cox, S. F. ('29), Research Librarian, Research Lab., Pittsburgh Plate Glass Co., Creighton, Pa.
Cox, William Paul (J'21) (BRJNYM), Mech. Engr., Ry. Div., Timken Roller Bearing Co., 1835 Dueber Ave., S.W., Canton 6, Ohio.
Coxon, Wilfred R. ('45) (SFK), Steam Engr., Solvay Process Co., Hopewell, Va.
Coykendall, William Ellis, Jr. (J'42), 56 Prospect Ave., Larchmont, N.Y.
Coyle, Daniel K. ('40) (ABC), 878 Acacia St., Sunnyvale, Calif.
Coyle, William G., Jr., (J'39), 331—4th St., Atlanta, Ga.
Cozzo, Sam Edmund ('36) (BLKD), Ch. Draftsman, Charge Designing, Austin Co., Engrs & Bldrs., Pittsburg; *for mail,* 363 Santa Clara Ave., Oakland 10, Calif.
Crabb, Gordon Henry ('43) (BHNMYC), Mech. Engr., Panama Canal, Balboa Heights; *for mail,* Box 564, Gamboa, C.Z.
Crabill, John Willard (J'42) (AJMFSC), Lead Preliminary Tool Engr., Boeing Aircraft Co., 7755 E. Marginal Way; *for mail,* 13342—34th Ave. S., Seattle 88, Wash.
Craft, Harry William (J'44) (CL), Mech. Foreman, Procter & Gamble Mfg. Co., 169 E. Grand, St. Louis 7; *for mail,* 5345 N. Kingshighway, St. Louis 15, Mo.
Crafton, Paul A. (J'44) (NMBCYZ), Mech. Engr., Research & Devel., Naval Research Lab.; *for mail,* 3940—2nd St., S.W., Washington 20, D.C.
Crafts, Curtis S. ('34) (GBCMY), Secy., Ch. Engr., Goss Ptg. Press Co., 1535 S. Paulina St., Chicago 8, Ill.
Crafts, Irving M. ('34), Pres., Pickering Governor Co.; *for mail,* 485 Main St., Portland, Conn.
Craig, Burnie McDonald ('38; '44) (ONBZ), Cons. Mech. Engr., 934 El Campo Dr., Pasadena 8, Calif.
Craig, Gordon Davis (J'45) (SMCKOE), Instr., Mech. Engrg., Univ. of Santa Clara, Santa Clara; *for mail,* 845 California St., San Francisco, Calif.
Craig, Howard B. (J'99), Retired; 161 Orchard Rd., Solvay 9, N.Y.
Craig, Jas. ('97), Retired; 3900 Post Rd., Apponaug, R.I.
Craig, John McKinley (J'44) (CKABDJ), Area Engr., E. I. du Pont de Nemours & Co.; *for mail,* Apt. 3, 1712 West St., Wilmington, Del.
Craig, John S. (J'36) (SKD), Sales Engr., Babcock & Wilcox Co., 2730 Koppers Bldg., Pittsburgh 19, Pa.
Craig, Ollison ('21) (FSKCJN), Engrg. Mgr., Riley Stoker Corp., 9 Neponset St.; *for mail,* 245 Burncoat St., Worcester, Mass.
Craig, Robert ('18; '24) (ABCES), Pat. Atty., Cooper, Kerr & Dunham, 233 Broadway, New York 7, N.Y.
Craig, William Hardy (J'43) (YLKSC), Constr. Supvr., Dow Chem. of Can. Ltd., Sarnia, Ont., Can.
Crain, Henry Lawson ('28; '39) (SFE), Plant Results Engr., Kansas City Power & Light Co., 115 Grand St., Kansas City 6, Mo.
Crain, John J. ('96; '08), E. Main St., Wallingford, Conn.
Crain, Richard Willson, Sr. (J'38) (SKDNLJ), Instr., Mech. Engrg. Dept., Univ. of Wash., Seattle 5, Wash.
Crais, Charles I. (J'38), Standard Elec. Time Co., Springfield 2, Mass.
Cramer, Jerome Paul (J'36) (DCNMLJ), Supt., Mch. Shop, Bay City Iron Works, 475—4th St., Oakland 7; *for mail,* 761 Spruce St., Berkeley 7, Calif.
Cramer, Luther William (J'32) (DB), Real Estate Agent, 231 N.E. 7th Ave., Camas, Wash.
Cramer, Robert, Jr. (J'31) (EBSN), Asst. Ch. Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7; *for mail,* Box 184, Hales Corners, Wis.
Crane, Edw. J. ('25), West. Elec. Co., Inc., Chicago; *for mail,* 623 N. Kenilworth Ave., Oak Park, Ill.
Crane, Eugene C. ('15; '22) (FJKS), Checker, Lee Wilson Engrg. Co., 20005 West Lake Rd., Rocky River; *for mail,* 2203 Eldred Ave., Lakewood 7, Ohio.
Crane, Harry O. (J'37), Lt., Picatinny Arsenal, Dover, N.J.
Crane, Henry Middlebrook ('20) (AE), Tech. Asst. to Chmn., Gen. Motors Corp., 1775 Broadway, New York; *for mail,* 40 E. 54th St., New York 22, N.Y.
Crane, Howard P. ('28; '32) (BMR), 593 Prospect St., West Boylston, Mass.
Crane, Robert Douglas (J'38) (CDJMOY), Sales Promotion, Wilmington Shops, E. I. du Pont de Nemours & Co., Maryland Ave., Wilmington, Del.
Crankshaw, Edwin ('42) (CENRAM), Asst. Ch. Engr., Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland, Ohio.
Cranley, Thomas Edward ('42) (BDNS), Secy., Engr., Wirthlin Mann Co., 66 Plum St., Cincinnati 2; *for mail,* 2647 Atlantic Ave., Cincinnati 9, Ohio.
Crapo, Philip Warren (J'41) (EHMDCJ), 84 Quinlan Ave., Port Richmond, S. I. 2, N. Y.
Crapon, Harry Dawley, Jr. (J'44) (CJS), Asst. to Div. Ch. Engr., Spec. Devices Div., Austin Co., 19 Rector St., New York 6, N.Y.; *for mail,* 515 Calleton Rd., Westfield, N.J.
Crapple, John W. (J'45) (CDJAEP), 6352 S. Lorel Ave., Chicago, Ill.
Crary, Selden B. ('43) (BSEHRK), Engr., Analytical Div., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Crater, Myron Lee ('33; '45) (SCFBZR), Ch. Steam Plant Engr., Pub. Serv. Dept., City of Glendale, 120 N. Howard St.; *for mail,* 631 N. Isabel St., Glendale 6, Calif.
Crater, Wilbur David (J'42) (BAFN), Application Engr. on Aircraft Gas Turbines, Gen. Elec. Co., 212 N. Vignes, Los Angeles, Calif.
Crawford, C. A. ('40) (JKS), Devel. Engrg., Internatl. Nickel Co., 67 Wall St., New York, N.Y.; *for mail,* 27 Dogwood Dr., Summit, N.J.
Crawford, Charles Campbell ('06; '16; '18) (EFHS), Pres., A. M. Lockett & Co., Ltd., 308 Whitney Bldg., New Orleans 7, La.
Crawford, Charles Wm. ('31) (BMSE), Head, Mech. Engrg. Dept., Agric. & Mech. College of Tex., College Station, Tex.
Crawford, Charles William ('44) (CJHDLB), V.P., Charge Sales, Consltd. Steel Corp., Box 6880, East Los Angeles Branch, Los Angeles 22, Calif.
Crawford, Chauncey H. ('16) (BCDORS), Mgr., Dept. of Reports & Appraisals, J. G. White Engrg. Corp., 80 Broad St., New York 4, N.Y.
Crawford, Clark Alvin (J'43) (BCNMJK), Devel. Engr., Corning Glass Works; *for mail,* 228 Strathmore Pl., Corning, N.Y.
Crawford, Ernest Augustus ('44) (BZCMRO), Mgr. of Engrg., Sperry Products, Inc., 15th & Willow Ave., Hoboken, N.J.
Crawford, Erwin Calyer (J'43), Technician, 5th Grade, G-2 Office, 2nd Inf. Div., Hdq., Ft. Lewis, Wash.
Crawford, George Edwin (J'43), U.S. Navy; 1454 N. Long Ave., Chicago 51, Ill.
Crawford, George W. (M'38) (CDFKS), Ch. Engr., E. Keeler Co., 238 West St., Williamsport 3; *for mail,* 108 Lafayette St., Muncy, Pa.
Crawford, Ian Cameron ('44) (CDM), 44 Franklin Terrace, Southbridge, Mass.
Crawford, James Vaile (J'40) (HKJLAB), Research Engr., Airesearch Mfg. Co., 9851 Sepulveda Blvd.; *for mail,* 8107 Airlane Ave., Los Angeles 45, Calif.
Crawford, Kenneth Zschokke (J'45) (BCM), Jr. Design Engr., Ford Instrument Co., Inc., 32-36—47th Ave., Long Island City 1; *for mail,* 49-30 Forest Dr., Douglaston, N.Y.
Crawford, M.H. (J'39) (FKS), Dow Magnesium Corp., Ludington, Mich.
Crawford, Melvin Wilbur (J'45) (MBCNO), 2535 Dunstan Rd., Houston 5, Tex.
Crawford, William F. ('43) (CSMJHK), Pres., Edward Valves, Inc., 1200 W. 145th St., East Chicago, Ind.; *for mail,* 6851 Constance Ave., Chicago 49, Ill.
Crawley, George Edwin ('08; '28) (CY), Gen. Mgr., Francisco Sugar Co., Francisco, Camagüey, Cuba.
Crawshaw, S. Leroy ('41) (CMBAEJ), West. Gear Works, 417—9th Ave. S., Seattle 4, Wash.
Creager, Paul Snyder ('45) (ZSL), Assoc. Prof. Elec. Engrg., Rutgers Univ., New Brunswick, N.J.
Crede, Charles E. (J'36) (BYTNAR), Ch. Engr., L. N. Barry Co., Inc., 179 Sidney St., Cambridge 39, Mass.
Crede, Richard E. (J'42) (CM), Gen. Mgr., Allkern Engrg. Co., 71 N. Clinton St., East Orange, N.J.; *for mail,* 201-18—109th Ave., Hollis 7, N.Y.
Credo, Julius ('16; '35), 2111 Edgehill Rd., Louisville 5, Ky.
Creech, Merl Daniel ('38; '44) (BNKLOH), Assoc. Prof. Mechanics, Univ. of Okla., Norman, Okla.

Creed, Edward Merrill (J'41) (BJS), 44 E. Washington St., Rutland, Vt.
Creel, William Hunt ('40) (ACEKL), Office Mgr., Phillips Petroleum Co., Bartlesville, Okla.
Crego, Donald Frank (J'41) (BESMHY), Mech. Engr., McCulloch Engrg. Corp., 3420 W. Capital Dr., Milwaukee 9, Wis.
Crenshaw, Bransford W. ('24; '34; '35) (CMJRLS), Sales Engr., Scullin Steel Co., 6700 Manchester St., St. Louis 10; *for mail,* 120 Orchard Ave., Webster Groves 19, Mo.
Crenshaw, William F. (J'37) (MPS), Petroleum Engr., Delta Drilling Co., Tyler, Tex.; *for mail,* 317 Edwards St., Ft. Collins, Colo.
Creutz, C. Emil ('39) (KLTWY), V.P., Am. Heat Reclaiming Corp., 1270—6th Ave., New York 20, N.Y.
Creveling, Doyle R. (J'45) (MNLBD), Engr., Am. Steel Band Co., Carnegie; *for mail,* 4810 Brownsville Rd., Pittsburgh 10, Pa.
Crewdson, Henry ('19) (BHOTY), Ch. Oper. Engr., Am. Viscose Corp., 35 S. 9th St., Philadelphia 7, Pa.
Crews, J. F. (J'39) (KLB), Ch. Engr., Design & Erection Process Equip., West. Condensing Co., 935 E. John St., Appleton, Wis.
Crews, Lloyd Albert (J'43) (HKS), 1st Lt., Army Air Forces; 84 N. Dearborn St., Indianapolis, Ind.
Crewson, George G. ('15; '19) (LKYTZF), Ch. Engr., Buffalo Electrochemical Co., Sta. B, Buffalo; *for mail,* 45 Coolidge Dr., Snyder, Buffalo 21, N.Y.
Crimp, George Bertram ('23; '32) (SJKFAD), Mech. Engr., Bur. of Engrg., Comptroller, City of New York, Rm. 625, Municipal Bldg., New York; *for mail,* 81 Columbia Heights, Brooklyn 2, N.Y.
Criner, Harry Everett (J'40) (BOAKH), Devel. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 17 Marion, Forest Hills, Pittsburgh 21, Pa.
Criswell, J. Carl, Jr., (J'38) (DCLMO), Engr., Am. Stores Co., 424 N. 19th St., Philadelphia 30, Pa.
Criswell, Wilbur Wilson, Jr. ('28; '44) (FSO), Fuel Consultant, United Eastern Coal Sales Corp., 420 Lexington Ave., New York 17, N.Y.
Critchlow, Howard Thompson, Jr. (J'44) (ACJ), Maj., Army Air Forces, 577 Rutherford Ave., Trenton 8, N.J.
Critz, Robert George (J'42) (BRS), Box 128, Shawnee, Okla.
Critzer, Ray D. (J'40) (CJM), 329 E. 326th St., Willoughby, Ohio.
Crocker, Allen S. ('05) (O), Private Cons. Engr., 311 Alexander St., Rochester 7, N.Y.
Crocker, George Holmes ('45) (CJM), Comdr. (Exec. Officer), Naval Ord. Plant, South Charleston, W. Va.
Crocker, John Walter ('38; '44) (SNEMLC), Sandy Hill Iron & Brass Works, Hudson Falls, N.Y.
Crocker, Sabin ('21; '25) (SFBEKH), *Junior Award, '23;* Sr. Engr., Mech. System Engrg. Dept., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 1037 Yorkshire Rd., Grosse Pointe Park 30, Mich.
Crocker, Sabin, Jr. (J'41) (SHBD), Asst. Engr., Shipbldg. Div., Bethlehem Steel Co., Quincy 69; for mail, 662 Main St., Hingham, Mass.
Crockett, Chas. H. ('18; '21; '35) (BW), Sales Engr., 221 Stow Ave., Troy, N.Y.
Crockford, Richard Hallas ('23; '28; '35) (FS), Mech. Engr., Reliance Engrg. Co., Box 1292, Charlotte 1; *for mail,* 1405 Pecan Ave., Charlotte 4, N.C.
Crofoot, George Emerson ('07; '13) (SRFWY), Prof. Mech. Engrg., Univ. of Pa., Philadelphia 39; *for mail,* 4535 Pine St., Philadelphia 43, Pa.
Croft, Huber Ogilvie ('20; '29; '32) (SKEF), *Manager,* '40-'43; Head Dept. of Mech. Engrg., Univ. of Iowa, 122 Engrg. Bldg., Iowa City, Iowa.
Croghan, John T. ('09; '14) (FCYLSN), Mech. Engr., Boston Woven Hose & Rubber Co., 29 Hampshire St., Cambridge 38; *for mail,* 574 Chestnut St., Waban 68, Mass.
Croke, Charles V. (J'41) (CJM), Falk Corp., 3001 W. Canal St., Milwaukee; *for mail,* 2512 N. 53rd St., Milwaukee 10, Wis.
Croker, Morris C. (J'39) (KFESO), Assoc. Mar. Engr., Puget Sound Naval Shipyard; *for mail,* 1215—4th St., Bremerton, Wash.
Croley, Jack G. (J'37), Capt., Btry. B, 113th Antiaircraft Arty. Gun Bn. (Mobile), A.P.O. 4774, c/o Postmaster, New York, N.Y.
Croll, Raymond H. ('26; '32; '35) (EHS), Dist. Mgr., Worthington Pump & Mchy. Corp., Rm. 2821, 436—7th Ave., Pittsburgh 19, Pa.
Cromer, Orville Charles ('40) (EF), Assoc. Prof. Mech. Engrg., Purdue Univ., Lafayette; *for mail,* 261 S. Salisbury St., West Lafayette, Ind.
Crompton, Edward Everett ('45) (BCDM), Pres., Gen. Mgr., Burke Elec. Co., 1202 Cranberry St.; *for mail,* 4424 Homeland Blvd., Erie, Pa.
Cromwell, Oliver C. ('14) (JKR), Retired; 2217 Elsinor Ave., Baltimore, Md.
Crone, James Murray ('45) (OCMAYZ), 402 W. Buffalo St., Ithaca, N.Y.
Cronemeyer, Henry C. ('12) (FKS), Designer, Jones & Laughlin Steel Corp., Aliquippa; *for mail,* 259 College Ave., Beaver, Pa.
Cronin, Frank Howard ('17), Mech. Engr., Natl. Vulcanized Fibre Co., Yorklyn, Del.
Cronin, John Robert (J'43) (MCAS), 7936 Clyde Ave., Chicago 17, Ill.
Crook, Frank P., Jr. (J'46), 1001 S. Windsor Blvd., Los Angeles, Calif.
Crooks, Homer L. (J'41) (KSFCL), Design Engr., C. F. Braun & Co., 1000 S. Fremont St., Alhambra; *for mail,* 801 S. Montecito Dr., San Gabriel, Calif.
Crooks, William Adam (A'44) (CJM), 6971 Dartmouth Ave., St. Louis, Mo.
Crosby, Edwin Stanislau ('21) (CKS), Pres., Johns-Manville Internatl. Corp., 22 E. 40th St., New York 16, N.Y.; *for mail,* 7 Washington Pk., Maplewood, N.J.
Crosby, George Frederick, Jr. ('34; '42) (BCJ), 596 Terrace Pl., Westfield, N.J.
Crosby, Robert William ('45) (ZLCW), Partner, Salesman & Engr., Crosby & Siegars, Co., 98 Exchange St., Portland 3; *for mail,* R.F.D., Raymond, Me.
Cross, Bertram J. ('36) (FKRSZ), Research Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Cross, Bertram J., Jr. (J'41), Route 9-W, Grand View, Nyack, N.Y.
Cross, Garett P. S. ('13) (NBJ), Research Div., United Shoe Mchy. Corp., Elliott St.; *for mail,* 6 Clifton Ave., Beverly, Mass.
Cross, Harold W. ('20) (EFKS), Dist. Turbine Specialist, Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Cross, Louis Frederick, Jr. (J'40) (OELFHN), Design Engr., Tide Water Associated Oil Co., E. 22nd St., Bayonne; *for mail,* 35 Kensington Ave., Jersey City 4, N.J.
Cross, Malcolm Albert (J'45) (LSD), Mech. Engr., Pa. Salt Mfg. Co., 1000 Widener Bldg., Philadelphia 7; *for mail,* 120 Rodgers St., Ridley Park, Pa.
Cross, Robert Ames ('21; '35) (P), Deere & Co.; *for mail,* 2335—23rd Ave., Moline, Ill.
Cross, Robert C. ('30; '36) (FNM), Head, Mech. Div., Dept. 817, Sears, Roebuck & Co., 925 S. Homan Ave., Chicago 7, Ill.
Cross, Sherman S. ('43; '44) (ACHJMN), Asst. to Supt., Weatherhead Co., E. 131st St., Cleveland; *for mail,* 241 Monroe St., Cuyahoga Falls, Ohio.
Cross, Wallace Jackson ('14; '21), (LKEFSH), Asst. Ch. Designing Engr., Sinclair Refining Co., 3500 Indianapolis Blvd., East Chicago, Ind.
Crossan, Thomas Edmund ('25; '35) (SFCKBE), Supt. Prod., Gulf States Utilities, Florida St., Baton Rouge 2, La.
Crossen John ('43) (JLM), Mech. Engr., Chem. Warfare Serv., War Dept., Dugway Proving Ground, Tooele, Utah.
Crossley, Francis Rendel Erskine (J'44) (BNEG), Visiting Lecturer (Asst. Prof.), Yale Univ., 400 Temple St., New Haven 11, Conn.
Crossley, Walter C. (J'27), Capt., Corps of Engrs., War Dept. Personnel Center, Serv. Comd. Unit 1773 Hdq. Armed Forces Induction Sta., Ft. Leavenworth, Kan.
Crossman, Albert ('03), Retired; 801 Florida Ave., York, Pa.
Crotty, Joseph James ('32) (BHM), V.P., Cent. Fdy. Co., 386—4th Ave., New York 16, N.Y.
Crouse, Edwin Roy (J'41) (NBJMHL), Mech. Engr., Design Presses, Verson Allsteel Press Co., 1355 E. 93rd St., Chicago; *for mail,* 7753 Crandon Ave., Chicago 49, Ill.
Crovatto, Philip R. ('32; '35), Draftsman, Design, Interborough Rapid Transit Co., Rm. 210, 2545—7th Ave.; *for mail,* 74 Clinton Pl., New York, N.Y.
Crowe, George Francis ('43) (BCEHNS), Engr., Hyd. Power, New England Power Serv. Co., 441 Stuart St., Boston; *for mail,* 167 Warren Ave., Wollaston 70, Mass.
Crowe, John Joseph ('45), Asst. V.P., Air Reduction Co., Inc., 181 Pacific Ave., Jersey City 4, N.J.
Crowell, Henry Walcott ('12) (C), Supvg. Elevator Engr., Eagle, Globe & Royal Indemnity Cos., 150 William St., New York 8, N.Y.; *for mail,* 25 Woodland Ave., Glen Ridge, N.J.
Crowell, Miss Lisbeth (J'45) (NZBM), Jr. Mech. Engr., Philco Products Inc., Tioga & C St., Philadelphia; *for mail,* 2103 Walnut St., Philadelphia 3, Pa.
Crowell, Samuel, III (J'39) (KFS), 19 Hillside Ave., Marblehead, Mass.
Crowley, Charles P. ('20; '23; '35) (SKE), Owner, Charles P. Crowley Co., 711 Gibbons St., Los Angeles 31, Calif.
Crowley, Henry L. ('18; '25; '27) (JKMYNL), Pres., Ch. Engr., Henry L. Crowley & Co., Inc., 1 Central Ave., West Orange, N. J.
Crowley, Raymond Woodbury (J'41) (EAFJNS), Assoc. Mar. Engr., Bur. of Ships, Navy Dept., 16th St. & Constitution Ave., N.W., Washington D.C; *for mail,* 403 George Mason Dr., Arlington, Va.
Crownfield, Albert Charles, Jr. ('45) (CMKFG), Prod. Supt., Allen Mfg. Co., Drawer 570, Hartford 1; *for mail,* 138 Clearfield Rd., Wethersfield 9, Conn.
Crownover, Joseph C. (J'36) (KLS), R.D. 1, Nashotah, Wis.
Crowser, Kenneth Eugene (J'41) (SODKME), Gifford, Idaho.
Cruickshanks, Benjamin Carpenter ('32) (BKSHEC), Prof. Mech. Engrg., The George Washington Univ., Washington 6; *for mail,* 5733—4th St., N.W., Washington 11, D.C.
Cruise, Donald Philip (J'40) (NJG), Designer. Babcock Ptg. Press Corp., 88 Pequot Ave.; *for mail,* 580 Colman St., New London, Conn.
Crum, John O. (J'40), Capt., Ord. Dept., Hdq., Advan., 5th Air Force, A.P.O. 713, Unit 1, c/o Postmaster, San Francisco, Calif.
Crumb, Carl Brown, Jr. (J'42) (AMSCZ), Box 1663, Santa Fe, New Mex.
Cryer, Charles Pickett (J'44), Lt. (j.g.), U.S.N.R., U.S.S. *Acree,* DE-167, F.P.O., San Francisco, Calif.
Csisek, John Jacob ('28; '33; '35), Asst. Ch. Inspr., Cameron Pump Div., Ingersoll-Rand Co., Phillipsburg; *for mail,* Box 54, West Portal, N.J.
Cubberly, Walter E., Jr. (J'41), 1635 Branard St., Houston 6, Tex.
Cubbidge, James Frederick (J'44), 4729 W. Deming Pl., Chicago 39, Ill.
Cucullu, Lionel Joseph ('36; '41) (SCEFK), Asst. to Ch. Engr., New Orleans Pub. Serv. Inc., 317 Baronne St., New Orleans 9, La.
Cuddeback, Alva Elston ('44) (CMLDSY), Plant Mgr., Continental Foods Inc., 1500 Hudson St., Hoboken, N.J.; *for mail,* 32-15—215th St., Bayside, L.I., N.Y.
• **Cudebec, Albert Bennett** ('41) (HON), V.P., Hydropress, Inc., 570 Lexington Ave., New York 22, N.Y.
Cudin, Lou Edward (J'45) (LMJDCG), Process, Prod. Engr., John Inglis Co. Ltd., 14 Strachan Ave., Toronto, Ont., Can.
Cuff, Harold B. ('26; '35) (YSKB), Mech. Engr., Am. Hard Rubber Co., 11 Mercer St., New York, N.Y.; *for mail,* 168—1st Ave., Little Falls, N.J.
Culbertson, Dan ('24; '26; '35) (RJYZSE), Gen. Matl. Inspr., Atchison, Topeka & Santa Fe Ry. Co., 80 E. Jackson Blvd., Chicago 4, Ill.
Culham, Harold Douglas (J'45) (NMHACD), Asst. Ch. Engr., Wallaceburg Brass Co., Wallaceburg; *for mail,* R.R. 4, Malton, Ont., Can.
Cullen, Thomas Joseph (J'25) (KLRS), Field Rep., Ruberoid Co., Baltimore 24, Md.; *for mail,* 234 Emerson Ave., Charlotte 4, N.C.
Cullen, Wallace John (J'44) (MESCKF), Cpl., 3rd Serv. Comd., U.S.A., Reception Sta. 21, Indiantown Gap Military Reservation, Indiantown Gap; *for mail,* 924 Fisk St., Scranton 9, Pa.
Cullen, William James Jr. (J'43) (ACS), 24 Washington Park, Maplewood, N.J.
Cullimore, Allan Reginald ('33) (CO), Pres., Newark College of Engrg., 367 High St., Newark 2, N.J.
Cullison, James Stephen (J'41) (MCDJY), Research Engr., Buhl Mfg. Co., Scotten Ave., Detroit, Mich.
Culp, Herbert Paul (J'41) (SHKBZC), Engr., Turbine Thermo. Research, Gen. Elec. Co., 920 Western Ave., Lynn 3; *for mail,* 15 Green St., Marblehead, Mass.
Cummings, Charles Damon, Jr. (J'46), 140 Silver St., Dover, N.H.
Cummings, Edgar Herbert ('45), Mech. Engr., Am. Monorail Co., 13107 Athens Ave., Cleveland 7; *for mail,* 1602 Maple Rd., Cleveland Heights 21, Ohio.
Cummings, Frank S. ('19; '24; '32) (FS), Mech. Engr., C. C. Moore & Co., Engrs., 450 Mission St., San Francisco; *for mail,* 71 Ross Circle, Oakland, Calif.
Cummings, James D. ('23; '31; '35) (YWT), Mem. Tech. Staff, Bell Tel. Labs., 463 West St., New York 14; *for mail,* 3515—164th St., Flushing, N.Y.
Cummings, Lloyd Anderson ('27) (ABCDJM), Ch. Engr., Marlin-Rockwell Corp., 402 Chandler St., Jamestown 55, N.Y.
Cummings, Orrie Pratt ('99), Sales Engr., Watson Elev. Co., 407 W. 36th St., New York, N.Y.
Cummings, Robert Felt ('23; '30; '35) (SFEKJB), Ch. Engr., Burns & Roe, Inc., 233 Broadway, New York 7, N.Y.; *for mail,* 169 Grand Ave., Leonia, N.J.
Cummings, Walter J. K. (J'41) (BEFKLZ), Engr., Atlantic Refining Co., 260 S. Broad St., Philadelphia; *for mail,* 963 N. American St., Philadelphia 23, Pa.
Cummins, Andrew Adair ('43) (SOLEFK), Pres. & Engr., Cummins & Barnard, Inc., 205 State Savings Bank Bldg; *for mail,* 1402 Henry St., Ann Arbor, Mich.
Cummins, Clessie L. ('31), Pres., Cummins Eng. Co., Columbus, Ind.
Cummins, Norman Wheeler ('18) (CMJ), Pres., Bade-Cummins Mfg., Inc., 2501 Jewel St., Louisville 12; *for mail,* 2531 Cherokee Pkwy., Louisville 5, Ky.
Cummiskey, William Michael ('14) (DME), Project Engr., Watson Elev. Co., Inc., 45 Cedar Lane, Englewood; *for mail,* 103 N. 9th St., Newark 7, N.J.
Cumner, Matthew Smith ('12) (JKLMN), Spec. Rep., Wyner Mch. Works, Inc., 251—3rd Ave., New York 10; *for mail,* 2215 Newkirk Ave., Brooklyn 26, N.Y.
Cundiff, Charles Reed ('44) (BKS), Comdr., U.S.N.; *for mail,* 3625 Greenway Pl., Alexandria, Va.
Cunning, Jas., Jr. (J'33), Machinist's Mate, 2nd Class, U.S.S. *Weehawken,* c/o Postmaster, New York, N.Y.; *for mail,* 2625 N. Meridian St., Indianapolis, Ind.
Cunningham, Francis ('30) (O), Treas., John A. Stevens, Inc., 16 Shattuck St., Lowell, Mass.

Cunningham, James Dalton ('18; '21; F'36), *Manager*, '29-'32, *Vice-President*, '32-'34; Pres., Republic Flow Meters Co., 2240 Diversey Pkwy., Chicago 47, Ill.
Cunningham, Robert O. ('42) (CDJLM), Pres., Clough, Roberts & Cunningham, 3510 Woodward Ave., Detroit 1, Mich.
Cunningham, Robert Sandys (J'42) (CMOLNH), Steam Turbine Engr., Allis-Chalmers Mfg. Co.; *for mail*, 4068 N. Lake Dr., Milwaukee 1, Wis.
Cuozzo, Dante (J'45) (BJM), Mech. Engr., Naval Shipyard, Boston; *for mail*, 110 Medford St., Charlestown 29, Mass.
Curcio, Anthony P. (J'38) (KHOS), 8 Hackett Circle W., Stamford, Conn.
Curlee, Conrad J. (J'40) (FKS), 4307 Byrd St., Flushing, N.Y.
Curley, Chester Clarence, Jr. (J'41) (JNYM DB), Engrg. Matl. & Equip. Inspr., U.S. Naval Inspc. Serv., Rm. 220, P.O. Bldg.; *for mail*, 208 Clymer St., Reading, Pa.
Curley, Matthew H. ('43) (FSKEC), Mech. Engr., 2nd Serv. Comd. Engrs., 270 Broadway, New York 7; *for mail*, Three Sisters Rd., St. James, N.Y.
Curley, Wm. S. J. (J'38), Lt., 393rd Bomb Sqd., Westover Field, Mass.
Curran, Henry Michael (J'42), Mech. Engr., Research & Devel. Gas Turbines, Elliott Co., 207 Walnut Ave., Greensburg, Pa.
Curren, Ralph Lawrence ('36) (MBC), Project Engr., West. Elec. Co., Inc., 529 W. 42nd St., New York; *for mail*, 63 Ross Ave., Staten Island 6, N.Y.
Curreri, Robert I. (J'34) (MCGKLJ), 1542—57th St., Brooklyn 19, N.Y.
Currie, Gardner David (J'43) (AME), Seaman, 1st Class, U.S.N.R., Unit 1, Ward C-3, U.S. Naval Hospital, Corona, Calif.
Curry, Edmund Charles (J'45) (BEHS), Asst. Mech. Design Engr., Worthington Pump & Mchy. Corp.; *for mail*, 38 W. Fassett St., Wellsville, N.Y.
Curry, Ezra Benham ('24; '35) (COAR), 1203 Laurel Ave., St. Paul 4, Minn.
Curry, Malcolm ('17) (TS), Gen. Engr., Am. Thread Co., 260 W. Broadway, New York 13, N.Y.
Curtis, Allen James (J'46) (BE), Ensign, U.S.N.R., U.S.S. *DuPage*, APA-41, F.P.O., New York, N.Y.; *for mail*, 1310 Bancroft St., Dayton 8, Ohio.
Curtis, Bruce Donald (J'46) (CGM), Mech. Engr., Allen Calculators, Inc., 678 Front Ave., N.W., Grand Rapids 4; *for mail*, 2230 Monroe Ave., N.W., Grand Rapids 5, Mich.
Curtis, Edma H. ('01; '26), Maint. Engr., Nivison Weiskopf Co., Main St., Reading 15; *for mail*, 702 Ridgeway Ave., Cincinnati 29, Ohio.
Curtis, Elbert Dean ('45) (CJM), Ch. Draftsman, West. Pipe & Steel Co., 5717 Santa Fe Ave., Los Angeles; *for mail*, 3717 Santa Ana St., Huntington Park, Calif.
Curtis, Jack Warren (J'42) (ABN), Draftsman, Calif. Packing Corp., 101 California St., San Francisco; *for mail*, 1824 7th Ave., Oakland 6, Calif.
Curtis, Ralph Edgar ('88; '01) (SFRE), Retired; 35 School St., Danvers, Mass.
Curtis, Robert Wade (J'41) (SONKJM), Lt. (j.g.), U.S.N., Aero. Engr., Box C, Naval Air Sta., Quonset Point, R.I.; *for mail*, R.F.D. 1, Phalanx Station, Ohio.
Curtis, William H. ('29) (ABJKSY), Resident Engr., Pacific Coast, Thompson Products, Inc., 2196 Clarkwood Rd., Cleveland 3, Ohio; *for mail*, 4141 Knob Hill Dr., Sherman Oaks, Calif.
Curtiss, Charles Benjamin ('15; '28) (NDCJS), Engr., Valley Welding & Boiler Co., 27th & Water Sts.; *for mail*, 924 Center Ave., Bay City, Mich.
Curtiss, Howard Crosby ('44) (FKSL), Mech. Engr., Union Iron Works, Erie, Pa.
Curtiss, William L. (J'37) (HK), Process Engr., Lago Oil & Transport Co., Aruba, N.W.I.
Cushing, Henry J. (J'22) (CFKS), Asst. Mgr., Commercial Relations, N.Y. Steam Corp., 130 E. 15th St., New York 3; *for mail*, 884 Annadale Rd., Annadale, S.I. 12, N.Y.
Cushing, Thomas Elwyn ('28; '30; '35) (BJMN RZ), Supvr. Mch. Tool Applications, SKF Industries, Inc., Front St. & Erie Ave.; *for mail*, 222 W. Johnson St., Philadelphia 44, Pa.
Cushman, John Arthur, Jr. (J'42) (CTBM), Asst. Plant Engr., Am. Viscose Corp., Lewistown, Pa.
Cushman, Paul Allerton ('19; '24) (JCBN), Metallurgist & Test Engr., McGill Mfg. Co.; *for mail*, 803 Brown Ave., Valparaiso, Ind.
Cuthbert, Ivan Norman ('28) (OMELK), Ch. Indus. Engr., Smith, Hinchman & Grylls, Inc., 800 Marquette Bldg., Detroit 26; *for mail*, 2881 N. Platt Rd., Ann Arbor, Mich.
Cutler, Arthur E. ('10) (CFS), Retired; R.F.D. 3, Attleboro, Mass.
Cutler, James B. ('17; '24) (HSNCJB), Mech. Engr., Pa. Water & Power Co., Lexington Bldg., Baltimore 1; *for mail*, 6103 Pinehurst Rd., Baltimore 12, Md.
Cutler, Wallace Milton ('27) (BCHMNO), Owner, Wallace M. Cutler Associates, 53 Hampshire St., Cambridge 39, Mass.
Cutton, Leverett H. ('23), Plant Engr., Mack Mfg. Corp; *for mail*, 2815 Washington St., Allentown, Pa.
Cutter, Geo. A. ('96; '04), 161 Pleasant St., Lynn, Mass.
Cygan, Roman (J'44) (KHS), Test Engr., Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.; *for mail*, 21 Garfield Ave., Easthampton, Mass.
Cyphers, James F. ('14; '21) (CM), V.P., Prod. Mgr., Walter Baker & Co., 1197 Washington St., Dorchester 24; *for mail*, 75 Hinckley Rd., Milton 87, Mass.
Cyrol, Edmund Alexander ('43; '45) (CDMO), Jr. Partner, McKinsey, Kearney & Co., 135 S. La Salle St., Chicago 3, Ill.
Czajkowski, Edward Carl (J'34), Shipfitter, 3rd Class, U.S.N., Constr. Bn. Maint. Unit 567 Hdq. Navy 157, F.P.O., New York, N.Y.
Czapek, Edmund Leon (J'43) (KBJLHS), Research Engr., Linde Air Products Co., East Park Dr. & Woodward Ave., Tonawanda, N.Y.; *for mail*, 96 Pine Grove Ave., Lynn, Mass.
Czekalski, Wallace Martin (J'40) (YZTJOC), Engr., World Bestos Corp., 52 Courtland St., Paterson; *for mail*, 314 Madison Ave., Paterson 4, N.J.
Czock, Jacob H. ('41) (EBCMJ), Branch Sales Office Mgr., Atlas Imperial Diesel Eng. Co., 102 New Montgomery St., San Francisco 5; *for mail*, 4697 Park Blvd., Oakland 2, Calif.

D

Daasch, Francis J. ('36; '41) (ESBJFN), Mech. Engr., Gulf Oil Corp., Box 661, Tulsa 2, Okla.
Dabney, Robert Raymond ('46) (FLS), Power Engr., City Pub. Serv. Bd., San Antonio 6, Tex.
da Costa, Gerson (J'31), Cent. Stands. Office for Rys., Ry. Bd., "N" Block, New Delhi, India.
Dadley, James W. ('15; '22) (LKDNC), Design Engr., Pa. Salt Mfg. Co., 1000 Widener Bldg., Philadelphia; *for mail*, 656 Fern St., Lansdowne, Pa.
Dague, Arthur Demarest (J'45), Students Trng., Westinghouse Elec. Corp., 3N-65, East Pittsburgh, Pa.
Dahl, Bernard Fred (J'42) (SMEO), Sales Engr., Bunting Brass & Bronze Co., 715-755 Spencer St.; *for mail*, Box 5026, Cleveland, Ohio.
Dahl, Norman Fritjoph (A'46), Supvr. of Installation, George S. May Co., 122 E. 42nd St., New York; *for mail*, 93—71st St., Brooklyn, N.Y.
Dahl, Peter Gudikson ('40) (SCFK), Asst. Ch. Engr., Commonwealth Edison Co., 3400 N. California Ave., Chicago 18, Ill.
Dahlman, Fred Albert ('44) (NHMBD), Mech. Research Engr., Corning Glass Works; *for mail*, 147 W. 5th St., Corning, N.Y.
Dahlquist, Daniel William (J'39) (KEBWDH), York Corp., 5051 Santa Fe Ave., Los Angeles; *for mail*, 1275—14th St., San Pedro, Calif.
Dahlquist, John L. (J'41), 732 S. Crescent Ave., Park Ridge, Ill.
Dahlstrand, Hans P. ('10) (SJBMK), Cons. Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 7037 Maple Terrace, Wauwatosa 13, Wis.
Dahlund, Ervin Lewis ('35; '45) (ENCSR), Asst. Ch. Engr., Diesel Devel. & Exper. Div., Fairbanks, Morse & Co., Lawton Ave.; *for mail*, 1163½ Milwaukee Rd., Beloit, Wis.
Dahms, Alfred Albert ('44) (CSKHBL), Kansas City Dist. Mgr., Allis-Chalmers Mfg. Co., 1410 Waldheim Bldg., Kansas City 6, Mo.
Daiger, George Pierce ('44) (CNMYJA), Exec. Engr., Hoover Co., North Canton; *for mail*, 1103—22nd St., N.E., Canton 4, Ohio.
Dailey, W. H., Jr. (J'34) (FL), Power & Fuel Engr., Irvin Works, Carnegie-Ill. Steel Corp., Dravosburg; *for mail*, R.D. 1, Ridge Rd., Library, Pa.
Daily, James Wallace (J'37) (HBSAE), Hyd. Engr., Hyd. Research, Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.
Dalby, Vernon L. (J'40) (JKS), Capt., Engr. Sec., Hdq., Seine Sec., A.P.O. 887, c/o Postmaster, New York, N.Y.
Dale, David Noel (J'34) (CEAMJ), Mech. Engr., Mosher Steel Co., 3910 Washington Ave., Houston, Tex.
Dale, G. Earl (J'45) (MEFXP), Design Sec., Bldg. 89, Naval Shipyard, Boston 29, Mass.
Dale, Paul I. (J'42) (AMNC), Ch. Draftsman, Stokermatic Div., Rheem Mfg. Co., 3425 Skedzie St.; *for mail*, 8418 Jeffery St., Chicago, Ill.
Dale, Robert Burdette ('17) (CB), Cons. Engr., Office 1650, J. H. Manning & Co., 120 Broadway, New York 5; *for mail*, 84-21—168th Pl., Jamaica 3, N.Y.
Daleda, Jos. (J'34) (CDL), Pat. Examiner, Div. 40, Rm. 7880, U.S. Pat. Office, Washington, D.C.
Dalessio, John (J'41) (HWA), c/o Miss Grace Badaracco, Stevens Inst. of Tech., 5th St., Hoboken, N.J.
Daley, Joseph Aloysius, Jr. (J'43) (AFEC), Aero. Eng. Lab., Naval Air Exper. Sta., Naval Air Matl. Center, Philadelphia 12, Pa.; *for mail*, 2275 Hollinshed Ave., Merchantville, N.J.
Daley, Joseph Raymond ('45), Ch. Engr., Willamette Iron & Steel Corp., 3050 N.W. Front Ave., Portland 10; *for mail*, 5030 S.W. Humphrey Blvd., Portland 1, Ore.
Dallas, Charles Frederick ('19) (ABCDOS), Mgr., Antillian Constr. Co., 64 Calle Cuba (P.O. Box 2170), Havana, Cuba.
Dallimonti, Renzo (J'43) (NBMS), Lt. (j.g.), U.S.N.R., Asst. Engrg. Officer, U.S.S. *Sitka*, APA-113, F.P.O., San Francisco, Calif.
Dallner, Ray Wendell (J'39) (CDKL), Plant Engr., Cent. Soya Co., Inc., N. 2nd St.; *for mail*, 430 N. 5th St., Decatur, Ind.
Dalrymple, Arthur Wickman ('23; '34) (EF SH), Mech. Engr., Ellicott Mch. Corp., 1611 Bush St., Baltimore, Md.; *for mail*, 48 Barker Ave., White Plains, N.Y.
Dalrymple, James Ross (J'44) (CBMFD), Mech. Engr., Dominion Fdys. & Steel Ltd.; *for mail*, 17 Fairholt Rd. S., Hamilton, Ont., Can.
Dalrymple, Philip W. ('36; '38), 56 Crescent Ave., Newton Centre, Mass.
Dalrymple, Stewart Willard (J'41) (MK), Lt., U.S.N.R., Naval Inspr. of Ord., Chrysler Corp., 6334 Lynch Rd., Detroit, Mich.; *for mail*, 56 Crescent Ave., Newton Centre, Mass.
Dalton, Howard H. ('20) (BDFKS), Power Engr., Am. Sugar Refining Co., 120 Wall St., New York 5, N.Y.
Dalton, Thomas Edward ('34; '35) (CGM), Prod. Mgr., Charge Mfg., Hearst Magazines, Inc., 959—8th Ave., New York 19, N.Y.
Dalton, Thomas Nelson (J'42) (CLKAFH), Process Engr., M. W. Kellogg Co., Foot of Danforth Ave., Jersey City 3; *for mail*, 471 Union Ave., Elizabeth 3, N.J.
Dalton, William ('02; F'44) (CBROA), Retired; R.D. 2, Schenectady, N.Y.
Daly, Edmund Joseph ('26) (BEFHKS), Pres., Treas., M. J. Daly & Sons, Inc., 541-75 Bank St., Waterbury 88, Conn.
Daly, George Francis ('44), Devel. Engr., Internatl. Business Mchs. Corp.; *for mail*, 502 Sunset Terrace, Endicott, N.Y.
Dalzell, R. Carson ('40) (CKLJS), Ch. Tech. Adviser, Revere Copper & Brass, Inc., Rome, N.Y.
Dalziel, Philip Smith (J'35) (LCMD), Project Engr., Giffels & Vallet, Inc., 1000 Marquette Bldg., Detroit 26, Mich.
Dam, Cyrus King ('29; '40) (AES), Maj., Industry Branch, Office of Military Govt. for Bavaria, A.P.O. 403, New York, N.Y.
D'Ambrosi, Amerigo A. (J'39) (SEL), Maritime Engr., Trans. Corps, Camp Gordon, Johnston, Fla.; *for mail*, 225 W. 232nd St., Bronx 63, N.Y.
Dame, Frank E. (J'33), Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail*, Apt. 1D, 70-10—108th St., Forest Hills, N.Y.
Damm, Nelson E. (J'41) (JMW), 916 Leahy St., Muskegon Heights, Mich.
Damon, John Henry ('08; '10) (BCMOST), Retired; 258 Court St., Plymouth, Mass.
Damon, Ralph Shepard ('41) (ACF), V.P., Gen. Mgr., Am. Airlines, Inc., 100 E. 42nd St., New York 17, N.Y.
Dams, Maurice (J'44) (ACJ), Student Engr., Stamping Div., Gen Motors Corp., 300 Allen Rd., S.W., Grand Rapids; *for mail*, 316 W. 13th St., Holland, Mich.
Dana, Benton Patterson ('44) (LDSNOC), Mech. Engr. III, Dept. of Chem. Engrg., Tenn. Valley Authority, Wilson Dam; *for mail*, 433 N. Walnut St., Florence, Ala.
Dana, Marshall Merritt ('39) (BCE), Capt., U.S.N., Pacific Coast Sec., Bd. of Inspc. & Survey, 105 Montgomery St., San Francisco 4, Calif.
Danatos, Steven (J'41) (EAFYZL), Instr., Stevens Inst. of Tech., Hoboken; *for mail*, 11 W. 33rd St., Bayonne, N.J.
D'Ancona, Vivaldo Levi (J'46), Cpl., Hdq., I Corps, G-2, A.P.O. 301, c/o Postmaster, San Francisco, Calif.
Danforth, John Philbrick (J'39) (SFEKB), Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 106 Park Ave., Williston Park, N.Y.
Daniel, Clarence Peck ('22) (CMDB), Pres., Enterprise Wheel & Car Corp., P.O. Drawer 151; *for mail*, 461 Euclid Ave., Bristol, Va.
Daniel, Thomas Allen (J'41) (NSEJMB), Estimating Engr., Consltd. Steel Corp. of Tex.; *for mail*, 306 Dewey Pl., Orange, Tex.
Daniele, Edmund ('40) (CMO), Comdr., U.S.N.; 3441—78th St., Jackson Heights, L.I., N.Y.
Daniels, Arthur N. (J'36), New Hampshire Ballbearings, Inc., 2 Main St., Peterborough, N.H.
Daniels, Clarence White ('16; '21) (SZJFCD), Plant Engr., Norton Co., 1 New Bond St., Worcester 6; *for mail*, 9 Metcalf St., Worcester 5, Mass.
Daniels, E. Ralph ('43) (OSKFCZ), Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia, Pa.
Daniels, Fred Harold ('18; '26) (KSF), Pres., Riley Stoker Corp., 9 Neponset St., Worcester; *for mail*, 190 Salisbury St., Worcester 5, Mass.
Daniels, Frederick Arthur ('44) (NMCYWH), Ch. Engr., Eagle Pencil Co., 713 E. 14th St., New York; *for mail*, 200 S. Broadway, Tarrytown, N.Y.
Daniels, George C. ('14; '18) (SFKD), Mech. Engr., Commonwealth & So. Corp., Consumers Power Bldg., Jackson, Mich.
Daniels, Malcolm G. (J'39) (HKS), 746 Oakridge Dr., Jackson, Mich.
Daniels, Robert Louis (J'45) (SKHFEJ), Jr. Engr., Trng. Program, Foster Wheeler Corp., Dansville; *for mail*, 1540 E. 17th St., Brooklyn 33, N.Y.
Danielsen, Alsed L. (J'40) (ZSE), Sales Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio; *for mail*, Box 365, Denver, Colo.
Danielsen, John E. ('25) (CKL), Planning Engr., Monsanto Chem. Co., 1724 S. 2nd St., St. Louis; *for mail*, 407 Lee Ave., Webster Groves 19, Mo.

Danielson, Lloyd Clifford (J'42) (KFSC), Ch. Engr., J. A. Jones Constr. Co., Inc., Box 299, Oak Ridge, Tenn.
Dankenbring, Henry Dieter (J'41) (ABN), Layout Draftsman, Glenn L. Martin Co., Baltimore 3; *for mail*, 6920 Old Harford Rd., Baltimore 14, Md.
Dankenbring, Siegfried F. (J'45) (BLS), Buell Engrg. Co., 70 Pine St., New York; *for mail*, 253 S. 9th St., Brooklyn, N.Y.
Danker, Frederick Richard (J'45), 1257 Park Ave., New York, N.Y.
Dann, Donald Albert (J'43) (NLHOBW), Designing Engr., Que. N. Shore Paper Co.; *for mail*, Box 136, Baie Comeau, Que., Can.
Dann, Willard Jerome ('33; '41) (RSCEYL), Asst. to Engr. of Tests, Lab., Chicago, Burlington & Quincy R.R., Aurora, Ill.
Dannehower, Gilbert Lane ('43), Propr., High Precision Products Co., 816 Boulevard, Westfield, N.J.
Danneman, Fred C. ('43) (ACM), Pres., Acme-Danneman Co., Inc., 203 Lafayette St., New York 12; *for mail*, 1669 Linden St., Brooklyn 27, N.Y.
Dannemann, Henry Frederick, Jr., (J'33) (MNEY), Mech. Engr., Ord. Dept., Frankford Arsenal, Philadelphia 37, Pa.; *for mail*, 406 Chicago Blvd., Sea Girt, N.J.
Dannettel, Raymond Charles ('24; '35) (SFCBK), Asst. to Elec. Engr., Consltd. Gas, Elec. Light & Power Co., Baltimore 3; *for mail*, 5718 Kenmore Rd., Baltimore 10, Md.
Dannettell, Harry W., Jr. (J'35) (CFS), Lt., QM. Corps, U.S.A., Commissary Sales Officer, Sheppard Field, Wichita Falls, Tex.; *for mail*, 230 Hampden Ave., Narberth, Pa.
Dans, Bernard David ('45), Design Engr., 6712—21st Ave., Brooklyn, N.Y.
Danse, Robert Atherton (J'37) (LECD), Engr., Designs Div., Stand. Oil Co. of Calif.; *for mail*, 1664 San Benito St., Richmond, Calif.
Dansie, George Walter, Jr. (J'40) (CBOWY), Lt. Comdr., U.S.N.R., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail*, 1285 Stratford Ave., Salt Lake City 5, Utah.
Danson, John Oliver (J'45) (EFS), Engr., Commonwealth Edison Co., 1111 W. Cermak Rd., Chicago 8; *for mail*, 7316 S. Aberdeen St., Chicago 21, Ill.
D'Antoni, Paul Anthony (J'43) (AKS), Apt. 6, 2460 Cedar Ave., Long Beach 6, Calif.
Danz, Harry O. ('38), Mgr., Dust Collector Dept., Am. Blower Corp., 6000 Russell; *for mail*, 3214 Vicksburg Ave., Detroit, Mich.
Danziger, Max J. (J'45), United Nations Relief & Rehabilitation Admin.; c/o Harris Danziger, 1460 Macombs Rd., Bronx 52, N.Y.
D'Aquila, Salvatore P. (J'41) (MCJ), Mech. Engr. (Sales Trainee), Fafnir Bearing Co., Booth St.; *for mail*, 48 Shuttle Meadow Ave., New Britain, Conn.
Darbee, William ('00; '12), Retired; New Preston, Conn.
Darby, Harry ('21; '22; '30) (CJ), Chmn. of Bd., Darby Corp., 1st & Walker Ave., Kansas City 15, Kan.
Darby, John ('07) (ABJNY), Retired; 195 Boyden Ave., Maplewood, N.J.
D'Arcey, Alfred Clemont ('34; '44) (HNMBJZ), Project Engr., Mar. Div., Bendix Aviation Corp., Norwood; *for mail*, 55 Lakeshore Dr., Westwood, Mass.
D'Arcy, Albert Joseph (J'36) (CLMAYJ), Gen. Pur. Dept., 8th Fl., Union Carbide & Carbon Corp., 30 E. 42nd St., New York 17, N.Y.
Darcy, Bernard Edward ('43) (YWMH), Ch. Engr., U.S. Cabinet Bed Co., 36 Milford St., Brooklyn, N.Y.
D'Arcy, Francis G. ('35), Process Engr., Bridgeport Brass Co., 30 Grand St., Bridgeport; *for mail*, 401 Hilltop Dr., Stratford, Conn.
D'Arcy, James Maurice (A'44) (BCW), 432 Pacific St., Brooklyn, N.Y.
Darden, Clarence Marsden ('41) (RSCDFB), Supt. Mchy., Nashville, Chattanooga & St. Louis Ry., Nashville; *for mail*, Forrest Park Ave., Nashville 5, Tenn.
Darke, Robert Stevens (J'39) (ZS), Sales Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail*, 15908 Euclid Ave., East Cleveland, Ohio.
Darling, Douglas George (J'46) (CMP), Demonstrator, Dept. of Mech. Engrg., Mech. Bldg., Univ. of Toronto; *for mail*, 343 Annette St., Toronto, Ont., Can.
Darling, Erwin E. ('29; '35) (BDM), Shop Engr., Dodge Main Plant, Chrysler Corp., 7900 Jos. Campau Ave., Hamtramck, Detroit; *for mail*, 230 Pleasant St., Birmingham, Mich.
Darling, Kenneth Mason (J'36) (REAS), Asst. to Supvr., Diesel Engines, Atchison, Topeka & Santa Fe Ry. Co., 1700 S. Wentworth Ave., Chicago; *for mail*, 324 N. Stone Ave., La Grange, Ill.
Darling, Leonard Bruce (J'44) (DZSLOK), Asst. Plant Engr., Distillation Products, Inc., 755 Ridge Rd., W.; *for mail*, 34 West Pkwy., Rochester 13, N.Y.
Darling, Philip E. ('41) (LSZKC), Ch. Design Engr., Pan-Am. Refining Corp., Box 401; *for mail*, 125—10th Ave. N., Texas City, Tex.
Darling, William Myron, Jr. (J'41) (AB), Capt., Army Air Forces, Project Engr., Air Tech. Serv. Comd., Wright Field; *for mail*, 75 Patterson Village Dr., Dayton, Ohio.
Darmody, William Joseph ('46) (JMZ), Lt. Col., Officer in Charge, Gage Lab. Div. & Inspc. Sub-Office, Office of Ch. of Ord.; *for mail*, Quarters 3, Frankford Arsenal, Philadelphia 37, Pa.
Darnell, Joseph Roy ('42) (FKS), Consultant, Legal Dept., Universal Oil Products Co., 310 S. Michigan, Chicago 4; *for mail*, 206 N. Columbia, Naperville, Ill.
Darrah, William Austin ('19) (CKJEBF), Pres., Intercontinental Engrs., Inc., 176 W. Adams St., Chicago 3, Ill.
Dart, Harry Edson ('17) (SE), Asst. Secy., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford 2, Conn.
Dart, Wm. C. (A'07), Chmn. of Bd., R.I. Tool Co., 142 West River St., Providence 1; *for mail*, 16 Stimson Ave., Providence 6, R.I.
Darwin, Dan P. (J'40) (BCS), Design Engr., Westinghouse Elec. Corp., Essington; *for mail*, 5 Mansion Apts., Highland & Stratford Aves., Lansdowne, Pa.
Das, Peter ('26) (SLK), Testing Engr., Stand. Brands, Inc., Peekskill, N.Y.
Dasaro, Nathan Anthony (J'44), 120 Grove St., Brooklyn 21, N.Y.
Dashefsky, George J. ('30; '35) (EBNASR), Prin. Mech. Engr., Head, Vibration Sec. M-96, N.Y. Naval Shipyard, Brooklyn; *for mail*, 15 Purdy Court, Rockville Centre, L.I., N.Y.
Dashiell, John Woolford, Jr. ('44) (LKBJNZ), Tester, Devel. Work, Gulf Research & Devel. Co., Box 2038, Pittsburgh 30; *for mail*, 333 Ashton St., Pittsburgh 7, Pa.
Dashiell, W. W. ('90), Chmn., Bd. of Dirs., N.Y. Lub. Oil Co., 116 Broad St., New York, N.Y.
Daskin, Walter (J'45), 66 Van Winkle Ave., Passaic, N.J.
Dasso, David ('12; '21; '33) (BFRSW), Chmn., Pres., Corporacion Peruana del Santa; *for mail*, Apartado 1233, Lima, Peru, S.A.
Dauber, Clarence Andrew (J'28) (SFEKHD), Supvg. Engr., Plant Design, Cleveland Elec. Illum. Co., 75 Public Sq., Cleveland 1, Ohio.
Dauber, Joseph (J'39) (ZBC), Physicist, V.P. of Scien. Div., Acme Indus. Co., 200 N. Laflin St., Chicago 7; *for mail*, 5119 Kimbark Ave., Chicago 15, Ill.
Daubner, Raymond E. (J'37) (HKS), Davis Engrg. Corp., 1064 E. Grand St., Elizabeth, N.J.
Daudet, Andre Cord (J'45) (AJO), Tool Designer, Dept. 9682, West. Elec. Co., Inc., 787—11th Ave., New York; *for mail*, 309 W. 93rd St., New York 25, N.Y.
Daudt, Louis Rodney (J'41) (BRS), Asst. Project Engr., Dye Works, E. I. du Pont de Nemours & Co., Deepwater, N.J.; *for mail*, 2802 Baynard Blvd., Wilmington 270, Del.
Daugherty, Edgar Sterling ('39) (SKNCFJ), V.P., Secy., Cochrane Corp., 17th St. & Allegheny Ave., Philadelphia 32; *for mail*, 8375 Glen Rd., Elkins Park 17, Pa.
Daugherty, Frank ('09; '25) (DEFHOS), Pres., Engr., Scofield Engrg. Co., 1324 Commercial Trust Bldg., Philadelphia 2, Pa.
Daugherty, Frederick Willard (J'40) (YWMFEC), 4804 Raleigh St., Denver, Colo.
Daugherty, Robert Long ('19; F'36) (HSKEF), *Manager*, '25-'28, *Vice-President*, '28-'30; Prof. Mech. Engrg., Calif. Inst. of Tech., Pasadena 4, Calif.
Daugherty, Samuel Bovard ('05) (EBSHJK), Design Engr., Dir., Natl. Transit Pump & Mch. Co., 19 N. Petroleum St., Oil City, Pa.
Daum, John H. ('39) (BCHLMN), V.P., Sales Mgr., Cincinnati Planer Co., 3120 Forrer Ave., Oakley, Cincinnati 9; *for mail*, 705 Wakefield Dr., Cincinnati 26, Ohio.
Dauphinais, George Arthur (J'45) (BEKAMC), 523 Locust St., Greensburg, Pa.
D'Auria, Armand Rocco (J'42) (SMJCFK), Supvr., Gibbs & Cox, Inc., 21 West St., New York, N.Y.; *for mail*, 141 Broadway, Newark 4, N.J.
Dauwalter, Chadwick Schuyler (J'43) (ABN), Research Analyst, Lockheed Aircraft Corp., Burbank; *for mail*, 1115 Lura St., Pasadena 5, Calif.
Davantzis, James Constantine (J'44) (ACFJN), 553 W. 187th St., New York 33, N.Y.
Davenport, Gordon H. (J'39) (ENSBDO), Lt., U.S.N., Exec. Officer, U.S.S. *Redfin*, Submarine Force, Pacific Fleet; *for mail*, c/o Box 212, Brightwood Sta., Springfield, Mass.
Davenport, Granger ('25; '32; '35) (NJMC), Asst. Ch. Engr., Gould & Eberhardt, Inc., Irvington 11, N.J.
Davenport, James E. ('39), V.P., Charge of Engr., Devel. & Research, Am. Loco. Co., 30 Church St., New York 8, N.Y.
Daves, Howard Winfred (J'45) (BDO), Mech. Engrg. Staff, Charge of Design, Aluminum Co. of Am.; *for mail*, 78 Juniper Ave., Alcoa, Tenn.
Davey, Geoffrey Innes ('43), Cons. Engr., 60 Hunter St., Sydney; *for mail*, "Netherby," Bangalla St., Warrawee, N.S.W., Australia.
Davey, George W. ('25; '35) (OS), Gen. Supt., Riley Stoker Corp., Worcester; *for mail*, 17 Metcalf St., Worcester 5, Mass.
Davey, Peter ('26; '35) (BNZ), Partner, Vibroscope Co., 6 E. 39th St., New York 16, N.Y.
Davey, Richard Smith (J'44) (AEHYKN), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail*, 23-B Elizabeth Rd., Hampton, Va.
Davich, Michael (J'43), Gen. Elec. Co.; *for mail*, 261 Ocean St., Lynn, Mass.
David, Anthony Theodore (J'38) (CMN), Lt. (j.g.), U.S.N.R.; *for mail*, 1595 Unionport Rd., New York 62, N.Y.
David, Ernest Victor ('29) (JLRCKN), Asst. Mgr., Tech. Sales Div., Air Reduction Co., 60 E. 42nd St., New York 17, N.Y.
David, John Kuhn (J'39) (KYDC), Mgr., Appliance Parts Co., 123 E. New York St., Indianapolis 4; *for mail*, 4049 Central Ave., Indianapolis 5, Ind.
Davidoff, Morris (J'41) (ANKYEW), Project Engr., Ranger Aircraft Eng. Div., Fairchild Eng. & Airplane Corp., Farmingdale, L.I.; *for mail*, 2615 Ave. K, Brooklyn 10, N.Y.
Davidson, David J. (J'44) (BCL), Assoc. Engr., Plant Engrg. Dept., Dravo Corp., Pittsburgh 25; *for mail*, 805 Heths St. E.E., Pittsburgh 6, Pa.
Davidson, Edward Henry ('31; '35) (ER), Asst. Dir., Bur. of Loco. Inspc., Interstate Commerce Comm., 12th & Constitution Ave., Washington 25, D.C.
Davidson, Edward Henry, Jr. (J'45) (AHJN), 1445 Ogden St., N.W., Washington 10, D.C.
Davidson, Eugene Neil (J'43) (CZLM), Contract Engr., Republic Flow Meters Co., 2240 Diversey Pkwy., Chicago 47; *for mail*, 1211 Elmdale Ave., Chicago 40, Ill.
Davidson, Gene Elton (J'38) (AEJLNY), 349 Sharon Ave., Glendale, Ohio.
Davidson, James Rollin (J'40) (ZNCM), Project Engr., Taylor Instrument Cos., Box 110, Rochester 1, N.Y.
Davidson, Jesse Irvine ('37; '42) (SFCL), Power Engr., Calco Chem. Div., Am. Cyanamid Co.; *for mail*, 131 Walnut St., Bound Brook, N.J.
Davidson, John Leander ('41) (CMR), V.P., Valve Pilot Corp., Rm. 927, 230 Park Ave., New York; *for mail*, 16 Winthrop Lane, Scarsdale, N.Y.
Davidson, Kenneth S. M. ('24; '31; '35) (HR), Dir., Exper. Towing Tank, Stevens Inst. of Tech., 6th & River Sts., Hoboken, N.J.
Davidson, Maxwell S. ('45) (DFHKLS), Ch. Engr., Am. Crystal Sugar Co., Boston Bldg., Denver 1, Colo.
Davidson, Philip ('37; '38) (CFJMNS), V.P., Ch. Engr., Robinson Engrg. & Devel. Co., Ltd., Box 100; *for mail*, 1623 Senlac St., Calgary, Alta., Can.
Davidson, Rupert Rolston ('30; '40) (JDCF), Capt., Hdq., Army Air Forces Tech. Trng. Comd., Scott Field, Ill.
Davidson, Sidney (J'36) (AC), Pvt., Army Air Forces, Miami Beach, Fla.; *for mail*, 1572 E. 8th St., Brooklyn 30, N.Y.
Davidson, Sidney ('45) (CMD), Supvg. Engr., George S. May Co., 122 E. 42nd St., New York 17; *for mail*, 300 E. 163rd St., New York 56, N.Y.
Davidson, Ward Follet ('27; F'41) (BFSK), Research Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Davidson, William Harold ('20; '25) (SKLAEF), 1600 Walnut St., Philadelphia 3, Pa.
Davies, Albert W. ('30) (CM), Dir., Mavor & Coulson Ltd., 47 Broad St., Mile End, Glasgow, Scotland.
Davies, C. E. ('15; '23; '26) (CM), *Secretary*, A.S.M.E., '34 to date, 29 W. 39th St., New York 18, N.Y.
Davies, Charles ('45) (NKGO), Head, Charles Davies & Associates, 250 E. 43rd St., New York 17; *for mail*, 265 Coligni Ave., New Rochelle, N.Y.
Davies, Henry Eugene, Jr. (J'45), U.S.N.; Box 692, Hyannis, Mass.
Davies, John Griffith (J'39) (NJBCH), Mech. Engr., D. W. Onan & Sons, 43 Royalston Ave., Minneapolis 5; *for mail*, 4123 Wentworth Ave., Minneapolis 8, Minn.
Davies, Ray E. ('38) (CDM), Pres., Indus. Constr. Corp. Ltd., 826 Santa Fe Ave., Los Angeles; *for mail*, Apt. 12, 2012 Huntington Dr., South Pasadena, Calif.
Davies, Robert Edmund (J'40) (BNY), Suspension Engr., B. F. Goodrich Co., 500 S. Main St., Akron 11; *for mail*, 1712 Redwood Ave., Akron 1, Ohio.
Davies, Robert Henry ('45) (ACH), Tech. Serv. Mgr., Parker Appliance Co., 17325 Euclid Ave., Cleveland, Ohio.
Davies, Thomas Llewellyn, Jr. (J'41) (AC), Part Propr., T. L. Davies Jewelry, Falls City, Neb.
Davies, William Michael, Jr. (J'41) (CMJ), V.P., Factory Mgr., Hemphill Co., 131 Clay St., Pawtucket; *for mail*, 199 Walker St., Saylesville, R.I.
Daviet, Claude E. (J'36) (BCKZ), Estimating Engr., Ethyl Corp., Box 341, Baton Rouge 1; *for mail*, 3109 Choctaw Dr., Baton Rouge 7, La.
Davis, A. Sherman, Jr. (J'40) (SFOC), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail*, 560 Chestnut St., Waban, Mass.
Davis, Alton Frank ('43) (CNMO), V.P., Secy., Lincoln Elec. Co., 12818 Coit Rd., Cleveland 1, Ohio.
Davis, Alvan Lamson ('26) (BCJMW), Retired; 25 Concord St., Waterbury 57, Conn.
Davis, Benjamin Haskell ('44) (YNHKB), Sales Engr., John Royle & Sons, 10 Essex St., Paterson 3; *for mail*, 928 Glenview Rd., Ridgewood, N.J.
Davis, Carlton Carlisle, Jr. (J'44) (COSELK), Field Engr., Elliott Co., 225 Broadway, New York 7, N.Y.; *for mail*, 545 Boulevard, Westfield, N.J.

Davis, Cecil Reginald ('24; '38) (ZLDCMJ), Pres., Ch. Engr., Davis Automatic Controls Ltd., 5 Blackmore St., Toronto 5, Ont., Can.
Davis, Charles A. ('33; '38), Pur. Dept. Engr., Caterpillar Tractor Co., Peoria 8; *for mail,* 105 Overland Dr., Peoria 5, Ill.
Davis, Charles Baker ('15) (M), C. B. Davis Engrg. Co.; *for mail,* 801—21st St., Bradenton, Fla.
Davis, Charles H. ('90), Pres., various corporations; Trustee, various estates; Bass River, Mass.
Davis, David Leon (J'39) (EABNMC), Mech. Engr., Replacement Unit Co., 1505 Rockwell St., Cleveland 14; *for mail,* 1496 Rockaway Ave., Lakewood 7, Ohio.
Davis, Delacroix, Jr. (J'41) (HRSJMA), Field Engr., Thompson Pipe & Steel Co., 3001 Larimer, Denver 1; *for mail,* 1402-2nd St., Alamosa, Colo.
Davis, Edgar Lee (J'39) (CFKMS), Ch. Oper., Duke Power Co., Cliffside, N.C.
Davis, Edmund Crain ('33; '44) (FZJ), Combustion Engr., Bethlehem Steel Co., Sparrows Point 19; *for mail,* 2508 Liberty Pkwy., Baltimore 22, Md.
Davis, Edward J. (J'41) (JLS), 326 Lawn Ave., West Lafayette, Ind.
Davis, Eliot Robertson (J'45), Mech. Engr., Canadian Comstock Co. Ltd., 206 Laird Dr., Toronto, Ont., Can.
Davis, Emmett L. ('21; '29) (CDFL), Sales Engr., Combustion Engrg. Co., Inc., 1319 N. Branch St., Chicago 22; *for mail,* 1339 Elmdale Ave., Chicago 40, Ill.
Davis, F. R. ('33; '34; '35) (EFS), Lt. Comdr., U.S.N., Governor's Office, U.S. Naval Home, Philadelphia 46, Pa.
Davis, Floyd Eugene ('44), Designer, Gary Works, Carnegie-Ill. Steel Corp.; *for mail,* 3971 Washington St., Gary, Ind.
Davis, Frank Lawrence, Jr. ('30) (HNBA), Devel. Engr., Watson-Stillman Co., Aldene Rd., Roselle; *for mail,* 60 Arthur Terrace, Kenilworth, N.J.
Davis, Frank Wilson ('44), Ch. Metallurgist, E. B. Badger & Sons Co., 75 Pitts St., Boston 14, Mass.
Davis, Franklin L. ('25) (C), 7114 S.W. Corbett Ave., Portland 1, Ore.
Davis, G. Maslin (J'29) (FKS), Lt. Col., Coast Arty. Corps, U.S.A.; *for mail,* 2005 Evelyn Byrd Rd., Richmond, Va.
Davis, George Elvin (J'39) (EHM), 660 Theines St., El Monte, Calif.
Davis, George Henry ('13), 20 Exchange Pl., New York 5, N.Y.
Davis, Harold R. (J'32) (CDML), Cost Accountant, Indus. Staff, Peat, Marwick, Mitchell & Co., 70 Pine St., New York 5, N.Y.; *for mail,* 359 Prospect St., Ridgewood, N.J.
Davis, Harvey Nathaniel ('20; F'36) (SC), *Manager,* '29-'30, *Vice-President,* '30-'32, *President,* '38; Pres., Stevens Inst. of Tech., Hoboken, N.J.
Davis, Homer Snow (J'25) (CGMD), Circulation Accountant (in Charge), Ill. Publ. & Ptg. Co., 326 W. Madison St., Chicago 6; *for mail,* 1125 Madison St., Evanston, Ill.
Davis, Hunt (J'45) (ABH), Asst. Engr., Elliott Co., Jeannette, Pa.
Davis, Jess Harrison ('40; '45) (SKBFNE), Prof. & Head, Dept. of Mech. Engr., Univ. of Louisville, Belknap Campus, Louisville 8, Ky.
Davis, John C. (J'41) (AER), Lt., 4705 Norwood Ave., Baltimore 7, Md.
Davis, Joseph Dana ('28), Supvg. Engr., Coal Carbonization Sec., U.S. Bur. of Mines, 4800 Forbes St., Pittsburgh; *for mail,* 1307 Macon Ave., Pittsburgh 18, Pa.
Davis, Joseph Hubbard (J'37) (SFKC), 307 W. 30th St., Norfolk 8, Va.
Davis, Kenneth Newton, Jr. (J'46), Ensign, U.S.N.; 11 Atlantic Rd., Swampscott, Mass.
Davis, LeRoy Milton ('37) (SCHB), Pub. Utility Engrg. & Serv. Corp., 231 S. La Salle St., Chicago, Ill.
Davis, Lincoln Kingsbury ('45) (BO), Research Assoc., Underwater Sound Lab., Harvard Univ., Cambridge; *for mail,* 1330 Main St., Brockton 27, Mass.
Davis, Louis E. (J'42) (CMDL), Indus. Engr., West. Elec. Co., Inc., 195 Broadway, New York; *for mail,* 322 W. 104th St., New York 25, N.Y.
Davis, Miss Louise Frederika (J'44) (HNE), Research Asst., Worthington Pump & Mchy. Corp., Worthington Ave., Harrison; *for mail,* Hoxie House, Castle Point, Hoboken, N.J.
Davis, Marion Lee (J'44) (SACFM), Utilities Engr., A.C. Spark Plug Div., Gen. Motors Corp., 1300 N. Dort Highway, Flint 2; *for mail,* 2811 Mallery St., Flint 4, Mich.
Davis, Oscar Allen ('20) (NMLJG), Pres., Davis Engrg. Co., 849 Parkway, Alliance, Ohio.
Davis, Richard F. (J'41) (CMJ), Capt., Ord. Dept., U.S.A., Ord. Dist., Big 4 Bldg., Cincinnati; *for mail,* 33 Oakland Court, Springfield, Ohio.
Davis, Richard F. ('42) (SFC), Ch. Engr., Michigan City Generating Sta., No. Ind. Pub. Serv. Co., Box 318-A, Michigan City, Ind.
Davis, Richard W. (J'41) (ABC), 95 Ben Lomond St., Uniontown, Pa.
Davis, Robert Wheelright, Jr. ('34; A'43) (OMD), Price Supvr., Radio Div., West. Elec. Co., Inc., 120 Broadway, New York 5, N.Y.
Davis, Russell Gardiner ('21; '25; '35) (CAHGML), V.P., Gen. Mgr., Indus. Gear Div., Foote Bros. Gear & Mch. Corp., 5225 S. Western Blvd., Chicago 9; *for mail,* 844 S. Lincoln St., Hinsdale, Ill.
Davis, Spencer Harwood ('43) (DSKBFH), Designer, Philadelphia Elec. Co., Edison Bldg., 900 Sansom St., Philadelphia 5, Pa.
Davis, Thomas J. ('43) (CDFMN), Ch. Engr., Caldwell Plant, Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
Davis, Verner Franklin ('21) (SH), Pres., Atlas Valve Co., 282 South St., Newark 5; *for mail,* 59 Beverly Rd., West Orange, N.J.
Davis, Wells Lawrence (J'37) (ZBN), Asst. Prof., Elec. Engrg. Dept., Ohio State Univ., Columbus 10, Ohio.
Davison, Oliver Simonson (J'43) (EBWZGH), Mar. Serv. Engr., Sperry Gyroscope Co., Inc., 55 Johnson St., Brooklyn 1; *for mail,* 5 Atlantic Ave., East Rockaway, N.Y.
Davison, Robert Elwood (J'43) (ACKJMH), Engr. (Mech. Design), Boeing Aircraft Co., Seattle; *for mail,* 2839 Perkins Lane, Seattle 99, Wash.
Davol, Frank Herbert, Jr. ('16) (MR), Retired; Box 48, Stamford, Conn.
Dawe, William Llewellyn, Jr. (J'45), 2201 Mayfield Ave., Baltimore 13, Md.
Dawes, Herbert N. ('02) (SK), Cons. Engr., in New England, Ehret Magnesia Mfg. Co., 839 Albany St., Boston; *for mail,* 415 Washington St., Brookline 46, Mass.
Dawes, Lyman Miner ('40) (SBANJD), Asst. Prof. Indus. Applications, Mass. Inst. of Tech., Cambridge 39; *for mail,* 104 Cross St., Belmont 78, Mass.
Dawidowicz, William John (J'44) (DJNS), Asst. Stand. Engr., E. W. Bliss Co., 53rd St. & 1st Ave.; *for mail,* 205—19th St., Brooklyn, N.Y.
Dawley, Chester Grant ('40) (CGOTW), Engr.-Appraiser, Reconstruction Finance Corp., 10 Post Office Sq., Boston, Mass.; *for mail,* R.F.D., Allenton, R.I.
Dawley, Clarence A. ('04; '12) (ZBS), Propr., Ch. Engr., N.J. Meter Co., 120 Waynewood Pk.; *for mail,* 1234 Watchung Ave., Plainfield, N.J.
Dawley, Morgan Willis (J'40) (EHNABK), Engr., Engrg. Div., Chrysler Corp., 12800 Oakland Ave.; *for mail,* 12536 Lincoln, Highland Park 3, Mich.
Dawson, A. Ross (J'41) (MCOLAJ), Indus. Engr., Toronto Indus. Comm., 320 Bay St., Toronto, Ont., Can.
Dawson, Albert ('31; '35) (BES), 235 E. Bridge St., Berea, Ohio.
Dawson, Eugene Field ('43) (SEFK), Dir., Sch. of Mech. Engrg., Univ. of Okla., Norman, Okla.
Dawson, Harry W. A. (J'41) (AMCFEO), Flight Lt., Royal Canadian Air Force, O.C. Maint., 1659 Conversion Unit (Heavy Bombers), Overseas; *for mail,* 10 Douglas St., Victoria, B.C., Can.
Dawson, John Edgar (J'43) (BJK), Mech. Engr., Hercules Powder Co., 900 Market St., Wilmington 99, Del.
Dawson, John T. (J'39) (KSFE), Jr. Mech. Engr., J. T. Thorpe & Son, Inc., 941—16th, San Francisco 7; *for mail,* 1430 Arch St., Berkeley 8, Calif.
Dawson, Lewis J. (J'37) (HSL), Field Engr., Cameron Div., Ingersoll-Rand Co.; *for mail,* 130 Frost Ave., Phillipsburg, N.J.
Dawson, Lewis N. (J'39), 4445 Oakmont St., Philadelphia 36, Pa.
Dawson, Melville Deblois ('44) (SFKCNB), Mech. Supt., Union Elec. Co. of Mo., 12th & Locust St., St. Louis; *for mail,* 232 Orrick Lane, Kirkwood 22, Mo.
Dawson, Percy Beech, Jr. ('35; '44) (HAE), Serv. Supvr., Pelton Water Wheel Co., 2929—19th St., San Francisco 10, Calif.
Dawson, William N. (J'40) (KS), Lt. (j.g.), 9501 Lincolnwood Drive, Skokie, Ill.
Day, Anon D. (J'38), Box 429, Pasco, Wash.
Day, Carl Albert ('45), Asst. to Supt., Instrument Div., Bausch & Lomb Optical Co., St. Paul Blvd.; *for mail,* 197 Collingsworth Dr., Rochester, N.Y.
Day, Chas. I. ('10; '25), Pres., Gen. Mgr., W. & L. E. Gurley, 514 Fulton St., Troy, N.Y.
Day, Colin Charles ('31; '35), 167 Buxton Rd., Stockport, England.
Day, Donald Eugene (J'39) (BCDM), Partner, Fred A. Day Co., 300 Riverside Ave.; *for mail,* 60 Brewster Rd., Bristol, Conn.
Day, Edwin Terry ('31) (CMJD), Capt., U.S.N.R., Hull Inspc. Officer, c/o Supvr. of Shipbldg., U.S.N., Bethlehem Steel Co., Quincy; 324 Main St., Hingham, Mass.
Day, Emmett Elbert (J'46) (BJM), Asst. in Mech. Engrg., Mass. Inst. of Tech., Rm. 1-321, 77 Massachusetts Ave., Cambridge 39, Mass.
Day, Harry L. ('28; '35), Supt., Norma-Hoffmann Bearings Corp., Stamford; *for mail,* 9 Ferris Dr., Old Greenwich, Conn.
Day, James Arthur (J'35) (FSKEHD), Results Foreman, N.Y. Power & Light Corp., 126 State St., Albany 1; *for mail,* R.D. 6, 61 Highbridge Rd., Schenectady, N.Y.
Day, Leonard A. ('37), Cons. Engr., 4456 Floriss Pl., St. Louis 15, Mo.
Day, Ralph R. ('20; '35), Engr., Sanderson & Porter, 52 William St., New York, N.Y.
Day, Richard P. (J'46) (DLS), Mchy. Inspr., Sucrest Corp., 280 Richards St.; *for mail,* 1461 E. 7th St., Brooklyn, N.Y.
Day, Richmond A. (J'31) (CDG), Providence Lithograph Co., Providence; *for mail,* 109 Tallman Ave., Cranston 10, R.I.
Day, Wesley Harrison (J'44) (EAH), Lubricants Dept., Shell Oil Co., Inc., 900 United Bank Bldg., 2012 W. 25th St., Cleveland, Ohio.
Dayton, Frank ('28) (DEBS), Design Engr., Mining Dept., Colo. Fuel & Iron Corp.; *for mail,* 1046 Claremont Ave., Pueblo, Colo.
Dayton, Russell Wendt (J'41) (MZJNRA), Asst. Supvr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Deacon, Allin Patterson (J'36) (CDMN), Asst. Supt., Morrow Screw & Nut Co. Ltd.; *for mail,* 246 Wellington St., Ingersoll, Ont., Can.
Deacon, Walter Kinsman (J'42) (AKJBML), Ensign, U.S.N.R., Navy Liaison Office, Calif. Inst. of Technology, Pasadena, Calif.
Deady, Harold E. ('28; '34; '35), 1310 Wesley Ave., Evanston, Ill.
Deal, Edwin Jeremiah (J'43) (AEM), 502 S. Williams St., Columbia, Mo.
Deal, John R. (J'38) (EJL), Assoc. Engr., W. Wirt Young & Associates, 31 Central St.; *for mail,* 33 Bay View Rd., Wellesley 81, Mass.
Deale, Robert Corey ('30) (CMJODL), Ch. Engr., Henry Z. Russell & Sons, 77 W. Washington St., Chicago 2; *for mail,* 1508 Farwell Ave., Chicago 26, Ill.
Dean, Bob Wesley (J'45) (JKZ), Engr. (Design & Layout), Air Conditioning Engrs., Inc., 152 St. Louis St., Mobile 16, Ala.
Dean, Dion K. ('12; '20) (KSHBLY), Ch. Engr., Indus. Div., Foster Wheeler Corp., 165 Broadway, New York, N.Y.; *for mail,* 1245 Pierpont St., Rahway, N.J.
Dean, Edmund Stanley ('30; '35) (AS), English Elec. Co., Ltd., 28 Kingsway, London, W.C. 2, England.
Dean, Edward N. (J'44), Capt., Hdq., 183rd Ord. Bn., A.P.O. 21, c/o Postmaster, New York, N.Y.
Dean, Francis Furman ('30; '37) (SDTHKN), Gp. Engr., Tenn. Eastman Corp.; *for mail,* 1522 Linville St., Kingsport, Tenn.
Dean, Francis H. ('45), V.P., Gen. Engrg. & Mfg. Co., 1523 S. 10th St., St. Louis 4; *for mail,* 6723 Donald St., Normandy 20, Mo.
Dean, H. K. ('27; '35) (SFEK), Engr., Babcock & Wilcox Co., 49 Federal St., Boston 10, Mass.
Dean, Harold Churchill ('27) (C), Cons. Edison Co. of N.Y., 4 Irving Pl., New York 3, N.Y.
Dean, James Ward (J'38) (BHR), 4111 Mayfield Dr., Toledo 12, Ohio.
Dean, Peter Payne ('37) (GN), Owner, Pres., Payne Dean & Co.; *for mail,* Box 271, Madison, Conn.
Dean, Reeve Ward (J'43) (CFJK), Plant Engr., Charge of Maint., Niagara Mch. & Tool Works, 683 Northland Ave., Buffalo 11; *for mail,* R.F.D. 3, E. Main St., East Aurora, N.Y.
Deane, Chester Arthur ('41) (AMC), Devel. Engr., Westinghouse Elec. Corp., Wapok Rd.; *for mail,* Route 4, Allentown Rd., Lima, Ohio.
de Aragón, Orlando Carlos (J'39) (JCM), Ensign, 29th Naval Constr. Bn., F.P.O., San Francisco, Calif.
Dearasaugh, John P. ('29) (ACD), Engr., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh; *for mail,* 110 Youngwood, Pittsburgh 16, Pa.
Dearborn, C. Basil, Jr. (J'42) (JMZDC), Mech. Engr., Sherman & Co., 197 Canal St., New York 13, N.Y.; *for mail,* 30 Engle St., Tenafly, N.J.
Dearborn, Ernest R. ('42) (BDKM), Plant Engr., Angier Corp., Fountain St., Framingham; *for mail,* 299 Union St., Ashland, Mass.
Dearborn, William L. ('92; '17), Retired; Water St., Sandwich, Mass.
Dearden, Brian Burton (J'43) (BCDJM), Works Mgr., Ch. Engr., C. H. Johnson (Mchy.) Ltd., Adswood Rd., Stockport; *for mail,* 23 Thorn Rd., Bramhall, Cheshire, England.
Dearing, E. Richard ('24; '35) (SLDZO), Mech. Engr., Tenn. Coal, Iron & R.R. Co., Ensley Works, Ensley; *for mail,* 5 Pamona Ave., Birmingham 9, Ala.
de Arozarena, Rafael M. ('85), *Address Unknown.*
De Baufre, Wm. L. ('09; '13; F'38) (BKS), Chmn., Dept. of Engrg. Mechanics, Univ. of Neb., Lincoln, Neb.
DeBell, George William ('44) (YMNCO), Cons. Engr., Gurley Bldg., Stamford, Conn.
deBethune, Gaston S. P. ('29) (DHL), 36 Thames St., Newport, R.I.
DeBlois, Lewis Amory ('30) (CBM), Cons. Engr., Treeholme Pk., Chappaqua, N.Y.
Debney, George Charles ('45) (CLSOKM), Asst. to Ch. Engr., Magnolia Petroleum Co., Box 3311, Beaumont, Tex.
De Bodor, Andor ('46) (O), Cons. Engr., Associated Engrg. Co., 22 Lawrence St., Newark; *for mail,* 210 Stonehouse Rd., Glen Ridge, N.J.
De Boe, Bernard James (J'43) (NMJ), Mech. Engr., Aluminum Co. of Am., 3311 Dunn Rd., Detroit, Mich.
DeBoo, Joseph H. (J'36) (JMLDCB), Process Engr., Am. Gear & Mfg. Co., 6633 W. 65th St., Chicago 38; *for mail,* 3222 Park Ave., Brookfield, Ill.
Debski, Theodore Frank (J'34) (ZAG), Lt., Ord. Dept., U.S.A., Ord. Research Center, Aberdeen Proving Ground, Md.
DeBusk, Charles Francis (J'40) (S), Halifax Paper Co., Inc., Roanoke Rapids, N.C.

De Camp, Ray Edward ('42) (HJNOR), Ch. Engr., Devel. & Research Div., Consltd. Steel Corp., 5700 Eastern Ave., Los Angeles 22; *for mail,* 538 N. Jackson St., Glendale 6, Calif.
de Cazenove, Louis Albert ('05; '11), Retired; Theological Seminary, Alexandria, Va.
De Cenzo, Elbert P. (J'35) (KSHLE), Mech. Designer, Giffels & Vallet, Inc., 1000 Marquette Bldg., Detroit 26; *for mail,* 16261 Monica Ave., Detroit 21, Mich.
DeCesare, Roland J. (J'40), 2720 Oakley Ave., Baltimore, Md.
Deck, Albert E. (J'46), Duke Power Co., Cliffside, N.C.
Decker, Clayton Alger ('30; '35) (GMT), Field Engr., SKF Industries, Inc., 1976 Broadway, New York 23, N.Y.; *for mail,* 364 Meadowbrook Ave., Ridgewood, N.J.
Decker, Harold A. ('31; '41) (SEKFJC), Power Devel. Engr., Eastman Kodak Co., Kodak Pk., Rochester 4; *for mail,* 68 Walden Rd., Rochester 10, N.Y.
Decker, Harry L. (J'42) (RBKDZ), Asst. Mech. Engr., Pa. R.R., Rm. 607, Broad St. Sta., Philadelphia; *for mail,* 2218 Bryn Mawr Ave., Ardmore, Pa.
Decker, Herbert Richard ('44) (BLDHJS), Project Engr., A. E. Staley Mfg. Co., Decatur; *for mail,* 1023 W. Prairie St., Decatur 3, Ill.
Decker, Lewis M. ('37) (LS), Technician, Stand. Oil Co. of La., Baton Rouge; *for mail,* 2948 Brady Ave., Baton Rouge 7, La.
Decker, Walton Harris (J'40) (AFGN), Capt., Army Air Forces; *for mail,* c/o R. J. King, 217 S. Eucalyptus, Inglewood, Calif.
de Coriolis, Ernest G. ('21) (AFKZ), Dir. of Research, Surface Combustion Corp., Box 907 M.O., 2375 Dorr St., Toledo, Ohio.
de Crecy, Jacques ('44) (BCELMN), Supvr., Research Div., Smith, Hinchman & Grylls, 59 Pine St., New York, N.Y.
Dedrick, Floyd Frederick ('31; '39) (SJLFDH), Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Deeds, Edw. A. (A'00), *Vice-President,* '21-'23; 20 Exchange Pl., New York, N.Y.
Deegan, J. Wayne (J'35) (CMOD), Plant Indus. Engr. (Chief), Armstrong Cork Co., New Holland Ave.; *for mail,* 1053 Columbia Ave., Lancaster, Pa.
Deemer, Kenneth C. (J'38) (YMHB), Rockwood & Co., 88 Washington Ave., Brooklyn, N.Y.
Deering, George Bradford (J'42) (AFJ), 216 Euclid Ave., East Lynn, Mass.
De Ferranti, Marc A. ('45) (ACD), Indus. Engrg. Div., Gen. Elec. Co., Schenectady, N.Y.
Deffenbaugh, John L., Jr. (J'38) (CEF), Research Engr., Caterpillar Tractor Co., Peoria 8; *for mail,* 512 W. Maywood, Peoria 5, Ill.
de Florez, Luis ('16; '21) (AP), Pres., de Florez Engrg. Co., Inc., 19 Rector St., New York, N.Y.
DeFoe, Jonathan C. (J'40) (FKP), Mech. Engr., White Eagle Div., Socony-Vacuum Oil Co., Inc.; *for mail,* 1223 Euclid St., Augusta, Kan.
De Foreest, Edward T. (J'34) (RFNC), Enginehouse Foreman, Pa. R.R.; *for mail,* 42 N. West End Ave., Lancaster, Pa.
DeForest, C. W. ('22) (FKS), V.P., Columbia Engrg. Corp., 323 Plum St., Cincinnati, Ohio.
De Forest, Merle George ('37) (CEFM), Lt. Col., Corps of Engrs., U.S.A.; *for mail,* 31 Brookside Dr., Greenwich, Conn.
DeFrate, Louis Arthur (J'42) (BAJO), Lt. (j.g.), U.S.N.R., Aero. Research, Moffett Field, Calif.; *for mail,* 708 S. 8th Ave., Bozeman, Mont.
de Fremery, Donald ('18; '26), Union Ice Co., 354 Pine St., San Francisco 4, Calif.
De Garmo, E. Paul ('44) (MCJ), Assoc. Prof. Mech. Engrg., Univ. of Calif., B5 Engrg. Bldg., Berkeley 4, Calif.
Degen, Daniel Joseph (J'42), Leonardsville, N.Y.
Degen, Joseph William ('25; '30; '35) (CDLFS), Admin. Asst., Presbyterian Hospital, 622 W. 168th St., New York 32, N.Y.
Degler, Howard Edward ('27) (ESFK), Prof. Mech. Engrg., Univ. of Tex., Austin 12; *for mail,* 2420 Harris Blvd., Austin 21, Tex.
de Goirigolzarri, Manuel (A'18) (ABEHJK), Surveyor, Havana Div., Am. Bur. of Shipping, 47 Beaver St., New York 4, N.Y.; *for mail,* 72—32nd St., Miramar, Marianao, Havana, Cuba.
DeGray, William Grant (J'44) (BJ), Instr. Welding, Brazilian Air Ministry, Escola Técnica de Aviação, 1316 Rua Visconcle de Parnaiba, São Paulo, Brazil, S.A.
DeHamer, Janus Robert ('35; '42) (MNCZEJ), Asst. Ch. Engr., Shakespeare Co., 241 E. Kalamazoo Ave., Kalamazoo 2; *for mail,* 2332 S. Rose St., Kalamazoo 35, Mich.
De Hart, Harold F. (J'41) (JMB), Jr. Indus. Engr., De Laval Steam Turbine Co., Nottingham Way, Trenton; *for mail,* 19 Fairfield Ave., Dunellen, N.J.
de Haven, Frank Iredell, Jr. (J'37) (AHBMCD), Inspr., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.; *for mail,* 23 Gale Ave., Laconia, N.H.
De Haven, Irvin Clifton, Jr. (J'45) (SEAM), Plant 10, Allison Div., Gen. Motors Corp., Mun. Airport, Indianapolis; *for mail,* 808 E. 42nd St., Indianapolis 5, Ind.
DeHoff, Gerry Berne (J'41) (BFS), Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3, Ill.
DeHuff, Henry ('37) (DLRS), DeHuff & Hopkins, 1505 Race St., Philadelphia 2; *for mail,* 431 Bryn Mawr Ave., Bala-Cynwyd, Pa.
Deily, Arthur T. (J'13) (NBDEZO), Designing Engr., Aluminum Co. of Am., Gulf Bldg., Pittsburgh 19; *for mail,* 5632 Wilkins Ave., Pittsburgh 17, Pa.
Deimel, Richard Francis ('29) (BHZEKA), Prof. Mech. Engrg., Stevens Inst. of Tech., Hoboken, N.J.
Deimer, William H. L. ('46) (BCM), Staff Asst. to Works Mgr., Lunkenheimer Co., Cincinnati; *for mail,* 3942 Lovell Ave., Cheviot 11, Ohio.
Deissler, Robert George (J'43) (JLABYW), Ensign, U.S.N.R., Deck Officer, U.S.S. *Jason,* ARH-1, F.P.O., San Francisco, Calif.
Deist, Herbert Henry (J'40) (BNZAYK), Charge Design, Engrg. Lab., Firestone Tire & Rubber Co., 1200 Firestone Pkwy., Akron 17; *for mail,* 1608 Glenmount Ave., Akron 1, Ohio.
DeJean, Edward Alexander ('44) (BCN), Engr., L. DeJean & Co. Ltd., 1405 St. James St. W., Montreal 3; *for mail,* 4332 Coronation Ave., Montreal 28, Que., Can.
de Jonge, A. E. Richard ('27) (NBJZSO), Research Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
DeJuhasz, Kalman John ('31) (ABEHNZ), Prof. Engrg. Research, Pa. State College; *for mail,* 730 N. Atherton St., State College, Pa.
Delahay, Donald Smith (J'44), Test Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 801 Grant Ave., Schenectady, N.Y.
de la Macorra, Jose, Jr. ('22; '27), Asst. Mgr., Tech. Dir., San Rafael Paper Co., Box 469, Mexico, D.F., Mex.
DeLamar, Brown Bernard (J'42) (ACFOYS), Aviation Electronics Technician's Mate, 3rd Class, Carrier Aircraft Serv. Unit 5, F.P.O., San Francisco, Calif.
Delamater, Roland Eugene ('43) (NMC), V.P., Foote Burt Co., 13000 St. Clair Ave., Cleveland 8; *for mail,* 1247 Quilliams Rd., Cleveland Heights 21, Ohio.
Delaney, Edward Francis (J'40) (SEJ), Apt. 5, 1307 Grove Ave., Richmond, Va.
Delaney, Joseph James (J'38) (MLJ), 133 Farnsworth Ave., Bordentown, N.J.
Delano, Raymond Palmer, Jr. ('31; '44) (RBCN), Mech. Engr., Franklin Ry. Supply Co., Inc., c/o Balmar Corp., Woodberry, Baltimore 11; *for mail,* 4653 Briarclift Rd., Baltimore 29, Md.
de Lapotterie, Harry ('18) (BC), Dist. Mgr. of Mich., Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, Ohio; *for mail,* Wardell Sheraton Hotel, 15 E. Kirby St., Detroit 2, Mich.
DeLaune, Herbert Laurand ('37) (BEKM), J. B. Beaird Co., Inc., Shreveport, La.; *for mail,* 417 Emory, Marshall, Tex.
De Lear, Nicholas Louis (J'43) (ZLA), Engr., King & Shepherd, 50 Church St., New York 7, N.Y.
del Fungo-Giera, Philip L. E. ('46), Mongaup Valley, Sullivan Co., N.Y.
Dell, William H. ('27; '33) (FHS), Steam Engr., Charge Power, Philadelphia Refinery, Gulf Oil Corp., Girard Point; *for mail,* 1723 Johnston St., Philadelphia 45, Pa.
Dellinger, Llewellyn Miller ('45) (BJNO), Ch. Engr., Prod. Mgr., Daunt Corp., 120 Sutton St., Brooklyn 4, N.Y.; *for mail,* 518 Nottingham Rd., Baltimore 29, Md.
Dellis, Paul L. ('46) (COS), Engr., Amitas, Ltd., 50 Broadway, New York 4; *for mail,* 34 E. 50th St., New York 22, N.Y.
Dellplain, Morse ('18), Am. St. Illum. Co., 1500 Walnut St., Philadelphia, Pa.
Del Mar, Bruce Eugene (J'37) (AZBH), *Junior Award,* '45; Mech. Test Engr., Douglas Aircraft Co., Inc., Santa Monica; *for mail,* 11331 Denair St., Los Angeles 24, Calif.
Del Mar, Walter Homer (J'44) (ALM), Tool Engr., Douglas Aircraft Co., Inc., Ocean Park Blvd., Santa Monica; *for mail,* 549 Arbramar St., Pacific Palisades, Calif.
Delmonte, John ('39; '46) (YABLW), Tech. Dir., Plastics Inst., 186 S. Alvarado St., Los Angeles 4; *for mail,* 1018 Geneva St., Glendale 7, Calif.
Delmonte, Julian (J'41), Ensign, 2-V(S), U.S.N.R., 175 Pinehurst Ave., New York, N.Y.
DeLong, Arthur Fred (J'38) (MCDNOJ), Welding Engr., Edw. G. Budd Mfg. Co., 2450 Hunting Park Ave., Philadelphia; *for mail,* 1509 Allen Rd., Oreland, Montgomery Co., Pa.
de Lorenzi, Otto ('20; '22; '25) (FKSECD), Dir. of Education, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
De Lorenzo, Angelo Paul (J'45) (BKAODH), Tech. Supvr., Bruce Engrg. Co., 55 Van Dam St., New York; *for mail,* 1174—78th St., Brooklyn, N.Y.
del Paso, Alejandro ('42) (MNJCSA), Gen. Mgr., Inyecta, S.A., Ave. Progreso 54, Tacubaya; *for mail,* Anahuac 8, Roma Sur, Mexico, D.F., Mex.
De Luca, Anthony P. (J'44), 332 Washington Ave., Dover, N.J.
DeLuca, Ernest (J'36), Lt. (j.g.), U.S.N.R.; 3401 Shoshone, Denver 11, Colo.
De Luca, Joseph Attilio ('44) (CMWY), Safety Engrg., E. I. du Pont de Nemours & Co., Wilmington; *for mail,* 3 Perkins Ave., Silverside Heights, Wilmington 278, Del.
De Luchi, Frank Stephen (J'37) (COHD), Lt. (j.g.), U.S.N.R., Matl. Officer, Office of Asst. to Indus. Mgr., U.S.N., Ferry Bldg., San Francisco; *for mail,* 722 Taylor Ave., Alameda, Calif.
DeMarco, Allen V. (J'41) (BJM), Dir., Bridgeport Testing Lab., 2813 Dixwell Ave., Hamden, Conn.
De Marco, Ralph P. ('39) (SBFHKE), Mech. Designer, Am. Gas & Elec. Serv. Corp., 30 Church St., New York; *for mail,* 2375 Marion Ave., Bronx, N.Y.
Demarest, Laurens McGregor ('43) (O), Owner, L. M. Demarest & Associates, 71 Vanderbilt Ave., New York 17, N.Y.
Demarest, Richard Terhune (J'43) (KMCEFS), Asst. to Gen. Plant Mgr., Fitzgibbons Boiler Co., Inc., Oswego, N.Y.
De Matteo, Adolphe John (J'42) (BYHJ), Devel. Engr., Watson-Stillman Co., Aldene Rd., Roselle; *for mail,* 966 Arlington Ave., Plainfield, N.J.
de Mauriac, William James (J'31) (CJS), Asst. Supt. Maint., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 7; *for mail,* 238 Summit Rd., Springfield, Delaware Co., Pa.
Dement, Marvin Ernest (J'42) (ABM), Administrative Sr. Test Engr., Wright Aero. Corp., Paterson; *for mail,* 289 Highwood Ave., Ridgewood, N.J.
Demer, Louis Joseph (J'39) (BAJFKZ), 404 Church St., Honesdale, Pa.
Demmitt, Francis Hugh (J'42) (ZHMLSE), Naval Ord. & Mar. Serv. Engr., Sperry Gyroscope Co., Inc., 2331—7th Ave., Seattle 1, Wash.
Demorest, George Eugene (J'30) (BNMLZD), Asst. Supt., Charge of Exper. & Devel. Work, Plant 40, Fisher Body Div., Gen. Motors Corp., 1500 Ferry St., Detroit; *for mail,* 4207 Beaconsfield St., Detroit 24, Mich.
Demougeot, Maurice George ('27; '35) (CAM), Plant Mgr., Watson Flagg Mch. Co., 845 E. 25th St., Paterson; *for mail,* 193 E. 4th St., Clifton, N.J.
Dempsey, Michael J. ('16; '25) (BCM), Mech. Supt., Chase Brass & Copper Co.; *for mail,* 60 Country Club Rd., Waterbury, Conn.
Denari, Edward John (J'45) (OESLBK), Mech. Engr., Austin Co., 618 Grand Ave., Oakland 10; *for mail,* 117 Ashton Ave., San Francisco 12, Calif.
DenBleyker, E. Thornton (J'45) (AMHNC), Mech. Engr., Design, Finch Telecommunications, Inc., 4th & Virginia Sts., Passaic; *for mail,* 149—7th Ave., Hawthorne, N.J.
Den Hartog, Jacob Pieter ('29; '35) (B), Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge 39, Mass.
Denig, Fred ('22; '26; '35) (DEFKLW), V.P., Dir. of Research, Koppers Co., Inc., Koppers Bldg., Pittsburgh 19, Pa.
Denise, John V. (J'37), Capt., 234th Engrs. Combat Bn., A.P.O. 230, c/o Postmaster, New York, N.Y.
Denison, Griswold ('18) (SF), Asst. to Mech. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.; *for mail,* 17 North Terrace, Maplewood, N.J.
Deniston, R. Fred (J'39), Box 394, Route 4, East St. Louis, Ill.
Denne, Walter William Forbes ('45), Ch. Engr., Works Mgr., Truth & Sportsman of Australia Ltd., Hoskin House, Hoskin Pl., Sydney, N.S.W., Australia.
Dennis, Basil Wrenn ('11; '18) (ESFKRA), Pres., Md. Diesel Assn., Inc., 19 W. Franklin St.; *for mail,* 2908 Loudon Ave., Baltimore 16, Md.
Dennis, Edwin L. ('37), Cons. Engr., 416 Louisiana Natl. Bank Bldg., Baton Rouge 6, La.
Dennis, Richard Earl (J'38) (CBATFL), Lt., A.P.O. 14983, c/o Postmaster, San Francisco, Calif.; *for mail,* 1001 N. 7th St., Terre Haute, Ind.
Dennison, Edward S. ('24; '32) (EBKH), Dir., Research & Devel., Elliott Co., Jeannette, Pa.
Densen, David A. ('40) (SHLKB), Designer, 124 S. Central Ave., Elmsford, N.Y.
Dent, John Adlum ('11; '16; '25) (SEBKN), Prof. Mech. Engrg., Univ. of Pittsburgh, Pittsburgh 13, Pa.
Dentler, Arnold E. ('41) (BJRY), Test Engr., W. H. Miner, Inc., 2532 S. Rockwell St., Chicago 8, Ill.
Denton, A. Penn ('26) (CEH), Asst. City Engr., City of Kansas City, Mo., 19th Fl., East City Hall, Kansas City, Mo.
Denton, Louis I. ('18), Mech. Engr., Pacific Fruit Express Co., 64 Pine St., San Francisco; *for mail,* 4119 Mountain Blvd., Oakland, Calif.
Denworth, Harry S. (J'40), Lt. (j.g.), U.S.N., S.P.D.C. Naval Supply Depot, Navy 117, F.P.O., New York, N.Y.; *for mail,* 203 Broad St., Mt. Holly, N.J.
Denzler, Rudolph Emil (J'38) (JAZ), Engr., Design & Prod. Work, Paul O. Abbe, Inc., 143 Center Ave., Little Falls; *for mail,* 138 Morgan Pl., North Arlington, N.J.
Depairon, Jean ('29; '35) (BJNMWD), Mech. Engr., Pollard Mfg. Co. Ltd., Ferguson St.; *for mail,* Box 124, Niagara Falls, Ont., Can.
De Pould, Frank (J'40) (CMJDL), *Charles T. Main Award,* '40; Lt. Comdr., U.S.N.R., Admin. Officer, c/o Bur. of Aero. Gen. Rep., Suite 624, 210 W. 7th St., Los Angeles 14, Calif.
Deppeler, John Howard ('15; '23) (ABEJK), Ch. Engr., Metal & Thermit Corp., 120 Broadway, New York 5, N.Y.
Depue, Clark A., III (J'39), Capt., Wright Field, Dayton; *for mail,* 221 Emerson Dr., Osborn, Ohio.
DePuy, Emerson Prouty (J'45), U.S.N.; *for mail,* 2815 Ridge Rd., Des Moines, Iowa.
Derby, Robert Ernest (J'43) (SOF), Mech. Engr. with Albert C. Wood, Cons. Engr., 1411 Walnut St., Philadelphia 2, Pa.

De Remer, Jay Grant ('13) (BMNOS), Cons. Engr., Hollow Tree Ridge Rd., Darien, Conn.
Deringer, Bronaugh Woodland, Jr. (J'36) (EYJ), Engr., Am. Hammered Piston Ring Div., Koppers Co., Inc., Box 626, Baltimore 3; *for mail*, 310 W. Maple Rd., Linthicum Heights, Md.
Derr, Thomas S. ('19; '35) (RSF), Pres., Am. Steam Auto. Co., 25 Kempton Pl., West Newton 65, Mass.
Derrickson, George W. (J'41) (ABS), Box 273, Johns Hopkins Univ., Baltimore, Md.
Derrig, Geo. J. (J'39) (ABE), 1535 Highland Ave., Chicago, Ill.
Derrom, Donald Laird ('43) (CMAJEB), Cons. Engr., Warren McArthur Corp.; *for mail*, Box 66, Bantam, Conn.
Derry, Gardner Clinton ('36) (ACDMRS), V.P., Gen. Mgr., B. F. Sturtevant Co. (Div. of Westinghouse Elec. Corp.), Damon St., Hyde Park, Boston 36, Mass.
De Santis, Fausto George (J'36) (BNMHAR), Ord. Engr., Ship Bldg. Div., Quincy Yard, Bethlehem Steel Co., E. Howard St.; *for mail*, 14 Edinboro Rd., Quincy 69, Mass.
DeSelm, Charles Hilton (J'44), Lt. (j.g.), U.S.N.R., Engr. Officer; *for mail*, 1711 Project Dr., Albuquerque, New Mex.
DeShazer, Grant A. ('44) BNJY), Mech. Engr., Supvr., David Taylor Model Basin, Bur. of Ships, Navy Dept., Washington; *for mail*, 6209 Vorlich Lane, Washington 16, D.C.
De Simone, Lazzaro Pasquale (J'44) (NMJC), Technician, 5th Grade, New Devel. Div., U.S.A., Box 1000, A.P.O. 827, c/o Postmaster, New Orleans, La.
DeSimone, Stephen Joseph (J'39) (CMD), Indus. Engr., Grasselli Chemicals Dept., E. I. du Pont de Nemours & Co., 151st St. & Kennedy Ave., East Chicago; *for mail*, 7601 Pine St., Gary, Ind.
DeSipin, Thomas J. (J'42) (AFS), Lt., Army Air Forces, 138 Maplewood Ave., Kirklyn, Delaware Co., Pa.
Desmon, Leland Golde (J'43) (ASF), Mech. Engr. (Research), Natl. Advis. Com. for Aero., Cleveland Airport; *for mail*, 4649 Grayton Rd., Cleveland 11, Ohio.
DeSpain, Thomas Harry ('44) (BCMNTY), Ch. Engr., So. Textile Mchy. Co., 2nd & Norton Sts., Paducah, Ky.
Destin, Panteleimon Theodor ('36; '38) (BCE JKN), Sr. Mar. Engr., Mchy. Supvr., Navy Dept., Asst. Indus. Manager's Office, Ferry Bldg., San Francisco; *for mail*, 1116 Oxford Rd., Burlingame, Calif.
de Stoppelaar, Lodewyk Paul ('44), Tramweg 3, Aerdenhout, Haarlem, Netherlands.
Detro, Howard Ira ('42) (CNJ), Mgr., Richmond Works, Am. Radiator & Stand. Sanitary Corp., Box 1312, Richmond; *for mail*, 33 Sunset Dr., Berkeley 8, Calif.
Dettloff, Adolph M. ('21; '25; '30) (OCLS), Prof. of Marketing, N.Y. Univ., Washington Sq., New York 3; *for mail*, 255 E. 49th St., New York 17, N.Y.
Detweiler, Abraham L. (J'44) (EJNSW), Devel. Engr., Aircraft & Diesel Equip. Corp., 4401 N. Ravenswood Ave., Chicago; *for mail*, 813 Sherman Ave., Evanston, Ill.
Deuhs, Gerhard Bernard ('42) (CKMOS), Lt. Col., Corps of Engrs., Ft. McArthur, Calif.; *for mail*, Long Prairie, Minn.
Deuster, Ralph William (J'43) (KY), Capt., Hdq., 30th Field Arty. Gp., A.P.O. 403, c/o Postmaster, New York, N.Y.; *for mail*, 3876 Yorkshire Rd., Detroit 24, Mich.
Deutch, Michael Joseph ('42) (CDEFMS), Spec. Asst. to Dir. & Chief, Planning Div., Office of War Mobilization & Reconversion, Rm. 1263, Lafayette Bldg., Washington, D.C.
Deutchman, Julius Max (J'33) (BEJMN), Pres., N & D Mfg. Corp., 10 John St., Yonkers; *for mail*, 1-3 Wellesley Ave., Yonkers 5, N.Y.
Deutsch, Irving N. (J'44) (BLMNYZ), S.A.D., 3rd Class, U.S.N., Engr., Special Devices Div., 610 H St., N.E., Washington, D.C.
Deutsch, John Tener ('42), Allen-Sherman-Hoff Co., 225 S. 15th St., Philadelphia, Pa.
Deutsch, Robert A. ('45) (BHKZ), Asst. Elec. Engr., New York City Tunnel Authority, 10 State St., New York 4; *for mail*, 4672 Broadway, New York 34, N.Y.
Deutsch, Zola Gotthard ('23; '37) (LJMOED), Deutsch & Loonam, Grand Cent. Terminal Bldg., 70 E. 45th St., New York 17, N.Y.
Deutschman, Aaron D. (J'44) (EBSM), Devel. Engr., Schirgun Corp., 136 W. 52nd St., New York; *for mail*, 2160 Bolton St., Bronx 60, N.Y.
Deutschman, Meyer W. (J'37) (SOBKW), Mar. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail*, 1901 Ocean Ave., Brooklyn 30, N.Y.
DeVarona-Aguero, Pablo ('27; '34; '35) (RBF CHS), Asst. Supt., Motive Power & Shops, Consltd. R.R.s of Cuba; *for mail*, Joaquin Aguero 104, Camagüey, Cuba.
Develin, William Jack ('44) (DLY), Mgr., Conveyer Dept., Sandvik Steel, Inc., 111—8th Ave., New York 11, N.Y.
Deverall, Charles N. ('24; '35) (KBJLM), Engr., Niagara Blower Co., 673 Ontario St., Buffalo 7; *for mail*, 71 Highgate Ave., Buffalo 14, N.Y.
Devereaux, William Allan (J'37) (ZSLCF), Sales, Serv. Engr., Bailey Meter Co. Ltd., 307 Royal Bank Chambers; *for mail*, Box 715, Halifax, N.S., Can.
Devereux, Henry Milburn (J'33) (WCMH), Lt. Comdr., U.S.N., Bur. of Ships, Navy Dept.; *for mail*, 1010—16th St., N.W., Washington 25, D.C.
DeVilbiss, Thomas Allen (J'42) (NLYBC), Mech. Devel. Engr., Bakelite Corp., River Rd., Bound Brook; *for mail*, 1003 Park Ave., Plainfield, N.J.
DeVine, Donald Frederick (J'43) (ZKSLJA), Instrumentation Engr., Minneapolis-Honeywell Regulator Co., 122 N.E. Broadway, Portland, Ore.
deVisser, John H. ('14; '18), V.P., Treas., Coon-deVisser Co., 2051 W. Lafayette Blvd., Detroit 16, Mich.
Devlin, Edward John ('20) (LKJCFP), Research Engr., Brooklyn Union Gas Co., 176 Remsen St., Brooklyn 2, N.Y.
DeVoe, John Mixsell (J'32) (CDNL), Supt., U.S. Tobacco Co., 905 Harrison St., Nashville 2; *for mail*, Overhill Dr., Donelson, Tenn.
Devoy, Edmund B. ('22; '35) (CHM), Ch. Engr., Hawaiian Hotels Div., Matson Navigation Co., c/o Moana Hotel, Honolulu 11, T.H.
De Vries, Reinier Paul ('17; '35) (LZSDMJ), Ch. Engr., Revere Sugar Refinery, 333 Medford St., Charlestown 29, Mass.
Dew, Donald H. (J'16) (YCMSDK), Pres., Diemolding Corp., Rasbach St., Canastota, N.Y.
Dewar, Nolan Wilfred (J'43) (SFHEBC), Mech. Engr., Dept. Public Works, Phillip St., Sydney, N.S.W., Australia.
Dewey, Charles A., Jr. (J'39), c/o Duke Power Co., Charlotte, N.C.
Dewey, Raymond E. (J'39) (JKMN), Ch. Engr., Ohio Seamless Tube Co., W. Main St., Shelby, Ohio.
Dewey, William Vernon ('41) (KJLOC), Mgr., Metal Glass Products Co., 605 Washington, Chicago 6, Ill.
De Windt, Arthur Kellogg (J'43) (BEOYKF), 28-25—215th Place, Bayside, N.Y.
DeWitt, Chauncey Osborn ('42), Dist. Engr., Westinghouse Air Brake Co., 3400 Empire State Bldg., New York, N.Y.; *for mail*, 1055 Wilson Ave., Teaneck, N.J.
DeWitt, Edward James ('41) (CHKMN), Exec. V.P., Gen. Mgr., Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.
Dewling, Wm. Littell E. ('28; '35; '35) (BKL), Engr., Solvay Process Co., Hopewell; *for mail*, 1603 Berkeley Ave., Petersburg, Va.
DeWolf, Daniel W. (J'39) (CDL), Ensign, U.S.N.R., 25 Beverly St., Rochester 10, N.Y.
DeWolf, Thaddeus N. (J'42) (BNRY), Asst. Engr., Armour Research Foundation, 35 W. 33rd St., Chicago 16, Ill.
DeWolfe, Edward Charles ('99; A'06) (CGD), Propr., DeWolfe Adv. Serv., 2807 Forest Ave., River Grove, Ill.
Dexter, Albert J. ('20) (DYM), Ch. Draftsman, Engrg. Dept., U.S. Rubber Co., Fisk Tire Plant, Grove St., Chicopee Falls; *for mail*, 21 Montclair St., Springfield 4, Mass.
Dexter, Charles Francis (J'36) (FLS), Inspg. Engr., Power Dept., Boston Elev. Ry. Co., 536 Harrison Ave., Boston; *for mail*, 95 Lincoln St., Norwood, Mass.
Dexter, Gregory Mumford ('19) (CMOLSD), Cons. Indus. Engr., 32 Fenimore Rd., Scarsdale, N.Y.
Dexter, Harris E. ('17; '35) (C), V.P., Charge Commercial Relations, Cent. Hudson Gas & Elec. Corp., 50 Market St., Poughkeepsie, N.Y.
Dexter, Howard Walker, Jr. ('24; '46) (SEB LZ), Dir., Power & Steam Utilization, Investigation Div., Indus. Sales Dept., Duquesne Light Co., 6th Fl., 435—6th Ave., Pittsburgh 19, Pa.
Dexter, John Lindsay (J'41) (CGT), Research Dir., Nashua Mfg. Co., 48 Franklin St.; *for mail*, 137 Marlboro St., Boston, Mass.
deZafra, Carlos ('46) (CMO), Indus. Consultant, Corporacion de Fomento (Chile), Rm. 2623, 120 Broadway, New York 5, N.Y.
Dezeeuw, William Joseph (J'42), 1120 Bryant Ave., New York 59, N.Y.
DeZubay, Egon Alex (J'43) (SBA), Research Engr., Westinghouse Elec. Corp., Ardmore Blvd., East Pittsburgh; *for mail*, 74 N. Fremont Ave., Bellevue 2, Pa.
DiAddario, Alexander Nicodom ('32; '41) (CYM ADH), Pres., Engr., Office Mgr., Molds & Mfg., Inc., R.F.D. 5, Union Rd., Buffalo 6; *for mail*, 511 Northland Ave., Buffalo 11, N.Y.
Diakoff, Alexis John ('37) (ESBKNR), Cons. Engr., Am. Loco. Co., Schenectady 5; *for mail*, St. David's Lane, Schenectady 8, N.Y.
Diamant, Sidney ('17) (CJNAMD), Pres., Gen. Mgr., Diamant Tool, Mch. & Mfg. Co., Inc., 401 Mulberry St., Newark 2, N.J.; *for mail*, 666 West End Ave., New York 25, N.Y.
Diament, William Alexander ('45) (ZHCLAM), V.P., Field Mgr., Fischer & Porter Co., County Line Rd., Hatboro; *for mail*, 400 Runnymede Ave., Jenkintown, Pa.
Diamond, Arthur Richard (J'43) (CNDBJA), Design Engr., Edw. G. Budd Mfg. Co., 25th St. & Hunting Pk. Ave., Philadelphia 32; *for mail*, Greene Manor, Johnson & Greene Sts., Philadelphia 44, Pa.
Diaz-Compain, Jerónimo ('25; '32) (FKLSWZ), Ch. Engr., Central Senado, Senado, Camagüey, Cuba.
Dibble, Charles Gordon (J'42) (AHKSEF), 120 E. Palm Ave., Redlands, Calif.
Dibble, Clarence Heck ('45) (NMCOZB), Systems Engr., Elec. Research Products Div., West. Elec. Co., Inc., 6601 Romaine St., Los Angeles 38, Calif.
Dibble, Edward Fitzgerald (J'38) (ACH), 120 E. Palm Ave., Redlands, Calif.
Dibble, Frank Burton (J'43) (BNJZA), Mech. Engr., Gen. Precision Lab., Inc., 63 Bedford Rd.; *for mail*, Box 141, Pleasantville, N.Y.
Dibble, Harold L. (J'37) (GJM), Indus. Salesman, Dibble Hardware, 270-2 W. Ferry St.; *for mail*, 459 Ashland St., Buffalo, N.Y.
Di Cesare, Fred P. (J'40), F. P. Di Cesare Co., 142 S. 5th St., Steubenville, Ohio.
Dichtel, Robert Norman (J'44) (JKS), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.; *for mail*, Stanley Club, Pittsfield, Mass.
Dick, Bertham Gale ('43) (BCDJMS), Chief of Inspc. Unit, Dept. of Int., Bonneville Power Admin., 1300 N.E. Union Ave., Portland; *for mail*, 8205 N. Smith St., Portland 3, Ore.
Dick, Howard Douglas (J'39) (CMS), Mech. Engr., C. A. Parsons of Can. Ltd., 709 Royal Bank Bldg.; *for mail*, 773 Windermere Ave., Toronto 9, Ont., Can.
Dicke, Allen August ('45) (HJM), Pat. Atty., 120 Broadway, New York 5, N.Y.
Dickens, Charles Garth (J'43) (LMJ), Asst. Application Engr., Westinghouse Elec. Corp., 1500 Braddock Ave., East Pittsburgh, Pa.; *for mail*, 4104—6th Ave., Tacoma 6, Wash.
Dickerman, William Carter, Jr. ('45) (CMDN), East Hampton, N.Y.
Dickerson, Harold Shields ('12) (ECDJML), Mgr., Indus. Oil Dept., Pure Oil Co., E. Broad St., Columbus 15; *for mail*, 2319 Arlington Ave., Columbus 8, Ohio.
Dickerson, Kenneth John (J'33) (AJW), Warrant Officer (j.g.), U.S.A.; *for mail*, 10 D St., Sheridan Parkside, Tonawanda, N.Y.
Dickey, Arthur Jay ('15; '35) (CKMLFB), V.P., Gen. Mgr., C. A. Dunham Co. Ltd., 1523 Davenport Rd., Toronto 4; *for mail*, 9 Mossom Pl., Toronto 3, Ont., Can.
Dickey, Paul S. ('28; '35; '35) (SZFRMC), Ch. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio.
Dickey, Samuel J. ('26) (EF), Pres., Gen. Petroleum Corp. of Calif., 108 W. 2nd St., Los Angeles 12, Calif.
Dickey, Thomas A. (J'41), Oxford, Pa.
Dickey, William Kyle (J'43) (NMJBCK), Engr., Design Sec., Turbine Div., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, 15 Vinnin Sq., Salem, Mass.
Dickie, Alexander Jack ('20; '25; F'36) (DSEA HN), *Manager*, '31-'34; Editor, *Pacific Marine Review*, J. S. Hines Publ. Co., 500 Sansome St., San Francisco 11, Calif.
Dickins, Franque A. (M'43) (KFOMEC), Engr., 168 Van Brunt St., Brooklyn, N.Y.; *for mail*, Craighmont Hall, Boonton, N.J.
Dickinson, Don Manuel (J'44) (SO), 1st Lt., Corps of Engrs., Commanding Officer, 2938th Engr. Refrig. Maint. Det.; *for mail*, 1214 W. Princess St., York, Pa.
Dickinson, Geo. S. (A'30) (F), Berwind White Coal Min. Co., 1 Broadway, New York, N.Y.
Dickinson, Hobart Cutler ('19) (KEAFB), Retired; 4629—30th St., N.W., Washington 8, D.C.
Dickinson, John D. (A'42) (RCMJS), Asst. to Pres., Lima Loco. Works, Inc., 60 E. 42nd St., New York 17, N.Y.
Dickinson, Reginald W. (42) (CDMNAJ), Mgr., Mfg. Planning Dept., Lockheed Aircraft Corp., Burbank; *for mail*, 5109—2nd Ave., Los Angeles 43, Calif.
Dickinson, William Andrew (J'39) (CDHJM), Supt., Extrusion Dept., Internatl. Nickel Co., Inc.; *for mail*, 1030—9th St., Huntington 1, W. Va.
Dickinson, William Noble ('35) (D), Plastics Engr., Supply Mfg. Co., Inc., 34 University Pl., New York; *for mail*, 41 Woodland Ave., Rockville Centre, L.I. N.Y.
Dicks, Norman Allan ('45) (BNZYJD), Mech. Engr., Libbey-Owens-Ford Glass Co., 1701 E. Broadway, Toledo; *for mail*, 2648 Goddard Rd., Toledo 6, Ohio.
Dickson, Charles H. ('16; '26) (CDMLS), 248 Reynolds Terrace, Orange, N.J.
Dickson, Donald R. (J'41) (AKMEFC), Student Engr., Am. Blower Corp., Detroit 32, Mich.; *for mail*, 44 E. 37th St., Indianapolis 5, Ind.
Dickson, John ('42), Cons. Mech. Engr., 716 Times Bldg., Portland 4, Ore.
Dickson, Richard Paul (J'40) (CM), Capt., 126th Ord. Maint. Bn., A.P.O. 254, c/o Postmaster, New York, N.Y.
Dickson, Robert Barnes (A'38) (JFODMK), Pres., Kewanee Boiler Corp., Kewanee, Ill.
Dicmas, John Leonard (J'38) (HC), Sales Engr., Westco Pump Sales Corp., 2315 E. 8th St., Los Angeles 21; *for mail*, 133 Hampden Terrace, Alhambra, Calif.
Di Donno, Peter A. (J'37) (CDM), Mch. & Tool Designer, Shelton Tack & Tubular Rivet Co., Canal St., Shelton; *for mail*, 37 Ridge Rd., Hamden, Conn.
Didriksen, Herluff ('26; '33; '35) (SKB), Designer, Gibbs & Hill, Inc., Pa. Sta., New York, N.Y.; *for mail*, N. Mountain Ave., Bound Brook, N.J.
Dieckman, Clifford R. (J'40), 2419 Howard Ave., San Diego 4, Calif.
Diefendorf, D. W. (J'30), V.P., Gen. Mgr., Diefendorf Gear Corp., 920 W. Belden Ave.; *for mail*, 203 Summit Ave., Syracuse, N.Y.

Diefenthal, Stanley M. (J'39) (BJM), Mgr., Pur. Agt., So. Scrap Matl. Co., 4163 Bienville St.; *for mail,* 7030 Apricot St., New Orleans, La.

Dieffenbach, Ezra C. ('30; '32; '35), 2636 E. 14th St., Sheepshead Bay, Brooklyn, N.Y.

Diehl, Herman, Jr. (J'39) (CDM), Engr., Stevenson, Jordan & Harrison, 19 W. 44th St., New York, N.Y.; *for mail,* 49 Chapel St., East Hartford, Conn.

Diepenbrock, Joseph B. (J'41), Mech. Engr., Compressor Engrg. Dept., Ingersoll-Rand Co., 11 Broadway, New York, N.Y.; *for mail,* 38 Hawthorne Ave., Glen Ridge, N.J.

Dierckx, Jules (A'21), 136 W. 16th St., New York, N.Y.

Dierdorf, Claude C. (J'36) (CJM), 329 Robton St., Indianapolis 8, Ind.

Diescher, Samuel E. ('15) (OJMABN), Propr., S. Diescher & Sons, 1503 Farmers Bank Bldg., Pittsburgh 22, Pa.

Dieter, Frederick Alfred (J'39) (LOCS), Asst. Plant Engr., Am. Cyanamid Co.; *for mail,* 96 Forest St., Stamford, Conn.

Dieter, William ('42) (ACY), Retired; 67 Renner Ave., Newark 8, N.J.

Dietrichson, William Fredrik ('19) (BR), Gen. Mech. Engr., Am. Car & Fdy. Co., 9th & Oak Sts., Berwick 6, Pa.

Dietz, C. Raymond (J'43) (ABW), Radio Technician, 3rd Class, Co. 18-234, Naval Trng. Sch. (Radio Materiel), Navy Pier, Chicago, Ill.

Dietz, Carl Frederick ('03; '10) (D), Pres., Lamson Corp., Syracuse 1, N.Y.; Pres., N.Y. Mail & Newspaper Trans. Co., 101 Park Ave., New York 17, N.Y.; Pres., Boston Pneumatic Transit Co., 211 Congress St., Boston, Mass.

Dietz, William Francis ('45), Asst. Mgr., Westinghouse Elec. Corp., 1625 K St., N.W., Washington, D.C.; *for mail,* Broxburn Dr., Bannockburn, Bethesda, Md.

Dietzgen, Joseph E. (J'41) (CJM), Engr., Eugene Dietzgen Co., 954 W. Fullerton Ave., Chicago; *for mail,* 6937 S. Crandon Ave., Chicago 49, Ill.

Dievers, Grover Edwin ('27; '34; '35) (NBTZE), Layout Engr., Designing, Baldwin Southwark Div., Baldwin Loco. Works, Eddystone; *for mail,* 1426 Powell St., Norristown, Pa.

Di Federico, Mario Anthony (J'44) (ANCB), Flight Engr., Goodyear Aircraft Corp., 1210 Massilon Rd., Akron 15; *for mail,* 1254 N. Howard St., Akron 10, Ohio.

Digan, Thomas Joseph, Jr. (J'41) (CMJRSA), 257 S. Van Pelt St., Philadelphia, Pa.

Digges, Hampson Skinker (J'44), Ensign, U.S.N.R.; Apt. B 527, 2800 Terrace Rd., S.E., Washington, D.C.

Diggs, Donald Roger (J'44) (ACKMRS), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland, Ohio; *for mail,* 2130 Central Park Ave., Evanston, Ill.

Dignan, George Edward ('24) (CLSK), Gen. Mgr., Pittsburgh Coke & Chem. Co., Neville Island, Pittsburgh 25; *for mail,* 27 Priscilla Lane, Carnegie, Pa.

Di Ilio, Charles Carmen ('45) (ABKES), Asst. Prof., Dept. of Mech. Engrg., Pa. State College; *for mail,* Box 178, State College, Pa.

Dike, Melville Arthur (J'45) (NHBCJK), Partner, Dike-Polzin Co., 33 W. Union St., Pasadena 1; *for mail,* 734 W. Foothill Blvd., Monrovia, Calif.

Dilcher, Henry Stanford ('21; '35) (CMJSE), Comdr., U.S.N.R., Inspr. of Naval Matl., U.S.N., Detroit Dist., Free Press Bldg., Detroit 26, Mich.

Dilg, Walter C. (J'36) (ABR), Engr., Waugh Equip. Co., 420 Lexington Ave., New York 17; *for mail,* 42 Trinity Pl., Hewlett, N.Y.

Dill, Richard S. ('38) (KFS), Chief, Heat Transfer Sec., Natl. Bur. of Stands., Connecticut Ave. at Van Ness St., Washington 25, D.C.

Dill, Robert Peter (J'43) (ANMBCH), Design Engr., Douglas Aircraft Co., Santa Monica; *for mail,* 4833 Faculty Ave., Long Beach 8, Calif.

Dillaway, Robert Beachum (J'45) (AHJ), Test Engr., Devel. Dept., Carrier Corp.; *for mail,* 201 Berkeley Dr., Syracuse, N.Y.

Dillon, Edward Lucius ('40) (EHS), Cons. Engr., 2203 Carleton, Ft. Worth 7, Tex.

Dillon, Frank H. (J'24), Cons. Engr., 28 Gale St., Malden, Mass.

Dillon, Lewis Emmett (J'43) (ABHKFZ), Ensign, A-(V)S, U.S.N.R., Assembly & Repair, Structures, Naval Air Sta., Glenview, Ill.

Dillon, Sydney ('22) (JOC), Engr., U.S. Steel Corp. of Del., 436—7th Ave., Pittsburgh 30; *for mail,* 4609 Bayard St., Pittsburgh 13, Pa.

Di Lorenzo, Frank John (J'44) (YASBNJ), Technician, 5th Grade, G-3 Div., U.S.F.E.T. (Main), A.P.O. 757, c/o Postmaster, New York, N.Y.; *for mail,* 400 Columbia Ave., Cliffside Park, N.J.

Dimberg, Paul C. ('23), Asst. Engr., Steam Turbine Blading, Allis-Chalmers Mfg. Co., West Allis; *for mail,* 6525 Well St., Wauwatosa, Wis.

Dimm, William (J'44) (CWNMJB), Product Engr., Sperry Gyroscope Co., Inc., Great Neck, L.I.; *for mail,* 1409 Lincoln Pl., Brooklyn 13, N.Y.

Dimmick, Henry Stuart ('23; '35; '35) (CJLHTN), Dist. Mgr., SKF Industries, Inc., 3701 N. Broad St., Philadelphia 40; *for mail,* 341 Kathmere Rd., Havertown, Pa.

Di Monico, George S. (J'42), Mech. Designer, Indus. Research Bur., 24 W. 40th St., New York; *for mail,* 138 Nelson St., Brooklyn, N.Y.

Dines, Jack Elwood (J'43) (JMB), Designing Engr., Texassteel Mfg. Co., 3909 Hemphill, Ft. Worth 4; *for mail,* 2928 Jennings Ave., Ft. Worth 3, Tex.

Dinger, Henry C. (24), N.Y. Yacht Club, 37 W. 44th St., New York, N.Y.

Dinges, Rawlston Melvin ('44) (C), V.P., Continental Supply Co., 30 Rockefeller Plaza, New York 20, N.Y.

Dingus, George William (J'45) (KMS), Intermediate Engr., Freeport Sulphur Co., Box A; *for mail,* Tarpon Inn, Freeport, Tex.

Dinsmore, James Egan (J'40), Layout & Design Draftsman, Auto Flight Instrument Div., G. M. Giannini Co., 285 W. Colorado, Pasadena; *for mail,* 820½ Maltman Ave., Los Angeles 26, Calif.

Dinsmore, John Robert (J'45) (SBEJHK), Lt. Comdr., U.S.N., Asst. Repair Supt., N.Y. Naval Shipyard, Brooklyn 1; *for mail,* 475 E. 26th St., Brooklyn 26, N.Y.

Diodati, Giacomo Francis (J'45) (SM), 34 Bellevue Ave., Haverhill, Mass.

Dion, Frederic Eugene, Jr. (J'37) (CLMKND), Cashier, Dion's Men's Store, 111 Merrill Ave; *for mail,* 803 Meade Ave., Glendive, Mont.

DiPilla, Edward (J'46), 1621 E. 55th St., Chicago 15, Ill.

Di Prima, Anthony (J'43) (MOC), Designer, Sperry Gyroscope Co., Clinton Rd., Garden City; *for mail,* 272 Schaeffer St., Brooklyn, N.Y.

Diringer, Owen (J'45), 68-36—108th St., Forest Hills, N.Y.

Dirks, Henry B. ('07; '16; '30) (SKFBC), Dean of Engrg., Mich. State College; *for mail,* 637 Grove St., East Lansing, Mich.

DiSanto, Bartel John ('26; '36) (CFSJDL), Asst. Plant Mgr., Am. Smelting & Refining Co., State St., Barber; *for mail,* 764 Garden St., Rahway, N.J.

Dischinger, Harry R. (J'41), Jr. Engr., Am. Can Co., Boston & Hudson Sts.; *for mail,* 1407 Mt. Royal Ave., Baltimore, Md.

Diserens, Paul ('08; '16; F'43), Cons. Engr., Dir. of Research, Worthington Pump & Mchy. Corp., Harrison, N.J.

Dishington, Herman ('35; '41) (CDL), 20 Terrace Walk, Berkeley 7, Calif.

Disston, William Dunlop ('16; '21; '35) (CDMW), V.P., Henry Disston & Sons, Inc., Tacony, Philadelphia 35, Pa.

Ditmars, Walter E. ('29; '35), 230 Park Ave., New York, N.Y.

Ditson, J. D. (J'37) (BJAZ), Engr., Rock Drill Div., Ingersoll-Rand Co.; *for mail,* 889 Gates St., Phillipsburg, N.J.

Dival, Lawrence A. ('45) (LRS), Box 44, South Whitley, Ind.

Divan, Louis S. ('30; '36) (CBRMDN), Asst. to Works Mgr., Baldwin Loco. Works, Paschall Sta., Philadelphia 42; *for mail,* 1931 Franklin Ave., Morton, Pa.

Diver, Mortimer Levering ('22) (BKHFSA), Cons. Engr., Box 1016, San Antonio 6, Tex.

Dixon, Charles Francis ('03; '11) (SFEOKM), Supvg. Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5; *for mail,* Brookside Ave., Wayne, Pa.

Dixon, Edgar O. ('40) (JCSRBE), Ch. Metallurgist & Mech. Engr., Ladish Drop Forge Co., Cudahy; *for mail,* 2323 Newberry Blvd., Milwaukee 11, Wis.

Dixon, John Edward ('42), Pres., Lima Loco. Works, Inc., 60 E. 42nd St., New York 17, N.Y.

Dixon, John Kerner (J'46), Asst. Engr., Allis-Chalmers Mfg. Co., West Allis; *for mail,* 2323 E. Newberry Blvd., Milwaukee 11, Wis.

Dixon, Leon Snell ('21) (OMLCDE), Route 2, Bangor, Me.

Dixon, William (J'40) (AL), Engr., Singer Mfg. Co., Elizabethport; *for mail,* 1015 Central Ave., Plainfield, N.J.

Dixon, William Albright (J'39), Mech. Engr., U.S. Engr. Office, 428 Federal Bldg., Milwaukee 1, Wis.

Dixon, Winfield Eugene (J'45), U.S. Navy; 1617 E. 4th St., Santa Ana, Calif.

Dmitrieff, Boris A. ('37), Asst. Engr., Mech. Engrg. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.

Dmitroff, George A. (J'46), Matls. & Stands. Engr., Sikorsky Aircraft Div., United Aircraft Corp., South Ave., Bridgeport; *for mail,* Meadowside Rd., Milford, Conn.

Doan, Theodore Heberton ('28) (CNMHB), Pres., Foote-Burt Co., 13000 St. Clair Ave., Cleveland 8, Ohio.

Dobbins, Robert Newton ('39; '44) (ACJN), Lt. Comdr., U.S.N.R., 31 S. Prospect St., Verona, N.J.

Dobbs, John Francis ('43) (FJR), Ch. Chemist, N.Y. Air Brake Co.; *for mail,* 213 Stone St., Watertown, N.Y.

Dobkin, Herbert (J'40) (JAMO), Matls. Engr., M.W. Kellogg Co., Jersey City; *for mail,* 5 The Village, Magie Ave., Elizabeth, N.J.

Doble, William A. ('37) (ABCDMN), Cons. & Pat. Engrg., 1000 Mason St., San Francisco, Calif.

Doble, Warren ('37) (BAKN), Dept. Mgr., Aire-search Mfg. Co., 9851 Sepulveda Blvd., Los Angeles 45; *for mail,* 4726 Placidia Ave., North Hollywood, Calif.

Dobrolet, Miss Alexandra (J'39) (CMJZA), Mgr. of Planning, Plant 2, Fed. Products Corp., Eddy St.; *for mail,* 33 Darrow St., Pawtucket, R.I.

Dobrow, Harry Paul (J'37) (ABCMNO), Engr., Chassis Engrg. Div., Willys-Overland Motors, Inc., Wolcott Bldg., Toledo 1, Ohio; *for mail,* 14903 Sussex St., Detroit 27, Mich.

Dobson, John ('23) (EFJKS), Ch. Engr., Power Plant, Am. Molasses Co., 280 Richards St., Brooklyn; *for mail,* 6728 Ridge Blvd., Brooklyn 20, N.Y.

Dobson, John Gordon (J'39) (HKLS), Engr., Wallace & Tiernan Co., Inc., Box 178, Newark 1; *for mail,* Box 977, Short Hills, N.J.

Dobson, Wilson James (J'41), 258 Henry St., Manchester, Conn.

Dobyne, Stevenson Adam ('15; '35) (BCMN), Pres., Champion Distributing Corp., 3638 Olive St., St. Louis 8; *for mail,* 8306 Jackson Ave., Vinita Park, St. Louis 14, Mo.

Dockstader, Ernest Kneale ('34; '45) (CLS), Grasselli Chemicals Dept., E. I. du Pont de Nemours & Co., Philadelphia; *for mail,* 4027 School Lane, Drexel Hill, Pa.

Dockstader, John Hollis (J'41) (CMNJ), Design Engr., Continental Can Co., Inc., 70 Park St.; *for mail,* 131 Grove St., Rutland, Vt.

Dodd, George Vernon (J'45) (NMLBCJ), Jr. Mech. Engr., Kellogg Co., Porter St.; *for mail,* 40 James St., Battle Creek, Mich.

Dodd, John Akin ('24; '30) (SHKFEZ), Propr., John A. Dodd Co., 299 Techwood Dr., N.W., Atlanta, Ga.

Dodd, Samuel Thompson, Jr. (A'46) (ACD), Coordinator, Work Simplification, Lockheed Aircraft Corp., Burbank; *for mail,* 2172 N. Argyle Ave., Los Angeles 28, Calif.

Dodds, Robert Henry ('37) (GTNB), Mch. Designer, Sonoco Products Co., Willow St., Mystic, Conn.

Dodds, Robert Pierson ('37) (BSK), Squad Foreman, Koppers Co., Pittsburgh; *for mail,* 1330 Wood St., Wilkinsburg 21, Pa.

Dodds, William Clarence (J'40) (AEM), Capt., Army Air Forces, Flying Instr., Garden City Army Air Field; *for mail,* 212 Walnut, Garden City, Kan.

Doderer, Adolf W. ('38) (JFEH), Tool Engr., Globe Union, Inc., 900 E. Keefe Ave., Milwaukee 1; *for mail,* 4392 N. Wilson Dr., Milwaukee 11, Wis.

Dodge, Cleon C. (J'37) (JCAM), Serv. Engr., Taylor-Winfield Corp., New York; *for mail,* East Hampton, L.I., N.Y.

Dodge, Conant (J'37), Seaman, 1st Class, Radio Technician, U.S.N.; *for mail,* Kimberton, Pa.

Dodge, Eldon Raymond ('44) (BHJ), Hyd. Research Engr., Fairbanks, Morse & Co.; *for mail,* 727 Milwaukee Rd., Beloit, Wis.

Dodge, Kern ('02; '12) (ADMO), Cons. Engr., 225 S. 15th St., Phildelaphia 2, Pa.

Dodge, Robert Mapes (J'44) (D), Dist. Engr., Link-Belt Co., 2045 W. Hunting Park Ave., Philadelphia 40; *for mail,* Plymouth Meeting, Pa.

Dodge, Robert Thompson (J'44) (COJKBH), Asst. to Dist. Engr., Manhattan Dist., Corps of Engrs., Box E, Oak Ridge, Tenn.; *for mail,* 1220 S. St. Andrews St., Los Angeles 6, Calif.

Dodge, William Waldo, Jr. ('45) (OTN), Secy., Treas., Six Associates, Inc., 15 Church St.; *for mail,* 20 Woodlink Rd., Asheville, N.C.

Dodson, James Raymond, Jr. (J'39) (OLKAHS), Lt. (j.g.), U.S.N.R., Gunnery Officer, Topside, Aviation Patrol Bomber Sqd. 2, Advanced Trng. Unit 3, Box 2, Naval Air Sta., Banana River, Fla.; *for mail,* 2404 Pecan St., Texarkana, Ark.

Dodson, Weston ('44), Engr., Wilberding Co., Inc., 1822 Eye St., Washington, D.C.; *for mail,* 4657 S. 34th St., Arlington, Va.

Doering, John (J'37) (MJCD), Plant Engr., J. S. Thorn Co., 20th St. & Allegheny Ave., Philadelphia 32, Pa.

Doerr, Carl Frederick (J'38) (BHKNOS), Draftsman, Sargent & Lundy, 140 S. Dearborn, Chicago 3, Ill.

Doerr, Norman Edwin (J'40) (SOCJFL), Asst. Mech. Engr., Commonwealth Edison Co., 72 W. Adams St., Chicago 90; *for mail,* 1400 N. 12th Ave., Melrose Park, Ill.

Doersam, Charles Henry, Jr. (J'42) (AZKMNS), Mech. Engr., P-4, Naval Research Lab., S.S.S., Washington 20, D.C.; *for mail,* 7 Buckingham Rd., Palisade, N.J.

Doggett, Fred Fobes (J'43) (EA), 2707 Kane St., Highland Gardens, Chester, Pa.

Doggett, John, Jr. ('36; '45) (CJM), Mech. Engr., Gray Tool Co., 6102 Harrisburg St., Houston; *for mail,* 1120 Prospect St., Houston 4, Tex.

Doherty, Arthur W. (J'41) (A), Ensign, Sqd. 11-A, Corpus Christi, Tex.

Doherty, Robert Ernest (F'41), Pres., Carnegie Inst. of Tech., Schenley Park, Pittsburgh 13, Pa.

Dohn, Charles Walter (J'46) (CDO), Sr. Mar. Draftsman, Basic Design Dept., George G. Sharp, Naval Arch., 30 Church St., New York; *for mail,* 57-03 Catalpa Ave., Brooklyn 27, N.Y.

Dohrenwend, Clayton Oliver ('39; '44) (BJZANW), Dir., Dept. of Mechanics, Ill. Inst. of Tech., Technology Center, Chicago 16, Ill.

Dohrmann, Henry Charles (J'36), Ch. Engr., Buell Engrg. Co., Inc., 70 Pine St., New York, N.Y.; *for mail,* 101 Sherman Pl., Jersey City, N.J.

Doig, Godfrey Davidson ('27; '35) (EFS), Asst. Engr., Cunard White Star S.S. Co., Pier 56, North River; *for mail,* 201 E. 33rd St., New York, N.Y.

Doke, Ernest George (J'37) (RCJ), 1025 Willow Rd., Winnetka, Ill.
Doke, Geo. E. ('19; '21) (JR), Retired; 90 Caryl Ave., Yonkers 5, N.Y.
Dolan, Charles H., II ('30) (ABCM), V.P., Gen. Mgr., Allied Engrg. Corp., 294 Elm St., New Haven 11; *for mail,* 7 Grace St., Old Greenwich, Conn.
Dolan, Roger Michael (J'37) (CMN), Mgr., Labor Relations, T. A. Edison, Inc., 51 Lakeside Ave.; *for mail,* 3 Glennon Pl., West Orange, N.J.
Dolan, Thomas James ('36; '42) (BERAJO), Research Prof., Dept. of Theoretical & Applied Mechanics, 217 Talbot Lab., College of Engrg., Univ. of Illinois, Urbana, Ill.
Dolengo-Kozerovsky, Walter Paul ('39) (SKFE), Mar. Engr., War Dept., Trans. Office, Hdq., 2nd Serv. Comd., Governors Island, New York 4; *for mail,* 159-02 Cryders Lane, Beechurst, L.I., N.Y.
Dolesh, Frank J. (J'34) (CMDL), Methods Planning Sec., Western Elec. Co., Inc., Kearny; *for mail,* 538 Summer Ave., Newark 4, N.J.
Dolezal, Edward ('31; '39) (ET), 507 Creek, Bartlesville, Okla.
Dolhun, Leonard Nickolas (J'45) (NYAEHC), Asst. Ch. Engr., Chicago Rawhide Mfg. Co., 1301 N. Elston Ave., Chicago 22; *for mail,* 5304 W. Bloomingdale Ave., Chicago 39, Ill.
Doll, Alfred William ('38) (BGJNZ), Head, Mech. Engrg. Dept., Pratt Inst., 215 Ryerson St., Brooklyn 5, N.Y.
Dollar, Wm. M. ('96; '08), Retired; Pound Rd., Springbrook, N.Y.
Dolle, August R. C. ('45) (FHS), Plant Betterment Engr., Ebasco Services, Inc., 2 Rector St., New York, N.Y.
Dolve, Robert M. ('30) (HKS), Dean of Engrg., Sch. of Engrg., N. D. Agric. College; *for mail,* 1135—5th St. N., Fargo, N.D.
Dominguez, Arthur Robert (J'41) (DKNO), Engr., Suchar Engrg. & Sales Co., 82 Beaver St., New York 5; *for mail,* 208 W. 23rd St., New York 11, N.Y.
Dominguez, Carlos E. (J'34) (KLS), Capt., Army Serv. Forces, Staff Officer, Tech. Intelligence Office, Intelligence Div., The Pentagon, Washington, D.C.
Dommers, Walter Alexander (J'44) (ZAO), Ch. Engr., Revere Co., 322 North Cherry St., Wallingford, Conn.
Domonoske, Arthur Boquer ('12; '14; '35) (EANFBD), Box 335, Route 1, Willows, Calif.
Donaher, Frederick Leo ('41) (FSLREK), Dist. Mgr., Coal Bur., Norfolk & West. Ry. Co., 1740 Book Bldg., Detroit 26, Mich.
Donahoe, Charles Francis (J'45) (ADE), U.S.N., U.S.S. *Scott,* DE-214, F.P.O., New York, N.Y.; *for mail,* 96 Montvale Ave., Stoneham, Mass.
Donahue, Edward B. (J'41) (JMC), 834 E. Broadway, Cushing, Okla.
Donahue, James Florian (J'44) (CDGLM), Prod. Mgr., Sydney Ross Co., S.A., Calle Manuel M. Flores 142, Mexico, D.F., Mex.
Donahue, Paul (J'36) (JKS), Mech. Engr., Badenhausen Corp., Cornwells Heights, Bucks Co.; *for mail,* 4606 Pulaski Ave., Philadelphia, Pa.
Donald, Wm. J. (A'27) (CAS), Managing Dir., National Electrical Manufacturers Association, 155 E. 44th St., New York 17, N.Y.
Donaldson, Colin Smith ('45), Devel. Engr., English Elec. Co. Ltd.; *for mail,* 3—2nd Ave., Bradford Moor, Bradford, Yorkshire, England.
Donaldson, Robert Robb ('44) (SCZNM), Asst. Ch. Engr., Hagan Corp., 323—4th Ave., Pittsburgh 30, Pa.
Donehower, Robert William (J'42) (CJK), U.S. Army; 22 N. 5th St., Lewisburg, Pa.
Dong, James C. (J'42) (Z), Engr., Charge Aircraft Instruments, Millar Instrument Co., Inc., 22 Plane St., Newark 2, N.J.; *for mail,* 103 E. 10th St., New York 3, N.Y.
Donkersley, Albert Burton ('44) (SKCE), Ch. Engr., Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Donley, Earl H. ('43) (P), Salesman, Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 675 N. Terrace Ave., Mt. Vernon, N.Y.
Donley, Robert Edward (J'44) (ACS), Ensign (D)L, U.S.N.R., Sonar Officer, U.S.S. *Fury,* PG-69, F.P.O., New York, N.Y.
Donnell, Lloyd Hamilton ('29) (BAHN), Research Prof. Mechanics, Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.
Donnelly, Francis J. (J'45) (FSRKO), Asst. Mech. Engr., Gibbs & Hill, Inc., Pennsylvania Sta., New York 1; *for mail,* 28 Clinton Pl., Baldwin, L.I., N.Y.
Donnelly, Geo. E. (J'40), Asst. to Supt., *The News,* 220 E. 42nd St., New York, N.Y.
Donnelly, James A. ('11), Largent, W.Va.
Donnelly, James Joseph, Jr. ('45) (BJMF), Mgr., Engrg. Dept., Miller Co.; *for mail,* Finch Ave., Meriden, Conn.
Donnelly, Leo Francis (J'44) (ENBFKH), Asst. Mech. Engr., Research Div., Internatl. Plainfield Motor Co., Mack Mfg. Corp., 1120 S. 2nd St., Plainfield, N.J.
Donoghue, F. Francis (J'38) (ABKN), Research Engr., Worthington Pump & Mchy. Corp., Appleton St., Holyoke; *for mail,* 60 Green St., Worcester 8, Mass.
Donovan, Daniel Edward ('44) (ACEFK), Pres., Steamship Serv. Corp., Sparrows Point 19; *for mail,* 6823 Dunhill Rd., Dundalk 22, Md.
Donovan, Edw. L. ('24; '38) (CFK), Mech. Engr., Dept. of Pub. Works, Municipal Bldg., New York; *for mail,* 647 Warburton Ave., Yonkers, N.Y.
Donovan, Edward T. (J'21) (BSKFEN), Assoc. Prof. Mech. Engrg., Univ. of N.H., Durham, N.H.
Donovan, George Clement (J'42) (ESBA), Engrg. Draftsman, Caterpillar Tractor Co., Peoria; *for mail,* 911 Wisconsin Ave., Peoria 4, Ill.
Donovan, Michael Joseph ('42) (REB), Mech. Engr., Lima Loco. Works, Inc.; *for mail,* 1804 Wendell Ave., Lima, Ohio.
Donovan, William James (J'40) (KESNOL), Engr., Cecil Boling Co., 415 Lexington Ave., New York; *for mail,* 144 Knollwood Ave., Mamaroneck, N.Y.
D'Ooge, Leonard ('45) (CDKMNZ), Engr., Bercut-Richards Packing Co., Box 1295, Sacramento 14, Calif.
Dooley, James L. (J'42) (AHEKSB), Project Engr., Harvey Mch. Co., Inc., 6200 Avalon Blvd., Los Angeles; *for mail,* 844 Valley St., Burbank, Calif.
Doolin, Elmer Mathew ('33; '35) (FS), Pres., Am. Refrigeration Serv. Co., 6200 Marlboro Pike, S.E., Washington 19, D.C.
Doolittle, Harold L. ('10; '16; F'36) (GHS), *Manager,* '30-'33, *Vice-President,* '33-'35; Retired; 1520 Rose Villa St., Pasadena 4, Calif.
Doolittle, James H. (Non-Member), *Spirit of St. Louis Medallist,* '38; Vice-Pres., Shell Union Oil Corp., 50 West 50th St., New York 20, N.Y.
Doolittle, Jesse Seymour ('39) (EFSB), Assoc. Prof. Mech. Engrg., Pa. State College; *for mail,* 210 Adams Ave., State College, Pa.
Doom, Lewis Garth (J'45) (BCO), Jr. Engrg. Scientist, War Dept. Contr., Box 1663, Santa Fe, New Mex.
Doordan, Robert Emmett (J'46), W. Va. Pulp & Paper Co.; *for mail,* Westvaco Country Club, Covington, Va.
Dopp, Carl A. ('28) (SHEBMY), Directing Engr., Dept. Mgr., Crane Co., 4100 S. Kedzie Ave., Chicago 32, Ill.
Doppel, Leonard ('32) (ABCEFH), Mgr., Cornelia Cab Co., Inc., 915 Knickerbocker Ave., Brooklyn; *for mail,* 61-45—62nd Ave., Maspeth, N.Y.
Doré, Arthur Joseph (J'35) (NDMJCO), Capt., Ord. Dept., U.S.A., Research & Devel. Liaison Officer, Chicago Ord. Dist., 38 S. Dearborn St., Chicago 4; *for mail,* 1707 N. 20th Ave., Melrose Park, Ill.
Doremus, George A. (J'35) (SRF), Mech. Engr., Ebasco Services, Inc., 2 Rector St., New York 6, N.Y.; *for mail,* 144 E. Magnolia Ave., Maywood, N.J.
Dorer, Oscar H. ('12; '21; '35) (BHJ), Cons. Engr., Centrifugal Engrg. Div., Worthington Pump & Mchy. Corp., Worthington St., Harrison; *for mail,* 47 Arsdale Terrace, East Orange, N.J.
Dorf, Marvin L. (J'41) (YSKA), Devel. Engr., Goodyear Tire & Rubber Co., Inc., 1144 E. Market St., Akron; *for mail,* 238 The Brooklands, Akron 5, Ohio.
Dorfan, Morton I. ('28) (BDLOF), Mgr., Dust & Fume Engrg. Div., Am. Fdy. Equip. Co., Mishawaka, Ind.; *for mail,* 1217 Malvern Ave., Pittsburgh 17, Pa.
Dorian, Richard M. (J'42) (CMST), Asst. Mgr., Plant Engr., A. & M. Karagheusian, Inc., Westfield Ave., Roselle Park; *for mail,* 214 Elizabeth Ave., Cranford, N.J.
Doring, Walter Ferdinand, Jr. (J'40) (TJNLM), Devel. Engr., Walter Kidde & Co., Inc., 675 Main St., Belleville; *for mail,* 27 W. Lawn Rd., Livingston, N.J.
Doriot, Georges F. ('23; '35) (CGJLMO), Brig. Gen., QM. Corps, U.S.A., Dir., Military Planning Div., Tempo A, 2nd & T Sts., S.W., Washington 25, D.C.; *for mail,* 12 Lime St., Boston, Mass.
Dorman, Neal W. ('13; '22) (BJM), Ch. Engr., Toledo Mch. & Tool Div., E. W. Bliss Co., 1420 Hastings St., Toledo; *for mail,* 5758 Adelaide Dr., Toledo 12, Ohio.
Dornbach, Robert Frank (J'45) (HBOEMK), Arresting Gear & Catapult Officer, U.S.N., U.S.S. *Kadashan Bay,* CVE-76; *for mail,* 2446 N. 70th St., Wauwatosa 13, Wis.
Dornbirer, Stanton D. ('39) (HJKNO), Constr. Engr., S. Morgan Smith Co., York, Pa.
Dornbrook, Fred L. ('22) (BSKFC), Ch. Engr., Power Plants, Wis. Elec. Power Co., Box 2046, Milwaukee 1; *for mail,* 4426 N. Farwell Ave., Milwaukee 11, Wis.
Dorner, Frederick H., Jr. (J'38) (SFK), V.P., Dorner Co., 2766 N. Downer Ave., Milwaukee 11; *for mail,* Box 1000, Milwaukee 1, Wis.
Dorow, Ray O. ('39; '45) (CDLMNT), 43 Blue Ridge, Kansas City 3, Mo.
Dorward, John Greig, Jr. (J'37) (BHMC), Engr., Designer, Dorward Pump Co., 210 Mission St., San Francisco 5; *for mail,* 467 Central Ave., Alameda, Calif.
Doster, Howard George ('40) (CJRS), V.P., Ch. Engr., Ohio Injector Co.; *for mail,* 253 Broad St., Wadsworth, Ohio.
Dotter, Richard A. (J'39), Bunker Place, Wantagh, N.Y.
Doty, Albert J. ('43) (SMLZ), Ch. Engr., Am. Box Board Co., 470 Market St., S.W., Grand Rapids 2; *for mail,* 1420 Sherman St., S.E., Grand Rapids 6, Mich.
Doubrava, Eugene N. (J'41) (ACM), Thompson Products, Inc., 2196 Clarkwood, Cleveland; *for mail,* 2624 S. Taylor Rd., Cleveland Heights, Ohio.
Dougherty, William F., Jr. (J'36) (BGH), Tester, Arma Corp., 254-36th St., Brooklyn, N.Y.; *for mail,* 17 Hoyer Ave., Westwood, N.J.
Doughtie, Charles Edwin, Jr. ('27; '35) (BEFKTY), V.P. & Mgr. Mech. & Elec. Depts., Robert & Co., Inc., 706 Bona Allen Bldg., Atlanta 3, Ga.
Doughtie, Venton Levy ('21; '32; '35) (NBMA), Prof. Mech. Engrg., Univ. of Tex., Univ. Sta., Austin 12, Tex.
Doughty, Charles John ('45), Mech. Supvr., Austin Co., 19 Rector St.; *for mail,* Murray Hill Hotel, 112 Park Ave., New York, N.Y.
Doughty, Franklin O. (J'39) (FHS), 1922 Seneca Rd., Canby Park, Wilmington, Del.
Doughty, Gerald N. (J'42) (NMC), Engrs. Asst., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 1579 S. 60th St., West Allis 14, Wis.
Doughty, Wm. F. ('99; '15), Suite 2824, 17 Battery Pl., New York, N.Y.
Douglas, Donald Craig ('23; '30; '35) (NHKFSB), Lab. Supvr., Dept. of Engrg. Research, Univ. of Mich.; *for mail,* 1320 W. Huron St., Ann Arbor, Mich.
Douglas, Frank Shelton (J'42) (AFBNSJ), Test Equip. Engr., Exper. Div., Wright Aero. Corp., Beckwith Ave., Paterson 3; *for mail,* 22 N. 13th St., Haledon, Paterson 2, N.J.
Douglas, Frederick Orrin (J'42) (BNJC), Engr., Spec. Screw Products Co., 5445 Dunham Rd., Bedford; *for mail,* 1305 Yellowstone Rd., Cleveland 21, Ohio.
Douglas, George Mellis ('22) (AES), Northcote, Lakefield, Ont., Can.
Douglas, John ('30), Redlac, Colinton, Edinburgh, Scotland.
Douglas, Melvin ('45), Rm. 1623, 53 W. Jackson, Chicago 4; *for mail,* 5236 N. New England, Chicago 31, Ill.
Douglass, Alfred E. ('21; '30) (CDL), Pres., Allentown Portland Cement Co., 124 Bridge St., Catasauqua, Pa.
Douglass, Dwight ('46) (FKS), Supt. of Power, Hartford Elec. Light Co., 266 Pearl St., Hartford 15, Conn.
Douglass, Malcolm E. (J'36) (BKM), Lab. Engr., Gen. Elec. Co., 40 Federal St., Lynn; *for mail,* Box 9, West Lynn, Mass.
Douglass, Roderick Brundage (J'41) (KHFBAM), Detail Engr., Harrison Radiator Div., Gen. Motors Corp.; *for mail,* 122 Cottage St., Lockport, N.Y.
Douglass, Walter Landon, Jr. (J'46) (BJM), Lukens Steel Co.; *for mail,* Rm. 435, Y.M.C.A., Coatesville, Pa.
Doukas, George (J'40) (HKL), Lt. (j.g.), U.S.N.R., 316 St. John St., Havre de Grace, Md.
Douthit, Judd H. (J'30), Badin, N.C.
Douwes, Hendrik Brugt (J'39) (BNJMOC), Engr., Cent. Mch. Co., 398 Market St., Newark; *for mail,* 1841 Arbor Lane, Union, N.J.
Dow, George L. ('44) (SODF), Mech., Elec. Supt., Albert Kahn Associated Archs. & Engrs., Inc., 345 New Center Bldg., Detroit 2; *for mail,* Box 405, Ann Arbor, Mich.
Dow, Herbert W. ('17), Exec. Sales Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee; *for mail,* 2524 E. Shorewood Blvd., Milwaukee 11, Wis.
Dow, Richard Francis ('21) (BHMY), Research Engr., Whitney Chain & Mfg. Co., 237 Hamilton St., Hartford 2; *for mail,* 162 Edgewood St., Hartford 5, Conn.
Dowd, Stanley B. (A'29) (CHBM), Sales Mgr., Leland-Gifford Co., 1001 Southbridge St., Worcester 1, Mass.
Dowdell, Samuel Hosmer ('44), Plant Engr., Florida Power & Light Co., Box 3100, Miami 30; *for mail,* Wimauma, Fla.
Dowden, Edmund Van Dyke (J'40) (CMDJBW), Plant Supt., Osborn Mfg. Co., Hamilton Ave., Cleveland, Ohio; *for mail,* 502 N. Mississippi River Blvd., St. Paul 4, Minn.
Dowden, William Morrill (J'45), Ensign, U.S.N., 589 Fellsway W., Medford 55, Mass.
Dowding, Leonard Frank (J'45) (KLP), Sr. Engr., Texas Co., 205 E. 42nd St., New York 17, N.Y.
Dowell, Dawson ('20; '22; '35), Prof. Mech. Engrg., Drexel Inst., 32nd & Chestnut St., Philadelphia, Pa.
Dowling, Donald Lynd ('26; '34; '35) (LEFGAB), Dist. Mgr., Roots-Connersville Blower Corp., 122 E. 42nd St., New York 17, N.Y.; *for mail,* 166 Carlisle Terrace, Ridgewood, N.J.
Down, Sidney George ('43) (R), 1st V.P., Westinghouse Air Brake Co., Wilmerding, Pa.
Downes, Daniel T. ('45) (JNYZOB), Mech. Engr., Research Lab., Pittsburgh Plate Glass Co., Creighton; *for mail,* 462 Biddle Ave., Pittsburgh 21, Pa.
Downes, Nate Worswick (11; '17; '25) (KFDSZH), Asst. Supt., Charge Bldgs. & Grounds, Kansas City, Mo., Sch. Dist., 1840 E. 8th St., Kansas City 1, Mo.
Downey, Bert Fought ('29; 35), Secy., Treas., Gen. Mgr., Yost Superior Co., Center & Jefferson, Springfield, Ohio.
Downey, Stephen Francis, Jr. (J'41) (MJYC), Engr., Stand. Container, Inc., 350 Hoover Ave., Bloomfield; *for mail,* 44 Bell St., Orange, N.J.

Downey, W. W. ('22), Dir. of Engrg., Mech. Engr., N.Y. Hospital, 523 E. 70th St., New York, N.Y.
Downing, Burton H. ('40), 80 W. Putnam Ave., Greenwich, Conn.
Downing, Fred Thomas ('42) (BFHNS), Mech. Engr., Div. Office, U.S. Engrs., 304 Santa Fe Bldg., Dallas 2, Tex.; *for mail,* 1041 S.E. 56th Ave., Portland 15, Ore.
Downing, Harry Frye ('19; '35) (NJB), Mech. Engr., Naval Shipyard; *for mail,* 135 Thaxter Rd., Portsmouth, N.H.
Downs, Andrew S. (J'41) (AES), Steam Serv. Engr., Westinghouse Elec. Corp., 1314 Texas Ave., Houston 2, Tex.
Downs, Charles Raymond ('32) (OLKDFY), Cons. Chem. Engr., Chemists' Bldg., 50 E. 41st St., New York 17, N.Y.
Downs, Eugene Lawrence (A'44) (BCDJ), Asst. to Gen. Supt., Automatic Elec. Co., 1033 W. Van Buren St., Chicago 7, Ill.
Downs, Francis T. (J'37) (CLM), Technician, 5th Grade, Detached Enlisted Men's List, Hdq. Det., S.C.U. 1908, Camp Cooke, Calif.
Downs, Herbert R. (J'37), Lt. Comdr., U.S.N.R., 486—23rd St., Niagara Falls, N.Y.
Downs, James Benedict T. (J'42) (BJSN), Asst. Prof. Mech. Engrg., Rice Inst., Houston 1, Tex.
Downs, John Nelson ('45), Drafting Supvr., Babcock & Wilcox Co., Barberton; *for mail,* Doylestown, Ohio.
Downs, John Whitney ('29; '35) (BDHJKN), Owner, Downs Crane & Hoist Co., 540 W. Vernon Ave., Los Angeles 37, Calif.
Downs, Sewell Henry ('23; '34), V.P., Ch. Engr., Clarage Fan Co., 619 Porter St., Kalamazoo; *for mail,* 1562 Spruce Dr., Kalamazoo 40, Mich.
Downs, Sydney Featherman, Jr. (J'45), Engr., York Corp., York; *for mail,* 817 Hoffman Terrace, Easton, Pa.
Downs, William David (J'39) (BEJKC), Power Plant Lab., Wright Field, Dayton, Ohio.
Dows, Harold W. ('20; '26; '35) (CJLM), Prof. Mech. Engrg., Worcester Poly. Inst., Boynton St., Worcester 2; *for mail,* 47 Barnard Rd., Worcester 5, Mass.
Dows, Samuel R. ('24; '26; '35) (DFK), 630 Curtis St., Albany 6, Calif.
Dowson, Harry R. ('16; '17; '27) (NBJMAL), Devel. Engr., Dixie Cup Co., 24th St. & Dixie Ave.; *for mail,* 1801 Fairview Ave., Easton, Pa.
Doxey, H. E. (J'42), 4250 Quincy St., N.E., Minneapolis, Minn.
Doyle, Edgar Dwight ('27) (ZBRSLH), Pat. Engr., Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44, Pa.
Doyle, Edwin Francis (J'44), 39 Riverside St., Amityville, N.Y.
Doyle, James Patrick (J'43) (SMCE), Engr., Gibbs & Cox, Inc., 21 West St., New York, N.Y.; *for mail,* 2801 Hudson Blvd., Jersey City 6, N.J.
Doyle, John Edmund (J'37) (ABN), Chief of Structures, G & A Aircraft, Inc., Willow Grove; *for mail,* Bristol Rd., Warrington, Pa.
Doyle, William Joseph (J'46), 6331 Sherman St., Philadelphia 44, Pa.
Doyle, William Leonard Hunt ('16; '40) (EBF NSJ), Asst. Dir. Research, Caterpillar Tractor Co., Peoria; *for mail,* 309 Moss Ave., Peoria 5, Ill.
Dozier, Leonard Campbell, Jr. (J'42) (ZA), Research Engr., Servomechanisms Lab., Mass. Inst. of Tech., Cambridge; *for mail,* 24 Prescott St., Cambridge 38, Mass.
Drabek, Stephen (J'41) (ANHBKF), Engr., Design & Devel., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 1699 Cambridge St., Cambridge 38, Mass.
Drabelle, John Mitchell ('19) (FSEK), Ch. Engr., Iowa Elec. Light & Power Co., Security Bldg.; *for mail,* 521 Fairview Dr., Cedar Rapids, Iowa.
Drace, Charles Albert (J'44) (MCJO), Engr., Joshua Hendy Iron Works, Sunnyvale; *for mail,* 271 Leland Ave., Palo Alto, Calif.
Drachman, Myron Joshua ('45) (NBZLSW), Engr., Peerless Labs., 467—10th Ave., New York 18; *for mail,* 81 Archer Dr., Bronxville 5, N.Y.
Dragon, Peter Frank (J'44), 5549 Merrilwood Dr., Oakland 11, Calif.
Drake, Charles Edward (J'39) (EFKLMO), Engr., Patterson-Kelley Co., Inc., Warren St., East Stroudsburg; *for mail,* 411 E. Bryant St., Stroudsburg, Pa.
Drake, Charles M. ('28; '35) (CSFE), President's Staff, Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 716 Pilgrim Ave., Birmingham, Mich.
Drake, Daniel Webster (J'41) (BAOGC), Research Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 75 N. Michigan, Pasadena 4, Calif.
Drake, Frank Babbitt ('40) (CMN), Pres., Johnson Gear & Mfg. Co., Ltd., 921 Parker St., Berkeley 2; *for mail,* 20 Santa Clara Ave., San Francisco 16, Calif.
Drake, Joseph Horace ('45) (SOFKEC), Cons. Mech. Engr., 10 S. Laura St., Jacksonville 1; *for mail,* 1312 Le Baron Ave., Jacksonville 7, Fla.
Drake, Robert W. ('17; '35) (BHOSZ), Cons. Engr., 18 South St., Middlebury, Vt.
Drake, William Victor ('27; '35) (SBCFKR), Mgr., Power Generation Dept., Monongahela Power Co., 301 Adams St.; *for mail,* 896—8th St., Fairmont, W.Va.
Drandell, Jack (J'45), *Charles T. Main Award,* '45; 1919½ N. Pearl, Dallas 1, Tex.
Draper, Chas. S. ('34; '35) (ABE), Prof. Aero. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge, Mass.
Draxler, John Alexander (J'43) (CDMN), Asst. to Dir. of Engrg., Elwell-Parker Elec. Co., 4205 St. Clair Ave., Cleveland 14, Ohio.
Dredge, Armiger Faulknor ('41) (OSHKC), Cons. Engr., 5046 Cochrane Ave., Oakland 11, Calif.
Dreffein, Henry A. ('13), Pres., Flinn & Dreffein Co., 308 W. Washington St., Chicago, Ill.
Dreffein, Otto F. (J'41) (KMS), Asst. to Ch. Engr., Perfex Corp.; *for mail,* 319 N. St. Louis Ave., Chicago, Ill.
Dreher, Robert Henry (J'42), Lt., U.S.N.R.; 18 Hone Ave., Oil City, Pa.
Dremer, Joseph John (J'44) (EAHBCM), Maint. Engr., Stand. Oil Co. of Calif., 225 Bush St., San Francisco 20; for mail, 2332 College Ave., Berkeley 4, Calif.
Drescher, Rowland Boyce ('45) (SZEFHL), Dist. Mgr., Republic Flow Meters Co., 237—4th Ave., Pittsburgh 22; *for mail,* 1705 Williamsburg Pl., Pittsburgh 21, Pa.
Drescher, Russell E. (J'41), c/o Ruthe Coxe, Sound View Ridge, Glenville, Conn.
Dresden, Daniel ('28) (HMS), Prof. Ir., Rotterdamsche Weg 107, Delft, Netherlands.
Drew, Thomas Bradford ('38) (KBSFL), Prof. Chem. Engrg., Columbia Univ., New York 27, N.Y.
Drew, Walter Edward ('36; '43) (CFKSY), Effic. Engr., Mun. Power Plant, City of Jacksonville, Talleyrand Ave.; *for mail,* 7916 Lorain St., Jacksonville 6, Fla.
Drew, William B. F. ('43) (HOSCEF), Secy., Treas., Ch. Engr., J. J. Murphy, Inc., 340 E. 44th St., New York 17, N.Y.
Drewelow, Earl James ('42) (SFEHCM), Supt., Kan. Plants, Empire Dist. Elec. Co., Joplin, Mo.; *for mail,* Box 72, Riverton, Kan.
Drewett, William A. ('88) (BCM), Retired; 65 Medway St., Providence, R.I.
Drewry, Ivey O. (J'39), Maj., 130th Ord. Maint. Bn., A.P.O. 258, c/o Postmaster, Camp Polk, La.
Drewry, Montrose Keller ('24; '32) (CFJKS), *Junior Award,* '31; Asst. Ch. Engr. of Power Plants, Wis. Elec. Power Co., Milwaukee 1; *for mail,* 3019 S. Shore Dr., Milwaukee 7, Wis.
Drewry, Thomas Watkins (J'35), Lombardy St., Richmond, Va.
Drexel, Charles Francis (J'44) (AEH), Ensign, U.S.N.R., Staff-Air Pac. Sub. Com. Fwd., Box 26, F.P.O., San Francisco, Calif.
Dreyer, Elmer John ('16; '24) (EFOS), Mech. Engr., Boiler Div., Henry Vogt Mch. Co., 10th & Ormsby Sts., Louisville 10, Ky.
Dreyer, Elmer L. (J'27) (CHFB), Supt., Pacific Coast Pipe Lines, Texas Co., Box 817, Wilmington, Calif.
Dreyer, John Arthur (J'40) (BLMDG), Lt., U.S.N., 1549 Ceres St., Crockett, Calif.
Dreyfus, Edwin David ('05; '11) (CMNOSW), Cons. Engr., 5849 Ferree St., Pittsburgh 17, Pa.
Driemeyer, George Fred ('45) (CR), V.P., Charge Sales, Gen. Steel Castings Corp., Eddystone, Pa.
Drinka, Joseph J. (J'41) (BRNE), Asst. Engr., Car Constr., Milwaukee Shops, Milwaukee R.R.; *for mail,* 323 N. 35th St., Milwaukee 8, Wis.
Driscoll, John Manning ('27; '40) (SHFKC), Prod. Supt., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 127 Colonial Pkwy., Manhasset, L.I., N.Y.
Driver, Arthur Harris ('13; '24; '35) (MND), Engr., Soulé Steel Co., 1750 Army St., San Francisco 24; *for mail,* 640 E. 15th St., Oakland 6, Calif.
Drobile, Albert W. ('29) (CST), 311 West Ave., Wayne, Del Co., Pa.
Drogue, James Albert ('29) (DFKMSY), Plant Engr., Metro-Goldwyn-Mayer, 120 Arlington Ave., Bloomfield, N.J.
Drucker, Daniel Charles (J'42) (ABH), Asst. Prof., Dept. of Mechanics, Tech. Center, Ill. Inst. of Tech., Chicago 16, Ill.
Drucker, Jules Herbert (J'41) (SKHF), Lt., Engr., U.S. Maritime Serv.; *for mail,* 160 Bergen Ave., Jersey City 5, N.J.
Drucker, Ned (J'41) (CDLMNO), Prod. Engr., Schenley Distilleries, Inc., 26 E. 6th St., Cincinnati 21; *for mail,* 4564 Paddock Rd., Cincinnati 29, Ohio.
Druhan, John Laurence ('44) (OCDRS), Staff Engr., DeLeuw, Cather & Co., 20 N. Wacker Dr., Chicago 6; *for mail,* 321 N. 1st St., Libertyville, Ill.
Drum, Leo J., Jr. (J'35) (CAMYKO), 7 Gilmer Ave., Montgomery 6, Ala.
Drummond, Arthur S. (J'35) (CFS), Box 5, R.R.2, Port Huron, Mich.
Drummond, Warren C. ('28) (EFS), Power Prod. Engr., Pub. Utility Engrg. & Serv. Corp., 231 S. La Salle St., Chicago 4, Ill.
Drummond, Wm. D. (J'40) (JS), Apprentice Engr., Edgar Thomson Works, Carnegie-Ill. Steel Corp., Braddock; *for mail,* R.F.D. 1, Bridgeville, Pa.
Drummond, William Duncan (J'42) (NJS), Designing Engr., Internatl. Harvester Co. Ltd.; *for mail,* 50 Garfield Ave. S., Hamilton, Ont. Can.
Drumpelmann, Carlton Theodor ('43) (CODM), Works Engr., Gen. Engrg. Dept., Natl. Carbon Co., Inc., Cleveland, Ohio.
Druschitz, Alexander (J'41) (AEJY), Radio Technician, 3rd Class, Navy Pier, Chicago; *for mail,* 5626 S. California Ave., Chicago 29, Ill.
Drutzu, S. T. ('21; '35) (CSML), Rate Engr., Ebasco Internatl. Corp., 2 Rector St., New York 6; *for mail,* 24 North Dr., Malba, L.I., N.Y.
Dryden, Hugh Latimer ('36) (BAH), Asst. Dir., Natl. Bur. of Stands., Connecticut Ave. & Van Ness St., Washington 25, D.C.
Dryer, Edward Lewis ('42) (CDMRY), Ch. Project Engr., Armstrong Cork Co.; *for mail,* 349 N. Charlotte St., Lancaster, Pa.
Drypolcher, William ('30; '37) (CMNJZ), Ch. Engr., Wyrstapes Co., 200 Hudson St., New York 13, N.Y.
Drysdale, Walter Douglas ('20) (FMS), Asst. Ch. Engr., Power Plants, Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Du, Darfoon (J'42) (FKLS), Engr., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Duban, Anthony J. (J'41) (BJS), Alexandria, Ky.
Dublan, Sterling Robert (J'44) (CHYG), Maint. Engr., Orleans Matls. & Equip. Co., 1556 Tchoupitolas St., New Orleans; *for mail,* 2512 Elysian Fields Ave., New Orleans 17, La.
Dubosclard, Paul Pierre Michel ('33; F'44) (GNO), Topanga, Calif.
Du Brul, Ernest F. (A'00) (C), 2612 Handasyde Ct., Cincinnati 8, Ohio.
Ducker, William Lyon ('31; '40) (NMBHY), Cons. Engr., 1004 S. College, Tulsa 4, Okla.
Duckworth, Douglas Haig (J'44) (LS), Fire Protection Engr., Sprinklered Risk Dept., Canadian Underwriters Assn., 44 Victoria St., Toronto; *for mail,* 527 Melrose Ave., Toronto 12, Ont., Can.
Dudas, John, Jr. (J'45) (AMLCJN), Spec. Process Engr., Wright Aero. Corp., Beckwith Ave., Paterson; *for mail,* 115 Luddington Ave., Clifton, N.J.
Dudko, Nicholas (J'42) (BRS), 1820 Holland Ave., Bronx 60, N.Y.
Dudley, John Herren (J'42) (ZAMOJL), 1540 Center St., Bethlehem, Pa.
Dudley, Samuel William ('04; '16; F'41) (BCR), *Manager,* '36-'39; Dean, Sch. of Engrg., Yale Univ., New Haven, Conn.
Dudley, Winston Mansfield ('36; '45) (BNJO ER), *Junior Award,* '42; Assoc. Prof. Engrg. Mechanics, Case Sch. of Applied Science, 10900 Euclid Ave., Cleveland 6, Ohio.
Dudnick, Solis (J'44) (ACE), 19 N. Rosborough Ave., Ventnor, N.J.
Duennes, Frank C. ('38), Asst. Ch. Engr., Calumet Sta., Commonwealth Edison Co., 3200 E. 100th St., Chicago, Ill.
Duer, Rufus King ('37; '45) (CMRSDK), Motor Specialist, Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Dueringer, Walter Edward ('36) (AFLRSZ), Asst. Sales Mgr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio.
Duerr, Francis Robert (J'43), Ensign, U.S.N.R., 2112 Eastview Ave., Louisville, Ky.
Duff, Jas. A. (J'38) (CDE), Ch. Engr., Westchester Country Club, Rye; *for mail,* 30 Lyons Rd., Scarsdale, N.Y.
Duff, Ralph Bonner ('42) (CLJDYZ), Engr., Stand. Oil Co. of N.J., North Baton Rouge; *for mail,* 2271 Ferndale Ave., Baton Rouge 15, La.
Duffey, Paul Raymond ('36) (CFJOS), Spec. Engr., Youngstown Sheet & Tube Co., Youngstown 1; *for mail,* 586 Tod Lane, Youngstown 4, Ohio.
Duffy, Edward Charles (J'43) (SCF), Supt., Gen. Sta. & Substa., Queens Borough Gas & Elec. Co., 1610 Far Rockaway Blvd., Far Rockaway, L.I., N.Y.
Duffy, James J. (J'45), 216 Elmhurst St., Highland Park 3, Mich.
Duffy, John Reid (J'43) (EFS), 2215 S. Ridgeway Ave., Chicago 23, Ill.
Duffy, Thomas Hill ('39) (BEFKNS), Asst. Fuel Serv. Engr., Chesapeake & Ohio Ry. Co., Richmond 10; *for mail,* 3911 Fauquier Ave., Richmond 22, Va.
Duffy, Thomas Joseph (J'32) (CDAM), Dist. Engr., Liberty Mutual Ins. Co., 1100 Healey Bldg., Atlanta 1, Ga.
Dugan, Paul David (J'42), Maj., 439th T.C. Gp., Hdq., A.P.O. 133, c/o Postmaster, New York, N.Y.
Dugan, Warren Gilbert (J'39) (BSRKH), 562 Whitney Ave., New Haven 11, Conn.
Duggan, Herbert Garrison (J'43) (AHS), Lt. (j.g.), U.S.N., LST-805, F.P.O., San Francisco, Calif.
Dukelow, Samuel G. (J'41) (ZSFKL), Sales-Serv. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio; *for mail,* 8527 Findlay, Houston 12, Tex.
Dull, Raymond William ('08) (BDHOY), Owner, 30 N. La Salle St., Chicago 2, Ill.
Dulmage, William Wineford ('44), Supt. of Power, Ford Motor Co.; *for mail,* 22750 Alexanderine, Dearborn, Mich.
Du Mond, Loyd Paine (A'38) (AS), Mgr., W. R. Roach Co.; *for mail,* Croswell, Mich.
Dumont, Arthur, Jr. (J'43) (CMJBDS), Ship Supt., Gulf Shipbldg. Corp.; *for mail,* 500—5th Ave., Chickasaw, Ala.

Dumsa, Alfred Victor (J'45) (ABCMOS), Sales Engr., Cyril J. Buke Constr. Mchy., 6451 E. McNichols Rd., Detroit; *for mail*, 17403 Kentucky Ave., Detroit 21, Mich.
Dun, Henry Walke, Jr. ('35) (CMOR), R.D. 2, Wilton, Conn.
Dunagan, Shirley V. ('28; '35; '35) (CDMGJ), Cost Control Supvr., Murray Corp. of Am., 7700 Russell St., Detroit 11; *for mail*, 4971 Ridgewood St., Detroit 4, Mich.
Dunaway, Reece Judson (J'44), Box 718, Coolidge, Ariz.
Dunbar, Aubrey Windley ('33; '44) (WCTM), Gen. Supt., Tomlinson of High Point, Inc., 305 W. High St.; *for mail*, 307 E. Farris Ave., High Point, N.C.
Dunbar, Norman Capen (J'43) (SLJ), 1st Lt., Army Air Forces; *for mail*, 168 Main St., Flemington, N.J.
Duncan, Carroll Allen ('29) (CKOS), Project Mgr., Sanderson & Porter, 52 William St., New York, N.Y.; *for mail*, Box 66, Williamsport, Md.
Duncan, David (J'42) (CN), Mech. Engr., Internatl. Nickel Co. of Can. Ltd., Copper Cliff; *for mail*, Apt. 10, 107 Kingsmount Ave., Sudbury, Ont., Can.
Duncan, Donald S. (J'36) (BG), 2414 Dayton Dr., Ann Arbor, Mich.
Duncan, Gerald D. (J'40) (NAEBHM), Peoria, Ariz.
Duncan, J. Robert (J'38) (LCDJ), Mech. Engr., Davison Chem. Corp., 20 Hopkins Pl., Baltimore 3, Md.
Duncan, John C. ('14) (CGMAYD), Owner, Firm of J. C. Duncan, 42 Broadway, New York 4, N.Y.
Duncan, John M. ('21; '35), Advis. Engr., Rural Electrification Admin., S. Bldg., Dept. of Agric., Washington, D.C.
Duncan, Sydney Ford ('31; '38) (NOJBAZ), Assoc. Prof. Mech. Engrg. Univ. of So. Calif., 3551 University Ave., Los Angeles 7; *for mail*, 3872 Welland Ave., Los Angeles 43, Calif.
Duncan, Theodore Norman (J'43) (AECBM), Lt., U.S.N., U.S.S. *Oahu*, ARG-5, F.P.O., Naval Oper. Base, Mole, San Pedro, Calif.
Duncan, Wm. M. ('11), Pres., Ill. Stoker Co., 102 W. 7th St., Alton, Ill.
Duncan, William Yeats, Jr. ('21; '26) (CHMY), Capt., U.S.N.; *for mail*, 62 Cleveland Rd., New Haven 15, Conn.
Dundas, William Alfred ('43) (BDHKS), Engr., Mech. Design, The Sanitary Dist. of Chicago, 910 S. Michigan Ave., Chicago 5; *for mail*, 1234 Cherry St., Winnetka, Ill.
Dunes, Walter Anton (J'43) (AMCKSO), Engr., Asst. to Project Engr., Apex Elec. Mfg. Co., E. 136th St. & Kuhlman Ave., Cleveland; *for mail*, 3318 Grantwood Dr., Parma 9, Ohio.
Dungan, Donald K. (J'21), Box 393, San Carlos, Calif.
Dungan, E. Root ('41) (FLOS), Mech. Engr., Birds Eye-Snider, Div. of Gen. Foods Corp., 40 Franklin St., Rochester; *for mail*, 221 N. Main St., Albion, N.Y.
Dunglinson, Burton ('39), Managing Dir., Electroflo Meters Co. Ltd., Abbey Rd., Park Royal, London, N.W. 10, England.
Dunham, Byron William ('14) (RHKMNA), Ch. Engr., Edgewater Steel Co.; *for mail*, 434—9th St., Oakmont, Allegheny Co., Pa.
Dunham, Elmer Joseph ('42) (DHBJMY), Ch. Engr., Clark Tructractor Div., Clark Equip. Co., Battle Creek, Mich.
Dunham, Lindsay, Jr. (J'45) (CJM), 300 W. 106th St., New York, N.Y.
Dunham, Walter Edward ('14) (CJR), Retired; 704 Main St., Glen Ellyn, Ill.
Duni, Robert Louis (J'40) (W), Asst. Mgr., Plant Engr., Goodyear Synthetic Rubber Corp., Box 207, Torrance, Calif.
Dunigan, Edw. B. (A'30), Catalog Prod. Mgr., Sears Roebuck & Co., Chicago; *for mail*, 140 S. Euclid Ave., Oak Park, Ill.
Dunkin, William Van ('09) (CDLMN), Prof., Ga. Sch. of Tech., 654 Cumberland Rd., N.E., Atlanta, Ga.
Dunkle, Heber Hamilton (J'43) (JBSHM), Engr. Dept., Goetze Gasket & Packing Co., Allen Ave., New Brunswick; *for mail*, R.F.D. 2, Giles Rd., Metuchen, N.J.
Dunlap, Charles Kirkland, Jr. (J'43), 1st Lt., U.S.A., 343rd Airdrome Sqd., 3rd Air Commando Gp., A.P.O. 919, c/o Postmaster, San Francisco, Calif.; *for mail*, R.F.D. 2, Hartsville, S.C.
Dunlap, Richard Morris (J'45) (ABK), Robert College, Istanbul, Turkey.
Dunlap, Robert Hawkins (J'43) (GJCELM), Magnesium Div., Revere Copper & Brass, Inc., 230 Park Ave., New York 17, N.Y.
Dunlop, Charles William ('29) (CM), V.P., Prod. & Engrg., Safety Car Htg. & Ltg. Co., Inc., Box 904, New Haven 4; *for mail*, 31 Marvel Rd., New Haven 15, Conn.
Dunlop, Harry James (J'42) (ACMH), Box 84, Route 2, Placerville, Calif.
Dunmire, C. Ward (J'45), 706 Westmoreland Ave., Jeannette, Pa.
Dunn, Carroll H. (J'38) (CD), Lt. Col., Corps of Engrs.; *for mail*, 231 E. Burlington, Iowa City, Iowa.
Dunn, Gano ('11; F'41; H'44), Pres., J. G. White Engrg. Corp., 80 Broad St., New York, N.Y.
Dunn, J. Jay ('17) (BHKZF), Retired; 200 Fountain Ave., Ellwood City, Pa.
Dunn, James Jay ('30) (CGLM), Res. Mgr., Louisville Div., Natl. Distillers Products Corp., Box 479, Louisville 2; *for mail*, 500 Kenwood Dr., Louisville 8, Ky.
Dunn, Raymond Gano (J'43) (BER), Engrg. Draftsman, Premier Gear & Mch. Works, 17th & Thurman, Portland; *for mail*, 37 N.E. Morgan St., Portland 11, Ore.
Dunn, Royal Charles (J'45) (KSHCE), 1914 Chicago Ave., Minneapolis 4, Minn.
Dunn, Samuel Orace (A'44) (R), Chmn., Simmons-Boardman Publ. Corp., 105 W. Adams St., Chicago 13, Ill.
Dunn, Stephen Monte ('26) (HESO), Ch. Mech. Engr., Water System, Dept. of Water & Power, 207 S. Broadway, Los Angeles 12; *for mail*, 1680 Hill Dr., Los Angeles 41, Calif.
Dunn, William John (J'43) (FKS), Combustion Engrg. Co., Inc., 80 Federal St., Boston 10, Mass.
Dunne, Samuel Brendan ('43) (JMOKLH), Design Engr., A. B. Murray Co., Green Lane, Elizabeth, N.J.
Dunnell, William W., Jr. ('27; '30; '33) (NMJHAR), Ch. Engr., Rivett Lathe & Grinder, Inc., 18 Riverview Rd., Boston 35; *for mail*, 86 Myrtle St., Boston 14, Mass.
Dunnican, George Walter ('30; '35) (EFSDCJ), Ch. Engr., Lehn & Fink Prod. Corp., 192 Bloomfield Ave.; *for mail*, 838 Broad St., Bloomfield, N.J.
Dunning, Harry ('28) (DBMOY), Partner, Dunning Lueckel Engrg. Co., 17 John St., New York 7, N.Y.
Dunnington, Francis Alden (J'42), Salvage Supvr., Charge Repairs, Airplane Div., Curtiss-Wright Corp., Kenmore & Vulcan St., Buffalo; *for mail*, 744 S. Tiorunda Dr., Buffalo 11, N.Y.
Dunnington, William ('38; '44) (CLS), Engr., E. I. du Pont de Nemours & Co., Wilmington; *for mail*, 2622 W. 18th St., Wilmington 47, Del.
Dunsford, Jan R. ('31) (CJM), 1007 Investment Bldg., Pittsburgh, Pa.
Dupont, Andrew Taylor ('18; '24; '29) (DESL), Comdr., U.S.N.R., Exec. Officer, Office of Asst. to Indus. Mgr., Norfolk Naval Shipyard, 112 S. Gay St., Baltimore 2; *for mail*, Box 299, Route 9, Baltimore 25, Md.
duPont, Benjamin Bonneau (J'44), Student, Yale Univ.; *for mail*, 2666 Yale Sta., New Haven, Conn.
Duppstadt, J. Robert (J'42) (ABESHW), Stress Analyst, Curtiss-Wright Corp., Genessee, Buffalo; *for mail*, 101 Norwood Ave., Buffalo 13, N.Y.
Du Pre, Frederick Augustus (J'43) (NEMRBO), Mech. Engr., Drafting & Designing, Oswald Engrg. Services, Inc., 504 Queen & Cresent Bldg.; *for mail*, Apt. G, 711 Royal St., New Orleans 12, La.
DuPriest, John R. ('11) (SENHKC), Prof., Mech. Engrg., Univ. of Minn., Minneapolis, Minn.
Durand, Nelson C. ('99) (CLM), Retired; 116 Prospect St., East Orange, N.J.
Durand, William F. ('83; H'34) (ABH), *Vice-President*, '11-'13, *President*, '25; *A.S.M.E. Medallist*, '45; Prof. Emeritus, Mech. Engrg., Stanford Univ., Stanford University, Calif.
Durant, Aldrich ('06; '13) (C), Business Mgr., Harvard Univ., Cambridge 38, Mass.
Durbeck, Albert C. ('21; '23; '35) (ACDNTZ), Sales Engr., Fed. Bearings Co., Inc.; *for mail*, 73 Hinkley Pl., Poughkeepsie, N.Y.
Durgin, Clyde Munroe ('22; '28) (BSNEHG), Reproduction Supvr., New England Power Serv. Co., 441 Stuart St., Boston; *for mail*, 20 Danville St., West Roxbury 32, Mass.
Durham, George Edwin (J'40) (RJC), Asst. to Pres., Wheeling & Lake Erie Ry. Co., 1657 Union Commerce Bldg., Cleveland 14, Ohio.
Durham, J. E., Jr. (A'40), V.P., Bonney Forge & Tool Works, Allentown, Pa.
Durkee, Chauncey Hilton ('19; '35; '35) (ACZM), Tech. Editor, Airadio Co., Inc., Barry Pl., Stamford, Conn.
Durland, Arthur C. (J'43) (EKS), 314 Orchard Ave., Sunnyvale, Calif.
Durland, Charles Mortimer (J'43) (SKHC), Ensign, U.S.N.R., Sullivan Dry Dock & Repair Corp., 23rd St., Brooklyn 32, N.Y.
Durland, Merrill A. ('22; '30) (NB), Prof. Mch. Design, Asst. Dean, Kan. State College, Manhattan, Kan.
Durland, William Pelton (J'31) (CFBHAM), Purchasing Engr., Worthington Pump & Mch. Corp., 27 Appleton St., Holyoke; *for mail*, 93 Bancroft Rd., Northampton, Mass.
Durley, Richard J. ('99), 3174 The Boulevard, Westmount, Que., Can.
Durnford, Errol R. (J'40), Sgt., Hdq., 41st Inf. Div., A.P.O. 41, c/o Postmaster, San Francisco, Calif.
Durrant, Oliver W. (J'40), 8552 S. Loomis Blvd., Chicago, Ill.
Dursin, Henry, Jr. ('18; '25; '35) (TSC), Secy., Mgr., Lafayette Worsted Spinning Co., Hamlet Ave., Woonsocket, R.I.
Dusenbury, William Henry ('43) (OSFLEK), Cons. Engr., W. H. & J. F. Dusenbury, 10 E. 40th St., New York 16, N.Y.
Dusinberre, George M. ('38) (SKE), Lt. Comdr., U.S.N., Retired; Va. Poly. Inst., Blacksburg, Va.
Dussel, Alexander Neil (J'42) (MOB), Lt. (j.g.), U.S.N., Mine Warfare & Engrg. Officer, YMS-80, F.P.O., New York, N.Y.
Dutcher, Frederick Harris ('39) (AEKOS), Asst. Prof. Mech. Engrg., Columbia Univ., Broadway & 116th St., New York 27, N.Y.
Dutcher, John E. ('22) (EFS), Mech. Engr., Schmidt, Garden & Erikson, 104 S. Michigan Ave., Chicago 20, Ill.
Dutton, Frederick Osbon ('34) (BCLJ), Supvr. of Inspc., Carbide & Carbon Chemicals Corp., 30 E. 42nd St., New York 17, N.Y.; *for mail*, 315 Cumberland Rd., South Orange, N.J.
Dutton, Henry P. ('30) (CM), Prof. Business Organization, Chmn., Indus. Engrg. Dept., Ill. Inst. of Tech., 3300 Federal St., Chicago 16; *for mail*, 2242 Pioneer Rd., Evanston, Ill.
Dutton, Robert Hunt (J'45) (BCM), Design Engr., Gen. Elec. Co., 5441 E. 14th St., Oakland 1; *for mail*, 1849 Monterey Rd., San Jose, Calif.
DuVivier, Charles Louis (J'45), U.S. Navy; 76 Sedgwick Ave., Darien, Conn.
Dvorak, John J. (J'35), Ch. Draftsman, Baytown Refinery, Humble Oil & Refining Co., Baytown; *for mail*, 6125 Washington Ave., Houston 7, Tex.
Dwight, Harold S. ('36), 1019 Redway Ave., Cincinnati, Ohio.
Dwight, Herbert V. (J'40) (ABC), 935 Work Dr., Akron 2, Ohio.
Dworetzky, Lawrence Herbert (J'44) (L), Jr. Engr., Transylvania Tanning Co., Brevard, N.C.
Dworzan, Joseph Stanley (J'42) (MACJBY), 1st Lt., Army Air Forces, Prod. & Procurement Engr., Hdq., Air Tech. Serv. Comd., TSBPAS, Wright Field, Dayton, Ohio.
Dwyer, John J. ('25) (SEFCLM), Sales Engr., M. W. Kellogg Co., 225 Broadway, New York 7; *for mail*, 1284 Carroll St., Brooklyn 13, N.Y.
Dwyer, Leo E. (J'41), Gen. Mgr., Charge of Design & Engrg., Codem Engrg., 1936 Shattuck Ave., Berkeley 4, Calif.
Dwyer, Paul Francis ('30; '45) (EK), Engr., Kellex Corp., 233 Broadway, New York; *for mail*, 18 Vermont Ave., White Plains, N.Y.
Dyck, Arnold William (J'45) (NMBCOA), Sr. Engr., Trans-Canada Air Lines; *for mail*, 6 Eastgate, Winnipeg, Man., Can.
Dye, Ira Willard ('13; '35) (CMWS), Sr. Manpower Consultant, War Manpower Comm., 1511—5th Ave., Seattle 1; *for mail*, 5950—32nd Ave. S., Seattle 8, Wash.
Dyer, Ralph Leighton ('31; '35) (EHS), Sales Engr., 812 Ins. Bldg., Seattle 4, Washington.
Dyer, Samuel, Jr. (J'43), Capt., U.S.A., 74th Serv. Gp., A.P.O. 149, c/o Postmaster, New York, N.Y.
Dyer, William E. S. ('27) (OCFSTY), Propr. Firm of William E. S. Dyer, Crosby Bldg., Buffalo 2; *for mail*, 800 West Ferry, Buffalo, N.Y.
Dyke, John Morley (J'43) (SKOHBE), Boiler, Heat Transfer Design Engrg., John Inglis Co. Ltd., Strachan Ave., Toronto, Ont., Can.
Dyke, Theodore Allen (J'42) (CLS), 10 Mitchell Pl., New York 17, N.Y.
Dykes, John Reid (J'41 (SKJFAB), Jr. Engr., Steam Div., Westinghouse Elec. Corp., Philadelphia, Pa.; *for mail*, 281 Lake St., Newark 4, N.J.
Dykhuizen, Horace Martin (J'44), Engr., Gen. Elec. Co., 2000 Taylor St.; *for mail*, 336 Killea St., Ft. Wayne, Ind.
Dym, Joseph Ben ('44) (BHJMWY), Sec. Engr. over Mfg. Engrg., Westinghouse Elec. Corp., Ardmore Blvd. & Braddock Rd., Pittsburgh 21; *for mail*, 6371 Douglas St., Pittsburgh 17, Pa.
Dyrenforth, William Phillip (J'43) (AES), Lt. (j.g.), U.S.N.R., LST-1140, F.P.O., San Francisco, Calif.; *for mail*, 2334 Bellaire St., Denver 7, Colo.
Dzurka, John Jospeh (J'43) (WSHBE), Ensign, U.S.N.R., Indus. Dept., Naval Shipyard, Philadelphia 12, Pa.; *for mail*, 532 E. 8th St., Brooklyn, N.Y.

E

Eaddy, Ernest Johnson ('42) (TCM), Engr. & Gen. Mgr., Textile Shops, Spartanburg, S.C.
Eadie, J. Keith (J'40) (BHL), Paper Mchy. Dept., Dominion Engrg. Works, Ltd., 1st Ave.; *for mail*, 4065 Broadway, Lachine, Que., Can.
Eadie, John C. ('17) (KLS), Eadie, Freund & Campbell, Cons. Engrs., 500—5th Ave., New York 18, N.Y.
Eagles, Burton Weller (J'45) (HJCESN), Engr., Proposition Dept., Babcock-Wilcox & Goldie McCulloch Ltd., Grand Ave.; *for mail*, 127 Concession St., Galt, Ont., Can.
Eagles, James Rowland (J'45) (CMS), Graduate Student Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail*, 203 Franklin Ave., Wilkinsburg, Pa.
Earhart, Joseph Stanley ('43) (KLHSC), Pres., Gen. Mgr., Preferred Equip., Inc., 4232 E. Whiteside St., Los Angeles 33, Calif.
Earl, Ralph ('44), 2125 Leonidas St., New Orleans 18, La.
Earl, Thomas Collins (J'42) (WCHOLS), Mech. Engr., Design & Constr., Union Bag & Paper Corp.; *for mail*, 506 Maupas Ave., Savannah, Ga.
Earle, Chester Reed ('44) (SFEZKH), Assoc. Editor, *Power Plant Engineering*, Tech. Publ. Co., 53 W. Jackson Blvd., Chicago 4, Ill.
Earle, Samuel Broadus ('05; '08; '14; F'39) (SEFKC), *Manager*, '37-'40, *Vice-President*, '40-'42; Dean, Sch. of Engrg., Clemson College, Clemson, S.C.
Early, Daniel Miller, Jr. (J'43), Lt., 27th Air Trans. Gp., A.P.O. 635, c/o Postmaster, New York, N.Y.

Eason, Clarence M. ('20) (ACMEN), Pres., Indus. Clutch Corp., 515 Frederick St.; *for mail*, 406 Windsor Dr., Waukesha, Wis.
East, Frank George ('30; '41) (NMJDBC), Design & Prod. Engr., Hamilton Gear & Mch. Co., 76 Van Horne St., Toronto 4; *for mail*, 64 Cliveden Ave., Toronto 9, Ont., Can.
East, Warren Errett ('19; '35) (FKS), Sales Engr., Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3, Ill.
Eastling, Harvey V. ('22; '30) (CDM), V.P., Pacific Div., Link-Belt Co., 400 Paul Ave., San Francisco; *for mail*, 1718 Easton Dr., Burlingame, Calif.
Eastman, Everett James (J'42) (BJE), Dow Chem. Co.; *for mail*, 602 Townsend, Midland, Mich.
Eastman, Fred Cecil ('28) (BHAYTK), Research Engr., United Shoe Mchy. Corp., Balch St., Beverly; *for mail*, 9 Pickwick Rd., Marblehead, Mass.
Easton, Elwood Merritt (J'43) (ABJNK), Design Engr., Propeller Div., Curtiss-Wright Corp., Route 6, Caldwell; *for mail*, 515 Mt. Prospect Ave., Newark 4, N.J.
Eastwood, Everett O. ('14; F'36) (KFSAC), *Manager*, '23-'26, *Vice-President*, '26-'28; Prof. Mech. Engrg., Head, Dept. of Mech. Engrg., Univ. of Wash., Seattle 5, Wash.
Eaton, Cecil L. ('30), Ch. Engr., Am. Associated Consultants, 250 Park Ave., New York, N.Y.; *for mail*, Wm. Penn Hotel, Pittsburgh, Pa.
Eaton, Edgar P., Jr. (J'44) (CEM), Sales Engr., Allis-Chalmers Mfg. Co., 1160 Pk. Sq. Bldg., Boston 16; *for mail*, 472 Broadway, Cambridge, Mass.
Eaton, George Curtis (J'25) (SFEABC), Asst. Supt. Prod., Boston Edison Co., 39 Boylston St., Boston 12, Mass.
Eaton, George Henry ('32; '35) (SOHFEK), Pres., Foskett & Bishop Co., Box 1593, 76 Blatchley Ave., New Haven, Conn.
Eaton, Herbert N. ('30), (BHK), Chief, Hyd. Lab. Sec., Natl. Bur. of Stands., Washington, D.C.; *for mail*, 3 E. Inverness Driveway, Chevy Chase, Md.
Eaton, Ian Duncan (J'44), 70 Gillett St., Hartford 5, Conn.
Eaton, Leon Schultz ('22) (FHNJKW), Visiting Prof. of Mech. Engrg., College of Engrg., River Campus, Univ. of Rochester, Rochester 3, N.Y.
Eaton, Paul B. ('22) (NCDL), *Manager*, '40-'42; Head, Mech. Engrg. Dept., Lafayette College; *for mail*, 4 East Campus, Easton, Pa.
Eaton, William Fay (J'45), Lt., U.S.N., LCI (L) 652, F.P.O., San Francisco; *for mail*, Apt. 12, 2124 McKinley Ave., Berkeley 3, Calif.
Eaton, Wyman ('26), Pres., Freyn Engrg. Co., 310 S. Michigan Ave.; *for mail*, 7340 Merrill Ave., Chicago, Ill.
Ebaugh, Newton C. ('32; '35; '35) (OLSHFK), Head, Mech. Engrg. Dept., Univ. of Fla., 101 Engrg. Bldg., Gainesville, Fla.
Ebdon, Hubert George ('40) (FS), Gen. Sales Mgr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Ebenbach, Robert (J'40) (RABM), Asst. Structures Engr., Ry. Engrg. Dept., Edw. G. Budd Mfg. Co., Red Lion Rd., Bustleton, Philadelphia; *for mail*, 235 Brookdale Ave., Glenside, Pa.
Eberhard, Walter Otto (J'46), 119-02—196th St., St. Albans, L.I., N.Y.
Eberhardt, Frank Edward ('07; '25) (NCBJMZ), Pres., Gen. Mgr., Newark Gear Cutting Mch. Co., Prospect St., Newark; *for mail*, 10 N. Ridgewood Rd., South Orange, N.J.
Eberhardt, Henry Ezra ('13; '21) (CJM), V.P., Gould & Eberhardt, Inc., 433 Fabyan Pl., Irvington 11; *for mail*, 2 Mountain View Terrace, Maplewood, N.J.
Eberhardt, John Dennett ('26) (CMWOJD), Indus. Engr., Ernst & Ernst, 50 Congress St., Boston; *for mail*, 248 Gray St., Arlington, Mass.
Eberhardt, U. Seth ('25) (BNC), Cons. Engr., 26 Oberlin St., Maplewood, N.J.
Eberhardt, Walter William (J'46), 1634 Northwood Dr., Cincinnati 16, Ohio.
Eberhardt, William Charles (J'40) (CHJ), Research Engr., Gould & Eberhardt, Inc., 433 Fabyan Pl., Irvington 11; *for mail*, 594 Duquesne Terrace, Union, N.J.
Eberhart, John Muth (J'46) (ABE), 2012 Forest Park Ave., Baltimore 7, Md.
Eberle, Fred Henry ('20; '35) (DEHKS), Asst. Supvg. Engr., Mech. Engrg. Div., Stand. Oil Devel. Co., Box 37, Elizabeth; *for mail*, 940 Maurice Ave., Rahway, N.J.
Eberle, James William (J'45) (BJSNKM), Turbine Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 245½ Mohawk Ave., Scotia 2, N.Y.
Eberth, William Frederick ('45) (CLDMH), Engr. Examiner, Reconstruction Finance Corp., 811 Vermont Ave., N.W., Washington 25, D.C.; *for mail*, Apt. C-422, 3111—20th St. N., Arlington, Va.
Eblen, Wm. F. ('25) (FES), Shaw & Eblen; *for mail*, 5619 Sherman St., Houston 11, Tex.
Eby, Earl E. ('18) (EMJBLS), Mgr., Power & Indus. Products, Gen. Motors Overseas Opera., 1775 Broadway, New York 19, N.Y.
Echelson, George (J'39) (ZLHMJE), Instrument Engr., Lago Oil & Transport Co., Ltd., Box 456, Aruba, Curaçao, N.W.I.
Echeverria, R. James (J'46) (CP), 956 Masson Ave., San Bruno, Calif.
Eck, Alexius Emanuel ('29) (BGSJ), V.P., Cent. Iron & Steel Co., S. Front St., Harrisburg; *for mail*, 1404 Walnut St., Camp Hill, Pa.
Eckart, William Rankine ('04) (LKHSBE), Mech. Engr., C. F. Braun & Co., 1000 S. Fremont Ave., Alhambra; *for mail*, 1525 Oak Grove Ave., San Marino 5, Calif.
Eckberg, Herbert Frederic (J'37) (BK), Capt., U.S.N., Commanding U.S.S. *Anne Arundel*, F.P.O., New York, N.Y.; *for mail*, 2108 Lake Ave., Baltimore 18, Md.
Eckerman, Edward Henry (J'39) (SNO), Instr., Mech. Engrg., Rose Poly. Inst., R.R. 5, Terre Haute, Ind.
Eckert, Arthur C. ('13; '19) (BEM), Propr., Firm of Arthur C. Eckert, 801 Title Guaranty Bldg., St. Louis, Mo.
Eckert, Henry R. ('21; '31), Propr., Hudson Refrig. Mch. Co., 144 Bayview Ave., Jersey City, N.J.
Eckert, Joseph Sigmund ('28) (CDFLSZ), Supt., Buckeye Sugar Co.; *for mail*, Box 207, 388 E. 3rd St., Ottawa, Ohio.
Eckert, Otto E. ('42), 919 W. Michigan Ave., Lansing, Mich.
Eckhardt, Carl John, Jr. ('29; '37) (SFEKHG), Prof. Mech. Engrg., Supt. Utilities, The Univ. of Tex., Austin 12, Tex.
Eckhardt, Floyd S. (J'36) (JDF), Ch. Engr., Bethlehem Steel Co., 1 Hamburg Turnpike, Lackawanna, N.Y.
Eckhart, William Henry (J'43) (B), Assoc. Mech. Engr., Naval Shipyard, Portsmouth, N.H.; *for mail*, 2 Howard St., Kittery, Me.
Ecklund, Charles (M'39) (CMNBJD), Asst. Supt., Dictaphone Corp., 375 Howard Ave., Bridgeport 5; *for mail*, 187 Buena Vista Rd., Bridgeport 4, Conn.
Eckman, Donald Preston (J'38) (ZBHK), Devel. Engr., Brown Instrument Co., 4497 Wayne Ave.; *for mail*, 5339 Greene St., Philadelphia 44, Pa.
Eckstrom, Albert Wallace ('35; '38) (CDHKLS), Asst. Ch. Engr., Evaporator Dept., "Buflovak" Equip. Div., Blaw-Knox Co., 1543 Fillmore Ave., Buffalo; *for mail*, 46 Kingsley St., Buffalo 8, N.Y.
Eddie, Charles Ernest Shepard ('44) (FZSCMO), Asst. Supt., Steam & Fuel Dept., Great Lakes Steel Corp., Ecorse; *for mail*, 1637—2nd St., Wyandotte, Mich.
Eddison, William Barton ('13; '21; '35; F'43) (BOZ), Cons. Mech. Engr., Ardsley-on-Hudson, N.Y.
Eddy, Harrison Prescott, Jr. ('27) (O), Partner, Metcalf & Eddy, 1300 Statler Bldg., Boston 16, Mass.
Eddy, James Walter (J'44), Asst. Design Engr., Eddy Shipbldg. Corp., Harrison at Cass, Bay City, Mich.
Eddy, Willard T. (J'33) (CFK), 530 Park Ave., Orange, N.J.
Edel, Walter Lester ('21; '26; '29) (SKBAFO), Dir., Engrg. & Research, Clayton & Lambert Mfg. Co., 1701 Dixie Highway, Louisville 10, Ky.
Edelen, Charles J. (J'37) (EF), 1410 Pipkin St., Beaumont, Tex.
Edelman, Beril ('40) (COMD), Ch., Contract Standardization & Coordinating Div., West. Elec. Co., Inc., 120 Broadway, New York 5; *for mail*, 580 E. 8th St., Brooklyn 18, N.Y.
Edelstein, Burton Herman (J'45) (KMC), Rating Engr., Graham Mfg. Co., 415 Lexington Ave., New York 17; *for mail*, 170-49 Cedarcroft Rd., Jamaica 3, N.Y.
Eden, Fitzroy L. ('23), 160 Paulin Blvd., Leonia, N.J.
Eder, James P. (J'34) (CWZ), Research Engr., Sperry Gyroscope Co., Inc., Great Neck; *for mail*, 29 Washington Sq., New York 11, N.Y.
Edgar, Alonzo John (A'43), Tech. Adviser, Gray Iron Founders Soc., Inc., 1201—19th St., N.W., Washington 6, D.C.
Edgar, Kenneth Kirk (J'40) (OCM), Sr. Engr., Rath & Strong, Inc., 80 Federal St., Boston 10; *for mail*, 59 Oxford Rd., Newton Centre 59, Mass.
Edge, Maurice Peterman (J'39) (ZFS), Sales & Serv. Engr., Paul B. Huyette Co. Inc., 401 N. Broad St., Philadelphia 8; *for mail*, 465 Flamingo St., Philadelphia 28, Pa.
Edge, Walter Clifford ('45) (SZCFOL), V.P., Paul B. Huyette Co. Inc., 401 N. Broad St., Philadelphia 8, Pa.
Edgell, Albert B. ('39), Power Div., Studebaker Corp., South Bend, Ind.
Edgerly, Robert Moffatt (J'45) (AENBHC), Flight Test Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail*, 544 Front St., Hempstead, L.I., N.Y.
Edick, George Wallace, Jr. (J'40), Capt., Hdq. Btry., Div. Arty., 16th AD., A.P.O. 412, c/o Postmaster, New York, N.Y.
Edlefsen, Robert P. (J'42), 1040 Weeks St., Palo Alto, Calif.
Edmiston, Maurice Oliver ('25; '35) (ORFE), Capt., Chem. Warfare Serv., A.P.O. 9224, c/o Postmaster, New York, N.Y.; *for mail*, 1960 Ivy St., Denver 7, Colo.
Edmondson, Don Elton (J'45), U.S. Navy; *for mail*, 2919 Maple Springs Blvd., Dallas, Tex.
Edmunds, Robert (J'42), 2456 Grunewald St., Blue Island, Ill.
Edris, Clair James (J'41) (MJDS), Tool Designer, Elliott Co., N. 4th St.; *for mail*, 917 Gaskill Ave., Jeannette, Pa.
Edwards, Albert B. ('30; '35) (CH), 806 Calif. State Bldg., Los Angeles 12, Calif.
Edwards, Arthur (J'40) (BEFKLS), Engr., Mech. Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Edwards, Austin Southwick, Jr. (J'42) (ABFNYK), Sr. Engr., Wright Aero. Corp., Beckwith Ave., Paterson 3; *for mail*, 329—19th Ave., Paterson 4, N.J.
Edwards, Clive L. ('30; '31; '35) (LMC), Elec. Engr., E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Edwards, Earl David ('25; '35) (FS), Supt., Elec. Dept., Cent. Ill. Light Co., 316 S. Jefferson St., Peoria 2, Ill.
Edwards, Fordyce A. (J'38) (FGNRYZ), Ch., Military Publications Branch, Office of the Ch. of Ord., The Pentagon, Washington 25, D.C.
Edwards, Frank William ('41) (HBCKSA), Prin. Engr., Chief Design Branch, U.S. Engr. Dist., 1709 Jackson, Omaha 2; *for mail*, 2332 N. 50th Ave., Omaha 4, Neb.
Edwards, G. E. (J'39), Comdr., U.S.N.R.; Box 117, Groton, Conn.
Edwards, Gilbert Middleton ('44) (CDLMY), Supt., Moulded Goods, Raybestos Manhattan Co., Whillett St., Passaic; *for mail*, 368 Hilltop Ave., Leonia, N.J.
Edwards, Harry D. ('18) (ESFDKY), Ch. Works Engr., Linde Air Products Co., 30 E. 42nd St., New York 17, N.Y.
Edwards, Hilbert Ephraim ('45) (BNMJCL), Sec. Engr., Supvr., Laundry Equip. Engrg. Dept., Westinghouse Elec. Corp., 246 E. 4th St.; *for mail*, 86 Western Ave., Mansfield, Ohio.
Edwards, Howard Berryman (J'36) (BAYJ), Mech. Engr., Nat. Advis. Comm. for Aero., Langley Field; *for mail*, Box 237, Hampton, Va.
Edwards, Joseph Arthur, Jr. (J'42) (ANLBCJ), Lt. (j.g.), U.S.N.R., Test Engr., Aircraft Eng. Lab., Naval Exper. Sta., Naval Air Matl. Center, Naval Base Sta., Philadelphia, Pa.
Edwards, Minot R. ('42) (MOJ), Treas., Holyoke Heater Co., 54 Waltham Ave., Springfield 9; *for mail*, 1903 Westfield St., West Springfield, Mass.
Edwards, Robert William (J'40) (CJ), Instrument Engr., Bausch & Lomb Optical Co., St. Paul St., Rochester; *for mail*, 1008 S. Goodman St., Rochester 4, N.Y.
Edwards, William Rowland (J'38) (BEJKLY), Jr. Engr., Humble Oil & Refining Co.; *for mail*, Box 327, Baytown, Tex.
Edwards, William Wesley ('13; '17; '35) (SKELFW), Lecturer, Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39; *for mail*, 16 Regent St., West Newton 65, Mass.
Effgen, Hans (J'45) (NJMC), Ensign, U.S.N., Educational Officer, Naval Oper. Base, Buckner Bay, Navy 3256, F.P.O., San Francisco, Calif.; *for mail*, 41 Riverview St., Dedham, Mass.
Effross, Max P. ('35) (LDBSKZ), Plant Mgr., Max Ams Inc., 376 Greenwich St., New York 13; *for mail*, 201 Clinton Ave., Brooklyn 5, N.Y.
Egan, Edward Francis (J'40) (CAK), 1st Lt., 499th Bomb Sqd., 354th Bomb Gp., A.P.O. 503, c/o Postmaster, San Francisco, Calif.
Egan, Kyran William ('28; '35) (DJWSNC), Safety Inspr., Constr., State Dept. of Labor, 350—5th Ave., New York 1; *for mail*, Box 374, R.D. 1, Schenectady, N.Y.
Egbert, Charles Coggill ('06) (OHSEN), Cons. Engr., 404 Jefferson Ave., Niagara Falls, N.Y.
Egbert, Harry Eugene (J'39) (CAM), Plant Engr., Boeing Aircraft Co., Box 3107, Seattle 14; *for mail*, 3246 E. 98th, Seattle 5, Wash.
Eger, George William, Jr. (J'40) (EFNSJB), 148 Prospect St., Leonia, N.J.
Eggebrecht, Edward Theodore, Jr. (J'40) (ACDJMN), Lt. (j.g.), U.S.N.R., F.P.O., New York, N.Y.
Eggert, Erwin H. ('29; '36) (DFL), Plant Engr., Procter & Gamble Mfg. Co., 1000 Haubert St., Baltimore; *for mail*, 802 Ridgeleigh Rd., Baltimore 12, Md.
Eggleston, Herbert Lyman ('28) (CLEFKS), *Manager*, '41-'44; Mgr., Gas Dept., Gen. Petroleum Corp. of Calif., Higgins Bldg., 108 W. 2nd St., Los Angeles 12; *for mail*, 1017 Cumberland Rd., Glendale 2, Calif.
Egilsrud, Fridtjof S. ('24; '25; '35) (SEFKBA), Assoc. Prof., Pratt Inst., 215 Ryerson St., Brooklyn, N.Y.
Egleston, Marvin Plimpton (J'33) (SZHKBD), Draftsman, A. A. Maycock Co., 234—7th Ave.; *for mail*, 136 E. South Temple, Salt Lake City, Utah.
Egleston, Oliver J. ('16) (BCDEFI), c/o Longwood Towers, 20 Chapel St., Brookline 46, Mass.
Egli, Huldreich ('41) (EFLOS), Egli & Gompf, Inc., 425 St. Paul Pl., Baltimore 2; *for mail*, 407 S. Rolling Rd., Cantonsville 28, Md.
Egloff, Gustav ('44) (EFKY), Petroleum Technologist, Universal Oil Products Co., 310 S. Michigan Ave., Chicago 4, Ill.
Egry, C. Robert ('40) (CMNDBK), Assoc. Prof. Mech. Engrg., The Univ. of Notre Dame du Lac, Notre Dame; *for mail*, 1017 E. Washington Ave., South Bend 17, Ind.
Ehbrecht, Adolph ('30; '36) (CMN), V.P., Gries Reproducer Corp., 780 E. 133rd St., New York 54; *for mail*, 186 Weyman Ave., New Rochelle, N.Y.

Ehlers, Henry Edward ('12) (SOFBHC), V.P., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2; *for mail*, 2031 Locust St., Philadelphia 3, Pa.
Ehlinger, Arnold Henry (J'41) (BNSLA), Asst. Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 3416 W. State St., Milwaukee 8, Wis.
Ehmann, Roy Leon ('38) (ACKS), Sales Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail*, 470 Willow Dr., Pittsburgh 16, Pa.
Ehrenkranz, Theodore Eugene (J'45), Merchant Mar.; *for mail*, 307 E. 77th St., New York 21, N.Y.
Ehrhart, George Weber ('33; '45) (KHJ), Partner, Ehrhart & Arthur, 417 S. Hill St., Los Angeles 13, Calif.
Ehrich, Louis S., Jr. (J'37) (HFKJBE), Engr., Indus. Div., Webster Elec. Co., Clark & De Koven St.; *for mail*, Y.M.C.A., Racine, Wis.
Ehrman, Bruno, Jr. (J'37) (MCJ), Prod. Engr., Keuffel & Esser Co., 300 Adams St., Hoboken; *for mail*, 145 N. Leswing Ave., Rochelle Park, N.J.
Eiben, Lawrence A. (J'41) (CHJ), Gen. Mgr., No. Blower Co., 6409 Barberton Ave., Cleveland, Ohio.
Eibsen, Louis J. ('19) (DEH), Mason & Hanger Co., Inc., 500—5th Ave., New York 18, N.Y.; *for mail*, 793 Fairview Lane, Cliffside Park 8, N.J.
Eich, Clarence Charles (J'46) (BHS), Seaman, 2nd Class, 149 Ivanhoe St., S.W., Washington 20, D.C.
Eichstaedt, Alvin William (J'40) (BOYHAL), Mch. Designer, U.S. Rubber Co., 6600 E. Jefferson Ave.; *for mail*, 20017 Schaenherr Rd., Detroit, Mich.
Eigenbrot, John L. ('21) (FSEK), Asst. to Pres., Gen. Coal Co., 123 S. Broad St., Philadelphia 9, Pa.
Einfeldt, Charles Louis ('29; '35) (SKCHB), Piping Design Engr., Detroit Edison Co., 2000—2nd Ave.; *for mail*, 5347 Allendale Ave., Detroit 4, Mich.
Einig, Alvin Bartholomew ('12; '21; '35) (CNMJ), Gen. Mgr., Motch & Merryweather Mchy. Co., 715 Penton Bldg., Cleveland 13; *for mail*, 2592 Dartmoor Rd., Cleveland Heights 18, Ohio.
Einstein, Sol ('13) (BCM), V.P., Cincinnati Milling Mch. Co., Oakley, Cincinnati, Ohio.
Einwaechter, Fred Henry, Jr. ('44) (BEFNRS), Ch. Engr., M. P. Equip., Baltimore & Ohio R.R. Co., Pratt St. & Arlington Ave., Baltimore 23; *for mail*, 1215 Circle Dr., Halethorpe 27, Md.
Eisele, Paul Theodore (J'42) (BNJZ), Mech. Engr., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3; *for mail*, 3474 W. 145th St., Cleveland 11, Ohio.
Eisenwinter, Edward Ernest ('45) (CDHJNS), Mech. Supt., Am. Brass Co., Waterbury; *for mail*, 104 Hamilton Ave., Watertown, Conn.
Eiskamp, Edward ('41) (JAM), 8735—109th St., Richmond Hill, N.Y.
Eisler, Chas. ('17; '19), Mech. Engr., Prop., Eisler Engrg. Co., 750 S. 13th St., Newark 3, N.J.
Eisler, Charles, Jr. (J'40) (CBN), Ch. Engr., Eisler Engrg. Co., 750 S. 13th St., Newark 3; *for mail*, 16 Hoffman St., Maplewood, N.J.
Eisnor, Judson B. (J'40) (GMW), Jr. Engr., New England Tank & Tower Co., 85 Tileston St.; *for mail*, 36 Waverly Ave., Everett, Mass.
Eister, William D. (J'34) (OAC), Lt. Col., U.S.A., Pacific Air Comd., Ft. McKinley, Manila, P.I., A.P.O. 925, c/o Postmaster, San Francisco, Calif.
Eitel, William Walter, Jr. (J'44) (SCN), Technician, 5th Grade, Prisoner of War Guard 6, 6th Bn., 1322nd Serv. Comd. Unit, Ft. Meade; *for mail*, 410 Dunkirk Rd., Baltimore 12, Md.
Ek, Alma ('17; '22), Box 541, Route 1, Merced, Calif.
Ek, Gustaf C. ('22; '35), Traveling Engr., Great West. Sugar Co., Sugar Bldg.; *for mail*, 1550 Magnolia St., Denver, Colo.
Ekholm, Carl Roger (J'43) (ABEN), Designer, Charge Engrg. & Devel., Aetna Mfg. Co., 250 Chicago Ave., Oak Park; *for mail*, 5344 N. Ashland Ave., Chicago 40, Ill.
Eklund, Henrik Jarl ('45) (NKMCD), Ch. Engr., D. J. Murray Mfg. Co., 3rd St.; *for mail*, 1506½—6th St., Wausau, Wis.
Eklund, Joel ('25) (SFKB), Steam Power Plant Engr., U.S. Rubber Co., 1213—6th Ave., New York; *for mail*, 54 Mountain Ave., New Rochelle, N.Y.
Eklund, Thor I. (J'42) (NBHA), Mech. Engr., Design, Naval Aircraft Factory, Naval Base; *for mail*, 207 S. 42nd St., Philadelphia 4, Pa.
Eksergian, C. Levon ('24; '34) (BJRN), Ch. Engr., Budd Wheel Co., 12141 Charlevoix Ave., Detroit 14, Mich.
Eksergian, Rupen ('21; F'40), *Worcester Reed Warner Medallist*, '39; Edw. G. Budd Mfg. Co., Philadelphia; *for mail*, Rose Tree Rd., Media, Pa.
Eksten, Carl Edward, Jr. (J'44) (BYZJLO), Project Engr., Taylor Instrument Cos., 95 Ames St., Rochester; *for mail*, Reeves Rd., Henrietta, N.Y.
Ekstrom, Carl William (J'43) (CMNKLO), Pres., Custom-Bilt, Inc., 909 Washington St., Mendota, Ill.
Elam, Emmett W. (J'43) (ABJ), Whitehaven, Tenn.
Eland, Frank H. ('30; '35) (DHJNSB), Ch. Engr., Dodge Mfg. Div., United Steel Corp. Ltd., 58 Pelham Ave., Toronto 9, Ont., Can.
Elbel, Roy E. (J'42), Jr. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail*, 63 Pratt St., Glastonbury, Conn.
Elder, Harris James (J'43) (FAL), Jr. Mech. Engr., Phillips Petroleum Co.; *for mail*, 610½ Keeler Ave., Bartlesville, Okla.
Elder, John D. ('28), 200 Montrose Ave., Catonsville 28, Md.
Elder, John Thomas (J'42) (SYZ), Serv. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio; *for mail*, 1226 Avon Ave., S.W., Atlanta, Ga.
Elder, Lorne Cavell (J'42) (BJN), Ch. Engr., Wallace Barnes Co. Ltd., 274 Sherman St. N., Hamilton, Ont., Can.
Elder, Norman James (J'41) (ABM), 285 Lloyd St., Akron 1, Ohio.
Eldert, John DeBevoise ('31; '38) (NTDL), V.P., Engr., Mch. Parts Corp., 271 Washington St., Providence 3, R.I.
Eldred, Byron E. ('99; '03), High Acres, Lime Rock, Conn.
Eldridge, Charles Douglas (J'45) (CDM), Indus. Engr., William L. Hoge & Co., Heyburn Bldg., Louisville; *for mail*, 112 Cambridge Dr., Louisville 8, Ky.
Eley, Robert V. ('23; '27) (DHMR), Mchy. Buyer, Ford Motor Co., Dearborn; *for mail*, 17315 Northlawn Ave., Detroit 21, Mich.
Elfors, Russell Albert (J'44) (AMJD), Signal Corps, Co. G, 34th Trng. Bn., B.I.T.G., Army Serv. Forces Trng. Center, Camp Crowder, Mo.; *for mail*, Box 186, Stromsburg, Neb.
Elfring, John B. ('20; '35) (BMC), Ch. Tool Engr., Cincinnati Milling Mch. Co., Marburg Ave., Cincinnati 9, Ohio.
Elizondo, Hector Raoul (J'43) (LMCFKE), Asst. Engr., Mills & Boilers, Hershey Corp.; *for mail*, Hotel Hershey, Central Hershey, Cuba.
Elkind, Mortimer Murray (J'44) (MNCBOA), Ensign, U.S.N.R., Communications & Electronics Officer, LST-605, F.P.O., San Francisco, Calif.; *for mail*, 2153 E. 23rd St., Brooklyn 29, N.Y.
Elkins, Douglas A. (J'38) (NMBJCD), Mech. Engr., U.S. Bur. of Mines, 1600 E. 1st South, Salt Lake City 1; *for mail*, 1875 E. 21st South, Salt Lake City 5, Utah.
Elkus, James H. (J'33) (ACDM), Mgr., Methods & Procedure Dept., Blaw-Knox Co., 1521 Farmers Bank Bldg., Pittsburgh; *for mail*, 4014—5th Ave., Pittsburgh 13, Pa.
Ellenberger, William Joseph ('34; '41) (CSER), Maj., Sig. Corps, Asst. Exec., U.S. Staff, London Munitions Assignment Bd., c/o Military Attache, U.S. Embassy, London, England; *for mail*, 6524 Luzon Ave., N.W., Washington 12, D.C.
Ellenwood, Frank Oakes ('13) (SEBAFK), Prof. Engrg., Head, Heat-Power Engrg. Dept., Cornell Univ.; *for mail*, 111 Harvard Pl., Ithaca, N.Y.
Ellicott, Charles Remington ('16) (RC), V.P., Westinghouse Air Brake Co., 350—5th Ave., New York 1, N.Y.
Ellingwood, Charles R. (J'43) (YC), Mold Designer, Mack Molding Co.; *for mail*, Box 389, Arlington, Vt.
Elliot, Augustus Hull ('14; '35) (RJD), V.P., So. Wheel Div., Am. Brake Shoe Co., 230 Park Ave., New York 17, N.Y.
Elliot, Walter Robertson (J'41) (KCSEF), Buffalo Forge Co., 490 Broadway, Buffalo 5, N.Y.
Elliott, Ben George ('11; '16; '23) (FSERK), Prof. Mech. Engrg., Univ. of Wis.; *for mail*, 2302 Commonwealth Ave., Madison 5, Wis.
Elliott, Clifton M. (J'38) (AHM), Research Engr., Chrysler Corp., 12800 Oakland Ave.; *for mail*, 650 W. Boston Blvd., Detroit 2, Mich.
Elliott, E. G. ('07), 5033—4th Ave., Los Angeles, Calif.
Elliott, Edward ('21) (RBACDN), Asst. Prin. Engr., Pullman-Stand. Car Mfg. Co., 27 Mountain St. W.; *for mail*, 6 Devens Rd., Worcester 6, Mass.
Elliott, Edward, Jr. ('36; '46) (JAFDON), Dist. Sales Mgr., Pressed Steel Tank Co., 225 Hanna Bldg., Cleveland 15, Ohio.
Elliott, Gilbert Drury ('45) (CEFJKS), Dir., John Thompson Water Tube Boilers Ltd.; *for mail*, 9 Leighton Rd., Penn, Wolverhampton, England.
Elliott, James Edward, Jr. (J'40) (JMS), 304 W. Howard Ave., Biloxi, Miss.
Elliott, John F. (J'42), Jr. Engr., E. I. du Pont de Nemours & Co., Belle; *for mail*, 1533 Quarrier St., Charleston, W.Va.
Elliott, John Milton (J'41) (ACKF), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8, Conn.; *for mail*, 42 Fairview Ave., West Warwick, R.I.
Elliott, Louis ('23) (SEFKHO), Cons. Mech. Engr., Ebasco Services, Inc., 2 Rector St., New York 6, N.Y.
Elliott, Milton D. (J'40), Jr. Gear Engr., Dodge Plant, Chrysler Corp.; *for mail*, 17162 Sioux, Detroit, Mich.
Elliott, Newton Ross ('17; '35) (BG), Engr., Draftsman, City of South Portland, 25 Cottage Rd.; *for mail*, 41 Lowell St., South Portland 7, Me.
Elliott, Orrin Matthews ('44) (SKFZHB), Water Cond. Engr., Sun Oil Co., 1608 Walnut St., Philadelphia 3; *for mail*, 85 Hemlock Rd., Media, Pa.
Ellis, Arthur L. ('31) (CMALJH), Mgr., Fed. & Mar. Dept., Waterbury Tool Div., Vickers Inc., E. Aurora St., Waterbury 91; *for mail*, Kelsey Hill Rd., Deep River, Conn.
Ellis, Benjamin Henry ('43) (OSCHEL), Ch. Mech. Engr., Gahagan Constr. Corp., 90 Broad St., New York 4; *for mail*, 112-06 202nd St., Hollis 11, N.Y.
Ellis, Bernard (J'42), Lt., Army Air Field, Myrtle Beach, S.C.
Ellis, Cedric Milton ('43) (SZFE), Sales Engr., Eshelman & Potter, 1116 Marton Bldg., Birmingham 3, Ala.
Ellis, Charles E. (J'29), 3607 Kingsboro Rd., N.W., Atlanta, Ga.
Ellis, Daniel Sayer ('29; '35) (RECFMK), *Manager*, '44-'45, *Director-at-Large*, '45-'47; V.P., Charge Mfr., Lima Loco. Works, Inc., S. Main St., Lima, Ohio.
Ellis, Frank Ashford ('39) (CMW), Owner, Frank A. Ellis & Assoc., 231 St. James St. W., Montreal, Que., Can.
Ellis, Frederic Robert ('27) (SKF), Supt. Engrg., Harvard Univ., Cambridge; *for mail*, 131 Beacon St., Hyde Park 36, Mass.
Ellis, Gershom Porter ('20; '25) (SFKCM), Dist. Mgr., Combustion Engrg. Co., Inc., 600 Gwynne Bldg., Cincinnati 2, Ohio.
Ellis, H. Alfred ('39), Box 63; *for mail*, 1332 W. Washington Blvd., Ft. Wayne, Ind.
Ellis, Herbert Bailey (J'38) (KHAJ), Research Engr., West. Research Div., Rheem Mfg. Co., 170 California St., Pasadena 5; *for mail*, Box 367E, Route 1, LaCanada, Calif.
Ellis, James ('21; '31; F'45) (CSNKTY), Gen. Supt., Tenn. Eastman Corp.; *for mail*, 1708 Orchard Lane, Kingsport, Tenn.
Ellis, John G. E. ('44) (CJL), Resident Engr., Shelton Div., Rayonier Inc.; *for mail*, 108 E. 17th Ave., Olympia, Wash.
Ellis, Owen William ('30) (JBL), Dir., Dept. Engrg. & Metal., Ont. Research Foundation, 43 Queen's Pk., Toronto, Ont., Can.
Ellis, Sanford (J'44), 32-27—32nd St., Long Island City 3, N.Y.
Ellis, Weldon Thompson ('08; '12) (SFKL), Prof. Power & Fuel Engrg., Supvr., Dept. of Htg. & Power, Va. Poly. Inst., Blacksburg, Va.
Ellis, William C. ('38) (CLM), Supt., Kingsport Fdy. & Mfg. Corp., Sullivan St., Kingsport, Tenn.
Ellison, Lewis M. ('11; '13), Pres., Ellison Draft Gage Co., 214 W. Kinzie St., Chicago, Ill.
Ellman, Louis ('21; '26) (EFKS), 2338 Oliver Bldg., Pittsburgh 22, Pa.
Ellson, H. S. ('22; '35), Ch. Engr., Speer Carbon Co., Internatl. Graphite & Electrode Corp., St. Marys, Pa.
Ellsworth, George Eric (J'38) (CBKNOH), V.P., Asst. Gen. Mgr., Toronto Iron Works Ltd., 629 Eastern Ave., Toronto 8, Ont., Can.
Elly, Robert Duncan (J'32) (CGMFSZ), Indus. Engr., Tenn. Coal, Iron & R.R. Co., Birmingham; *for mail*, 1509—29th St., Fairview Station, Birmingham 8, Ala.
Elmendorf, William T. E. (J'39) (ZEKL), Chem. Engrg. & Metal. Lab., E. I. du Pont de Nemours & Co., Wilmington 98; *for mail*, 1403 Riverview Ave., Wilmington 73, Del.
Elmer, Lloyd Armstrong ('21; '36) (BNJASE), Design Engr., Bell Tel. Labs., Inc., Murray Hill, N.J.
Elmer, William ('96; '13; F'37) (RSD), *Manager*, '25-'28, *Vice-President*, '28-'30; Retired; Box 393, Bay Head, N.J.
Elmes, Charles Warren ('04) (HY), Rm. 1828, 231 S. La Salle St., Chicago 4, Ill.
Elmsley, C. M. R. (J'41) (CMSOND), G.S.O. II, Directorate of Military Trng., N.D.H.Q., Ottawa, Ont., Can.
Elmzen, Harry Richard (J'41) (KMCHZD), Engr., Drying Systems, Inc., 1800 Foster Ave., Chicago 40, Ill.
Elo, Leo Gust (J'42) (ANBEFJ), Mech. Engr., Puget Sound Naval Shipyard, Bremerton; *for mail*, 2645 W. 59th St., Seattle 7, Wash.
Elrod, Harold Glenn, Jr. (J'43) (BKEFHS), Engr., Clayton & Lambert Mfg. Co., 11111 French Rd., Detroit; *for mail*, 55 E. Greendale St., Detroit 3, Mich.
Elsas, Norman E. ('21; '33) (GTDA), Pres., Fulton Bag & Cotton Mills, Box 1726, Atlanta 1, Ga.
Else, Harry D. ('45) (BATNKC), Departmental Mech. Engr., Westinghouse Elec. Corp.; *for mail*, 836 W. Spring St., Lima, Ohio.
Elsey, Warren R. ('41) (CMR), Asst. V.P., Pa. R.R. Co., 1752 Broad St. Sta. Bldg., Philadelphia 4, Pa.
Elsner, William H. ('20; '35), Mech. Engr., Great No. Ry., 175 E. 4th St.; *for mail*, 1456 Osceola Ave., St. Paul, Minn.
Elsworth, John Van Varick ('43; '45) (RSKW), Mech. Engr., N.Y. Air Brake Co., Starbuck Ave.; *for mail*, 524 Holcomb St., Watertown, N.Y.
Elsworth, Robert Merton (J'40) (ODBH), Supvg. Engr., Service Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Elton, Richard Leech, Jr. (J'45) (NMAJ), Engr., Charge of Maint., Shell Chem. Corp., Houston; *for mail*, 1109 S. Spooner St., Pasadena, Tex.
Elwell, Frank D. ('19; '25) (COSLMF), Retired; 2250 Nolen Dr., Flint 4, Mich.

Elwell, Richard Derby ('27; '34; '35), McKinsey & Co., 60 E. 42nd St., New York, N.Y.
Ely, Allen J. ('20; '25; '27) (ACDJOK), Ch. Constr. Engr., Stand. Oil Devel. Co., Box 37, Elizabeth, N.J.
Ely, Edwin William ('22; '35) (CMD), Chief, Div. of Simplified Practice, U.S. Dept. of Commerce, Natl. Bur. of Stands., Conn. Ave. & Van Ness St., Washington 25; *for mail,* 1725 Juniper St., N.W., Washington 12, D.C.
Ely, Frederic G. ('41) (FSKRZ), Research Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Ely, George William (J'42) (ABF), Design Engr., Bendix Aviation Corp.; *for mail,* 1221 N. Elmer St., South Bend 16, Ind.
Ely, Robert Eric (J'44) (KL), Engrg. Services, Rm. 414, Brower Bldg., Bakersfield, Calif.
Ely, Robert Grosvenor ('44) (CESDML), Asst. Gen. Power Rep., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Ely, Sumner Boyer ('96; '01) (FRK), Supt., Bur. of Smoke Prevention, City of Pittsburgh, City-County Bldg., Pittsburgh; *for mail,* 520 Roslyn Pl., Pittsburgh 6, Pa.
Embach, Edward Louis (J'41), 8552 Quincy, Detroit 4, Mich.
Embree, Ned Dexter (J'42) (AML), Lab. Engr., Gen. Elec. Co., 40 Federal St., West Lynn 3; *for mail,* 12 Rockingham St., Lynn 2, Mass.
Emeny, Frederick J. ('99; '21) (H), V.P., Deming Co.; *for mail,* 575 Highland Ave., Salem, Ohio.
Emerick, Robert Henderson ('43) (FKS), Capt., U.S.N.R., Repair & Conversion Officer, Naval Shipyard, Charleston, S.C.
Emerson, C. L. ('27), V.P., Ch. Engr., Robert & Co., Inc., 706 Bona Allen Bldg., Atlanta, Ga.
Emery, Hervey Ross, Jr. (J'42), 6810—46th Ave., N.E., Seattle 5, Wash.
Emery, Lorentz Densmore (J'43), Missionary, (under appointment to Colombia, S.A.), Bd. of Foreign Missions of Presbyterians in U.S., 156—5th Ave., New York; *for mail,* 209 Union St., Schenectady, N.Y.
Emhardt, Fred W. ('30; '42) (KSLMJE), Engr., Struthers-Wells Co.; *for mail,* 10—4th Ave., Warren, Pa.
Emin, G. Herbert (J'43) (ZKSO), Sales Engr., G. H. Emin & Associates, 616 S. Michigan Ave., Chicago 5, Ill.
Emley, Warren E., Jr. (J'39) (AEKOZ), Mech. Engr., Research, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland, Ohio.
Emmet, Herman Leroy ('31) (C), Works Mgr., Gen. Elec. Co., 2901 E. Lake Rd., Erie, Pa.
Emmons, Gilbert Collins ('42) (SKFL), Sr. Mech. Engr., Vern E. Alden, Engrs., 120 S. La Salle St., Chicago; *for mail,* 317 N. Elmwood Ave., Oak Park, Ill.
Emmons, Howard Wilson (J'35) (ABK), Assoc. Prof., Sch. of Engrg., Harvard Univ., Rm. 307, Pierce Hall, Cambridge, Mass.
Emory, John Brooks ('19) (FSK), Dist. Mgr., Combustion Engrg. Co., Inc., 516 Martin Bldg., Birmingham, Ala.
En Dean, Howard James ('46) (BPZ), Engr., Gulf Research & Devel. Co., Drawer 2038, Pittsburgh 30, Pa.
Ender, Joseph Jack (J'43) (CJMOLB), Asst. Ch. Engr., Grand Rapids Brass Co., 90 Scribner Ave., Grand Rapids 6; *for mail,* 101 Gold, S.W., Grand Rapids 4, Mich.
Endicott, George ('45) (CJM), Engr., Morgan Constr. Co., 15 Belmont St., Worcester 5, Mass.
Endlich, Wm. H. G. ('23) (EFS), Asst. Leading Mech. Draftsman, Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 180 Saratoga Ave., Brooklyn, N.Y.
Endsley, Louis E. ('13) (FKR), Cons. Engr., 516 East End Ave., Pittsburgh, Pa.
Enell, John Warren (J'40) (OMDACJ), Project Engr., Ignition Systems, Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* 259 E. 33rd St., Paterson 4, N.J.
Enes, James T. ('18; '35) (BJN), Asst. Dist. Engr., Ind. Harbor Works, Youngstown Sheet & Tube Co., East Chicago, Ind.; *for mail,* 8013 S. Harper Ave., Chicago 19, Ill.
Engberg, Ralph Edwin ('45) (NBHC), Design Engr., Allied Control Co., Inc., 2 East End Ave., New York 21; *for mail,* 179 Christopher Ave., Brooklyn 12, N.Y.
Engblom, Alex. N. ('14; '35; F'38) (CFMSTY), Dir., Tech. Mgr., Borås Wäfveri Aktiebolag; *for mail,* Varbergsvägen 25, Borås, Sweden.
Engdahl, Richard B. (J'40) (FKSBRZ), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Engdahl, Richard Emanuel (J'44) (ZOYNKH), Automatic Controls Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 17408 Dartmouth, Cleveland 11, Ohio.
Engebretson, Paul Albert (J'46) (ACL), 112½—6th St., Brookings, S.D.
Engel, Carl R. ('42) (FKSLDW), Mech. Engr., Great Lakes Carbon Corp., 333 N. Michigan Ave., Chicago, Ill.; *for mail,* Box 131, Route 1, Chesterton, Ind.
Engel, Fred Charles (J'43) (DBLJ), Engr., Westinghouse Elec. Corp., Wilkins Ave., Baltimore; *for mail,* 3315 Gwynns Falls Pkwy., Baltimore 16, Md.
Engel, R. A. (J'37) (LS), Ch. Engr., Fisher Governor Co.; *for mail,* 609 W. Main St., Marshalltown, Iowa.
Engelbach, Byron Adoph ('45) (LNBDCO), Plant Engr., New England Confectionery Co., 254 Massachusetts Ave., Cambridge 39; *for mail,* Box 107, Walpole, Mass.
Engelking, Walter Wm. ('20; '21; '35) (NBDHO), Ch. Engr., Superior Iron Works Co., 302 Grand Ave.; *for mail,* 36 Laurel Ave., Superior, Wis.
Engelman, William Henry ('27; '35) (SFH), Mech. Engr., Water Dept., City of Cleveland, 616 St. Clair Ave., Cleveland 14; *for mail,* 2184 Niagara Dr., Lakewood 7, Ohio.
Engels, Eugene O. ('40) (CK), Dir. of Engrg., Baker Perkins Inc., Hess & Williamson Sts., Saginaw, Mich.
Engelson, Howard Richard (J'45) (CJM), Asst. Engr., Fed. Mfg. & Engrg. Corp., 199-217 Steuben St., Brooklyn 5, N.Y.
Enger, Robert Carlson (J'43), Design Engr., Linde Air Products Co., Tonawanda; *for mail,* 593 W. Ferry St., Buffalo 13, N.Y.
Engesser, William Frederick (J'41) (CLMO), Asst. Prof., Ore. State College, Corvallis, Ore.
England, Frederic Earl (J'43) (FKS), 14616 McKinley Ave., R.F.D., Harvey, Ill.
Engle, Edgar W., Jr. (J'41) (CMJ), 911 N. Connecticut, Royal Oak, Mich.
Engle, Melvin Darke ('22; '31) (SFCDH), Mech. Engr., Head Mech. Engrg. Dept., Pa. Power & Light Co., 9th & Hamilton Sts.; *for mail,* 1610 Walnut St., Allentown, Pa.
Engle, Robert Worden (J'43) (MCDJ), Ensign, O-(V)S, U.S.N.R., PL-1, Bur. of Ord., Navy Dept., Washington, D.C.
Englehardt, Herbert Maser (J'45) (BCFM), Ensign, U.S.N.R., *for mail,* 10 Inwood Dr., Rochester 10, N.Y.
Engler, Walter George ('44) (CD), West. Mgr., Gifford Wood Co., 565 W. Washington St., Chicago 6, Ill.
English, Earl F. ('40) (CLEYO), V.P., W. A. Bechtel Co., 155 Sansome St., San Francisco 4, Calif.
English, Fred Steele ('22) (GBNO), Cons. Engr., Retired; 45 Squire St., New London, Conn.
English, Marvin Lee (J'41) (SEKA), 512 W. 22nd St., Minneapolis, Minn.
English, Philip Henry (A'28) (MJZC), Treas., New Haven Clock Co., 133 Hamilton St., New Haven, Conn.
English, Walter Myron ('35) (RMFCNB), Exec. V.P., Hanna Stoker Co., 5217 Brotherton Rd.; *for mail,* 3765 Harvard Ave., Mariemont, Cincinnati 27, Ohio.
Englund, John E. (J'36) (NBJMYW), Assoc., Mech. Engrg., Dept. of Mech. Engrg., Columbia Univ., New York 27, N.Y.
Engvall, Harry ('43), Asst. Ch. Engr., Helical Gear Dept., De Laval Steam Turbine Co., Trenton, N.J.
Eni, Louis Joseph (J'44), Seaman, 1st Class, (R.D.M.), LSM-388, F.P.O., San Francisco, Calif.; *for mail,* 1525 S. 10th St., Philadelphia 47, Pa.
Ennis, Herbert V. ('10; '21) (CR), Asst. to V.P., Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.
Ennis, James Edward ('40) (EFKRS), Engrg. Asst., N.Y. Cent. R.R. Co., 466 Lexington Ave., New York 17, N.Y.
Ennis, Joseph Burroughs ('09; F'45) (EFJKRS), Sr. V.P., Am. Loco. Co., 30 Church St., New York 8, N.Y.; *for mail,* 9 Pope Rd., Paterson, N.J.
Ennis, Robert L. ('41) (LSCDZH), Ch. Engr., Peter Cailler Kohler Swiss Chocolate Co., 555 S. 4th St., Fulton, N.Y.
Ennis, Walter Monroe, Jr. (J'41) (BMC), 1st Lt., Signal Corps, U.S.A., 696th Sig. Aircraft Warning Co., A.P.O. 915, c/o Postmaster, San Francisco, Calif.
Ennis, William Duane ('98; '07; F'39) (CMOL), *Treasurer,* A.S.M.E., '35-'44; R.F.D. 1, Wyckoff, N.J.
Eno, Wilber Sayre ('35; '45) (MJBC), Mem. Tech. Staff, Bell Tel. Labs., 463 West St., New York 14, N.Y.
Enochian, Warren S. (J'44), Box 575, Route 10, Fresno, Calif.
Ensign, Joseph D. (J'44) (EHS), 3456 Burns, Detroit 14, Mich.
Ensinger, Willis Bernard (J'31) (NABZSH), Prin. Engr., Bur. of Ord., Navy Dept., 17th St. & Constitution Ave., Washington, D.C.; *for mail,* 8303 Colesville Rd., Silver Spring, Md.
Enterline, Stevenson Monroe (J'41) (SNKDGZ), Jr. Mech. Engr., Devel. Engrg. Sec., Tech. Div., N.J. Zinc Co. of Pa., 4th St., Palmerton; *for mail,* 235 Hoffman Blvd., Ashland, Pa.
Enz, Karl ('20), Charge of Co. Engrg. Activities, China Corp., 20 Pine St., New York, N.Y.
Enzinger, L. Leonhard (J'43) (MOBDC), Indus. Mech. Engr., Am. Tool Engrg. Co., 1775 Broadway, New York; *for mail,* 496—4th St., Brooklyn, N.Y.
Epley, Frederic I. ('28; '35) (KFJBS), Superheater Engrg., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 79 Cedar Ave., Rockville Centre, N.Y.
Eppelsheimer, Paul Cyril ('45) (KS), Steam Specialist, Westinghouse Elec. Corp.; *for mail,* 2341 Dellwood Dr., N.W., Atlanta Ga.
Eppink, Herman John (J'42) (EFSNRO), 311th Engr. Bn., A.P.O. 450 c/o Postmaster, San Francisco, Calif.; *for mail,* Sioux Center, Iowa.
Eppler, Albert L. (J'40) (CMJ), Mfg. Supt., Spec. Products Div., Thomas A. Edison, Inc., 51 Lakeside Ave.; *for mail,* 29 Mt. View St., West Orange, N.J.
Epstein, Leonard Ira (J'45) (BJNDA), Pvt., U.S.A., Okinawa Univ., G-3 Obascom, A.P.O. 331, c/o Postmaster, San Francisco, Calif.; *for mail,* 801 West End Ave., New York 25, N.Y.
Epstein, Nathan (J'40) (MNOCLJ), Co-Owner, Engr., Designer, Accurate Mch. & Mfg. Co., 128-38 Mott St., New York 13; *for mail,* 45 Linden Blvd., Brooklyn 26, N.Y.
Epstein, Sherwin (J'38) (ASZ), Ensign, U.S.N.R., 1312 E. California St., Pasadena 5, Calif.
Erb, Edmund M. ('18; '35) (ABHJ), Ch. Engr., Hartford Spec. Mchy. Co., 287 Homestead Ave., Hartford 5, Conn.
Erb, Harold E. ('38) (CES), Supt. Engrg. & Maint., Abraham & Straus, Inc., 420 Fulton St., Brooklyn; *for mail,* 130-25—226th St., Laurelton, L.I. 13, N.Y.
Erb, Lester D. (J'40) (JMNKB), Welding Engr., Bur. of Reclamation, Dept. of Interior, New Customhouse, Denver 2; *for mail,* 1470 S. Birch, Denver 7, Colo.
Erdahl, John Maynard (J'41) (NBJHMC), Asst. Mech. Engr., Elec. Control Div., Allis-Chalmers Mfg. Co., West Allis; *for mail,* 2574 N. 64th St., Wauwatosa 13, Wis.
Erdman, Francis Hickok (J'44) (BANYC), Sr. Engr., Wright Aero. Corp., Paterson; *for mail,* 568 Doremus Ave., Glen Rock, N.J.
Erdman, Frederick Seward ('28; '36) (HKRSW), Assoc. Prof. Mech. Engrg., Cornell Univ., Ithaca, N.Y.
Erdoss, Bela Kalman ('43) (ABHZTO), Research Assoc., Asst. Prof., Dept. of Mech. Engrg., Lehigh Univ., Bethlehem; *for mail,* 158½ Shawnee Ave., Easton, Pa.
Ergeneman, Nuh (J'46) (ABK), Jr. Test Engr., Hudson Motor Car Co., Detroit 14; *for mail,* 1256 Newport St., Detroit 15, Mich.
Ergler, Paul Charles (J'41) (ABNMJC), Sr. Stress Analyst, Glenn L. Martin Co., Baltimore; *for mail,* 5317 Plymouth Rd., Baltimore 14, Md.
Erhard, Ralph, Jr. (J'40) (ACF), Matls. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 818 Hill Ave., Wilkinsburg, Pa.
Erhard, William Arnold ('45) (MNJCLY), Instr., Stevens Inst. of Tech., 5th & Hudson St., Hoboken; *for mail,* 711—80th St., North Bergen, N.J.
Erhardt, Walter Louis ('39; '45) (DCHMZ), Mech. Engr., Florence Pipe Fdy. & Mch. Co., Florence; *for mail,* 134 Warren St., Beverly, N.J.
Erickson, Alfred Louis ('25; '32; '35) (SFK), V.P., J. T. Thorpe, Inc., 948 E. 2nd St., Los Angeles 12, Calif.
Erickson, Arnold C. (J'41) (CES), Marlow Pumps, Ridgewood; *for mail,* 48 Rock Rd., Glen Rock, N.J.
Erickson, E. Vincent (J'39) (CHM), V.P., Gen. Mgr., Wm. H. Keller, Inc., Fulton St., Grand Haven, Mich.
Erickson, Edward A. (J'37) (FSNKBJ), Research Engr., Fuels & Combustion Equip., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Erickson, Edward Allen ('42; '44) (DKLMOZ), Mech. Engr., Commercial Solvents Corp., 1331 S. 1st St., Terre Haute, Ind.
Erickson, Erick Gustaf ('28; '35) (NMCBDJ), Am. Mch. & Metals, Inc., East Moline; *for mail,* 2419—11th Ave. A, Moline, Ill.
Erickson, Harry A. (J'43) (ABHRS), Mech. Engr., D. A. Stuart Oil Co. Ltd., 2727 S. Troy St., Chicago 23, Ill.
Erickson, Ole Peter ('28) (HNBJSE), Gen. Supt., Hendry Corp., Rattlesnake; *for mail,* 819 Grove Park Ave., Tampa 6, Fla.
Erickson, Ralph Erick E. ('32; '42) (NCBT), Sales Mgr., Speed Selector, Inc., Noble Court, Cleveland; *for mail,* 2629 Kingston Rd., Cleveland Heights 18, Ohio.
Erickson, Robert (J'27) (CJM), Plant Mgr., RCA Victor Div., Radio Corp. of Am., 501 N. La Salle St., Indianapolis, Ind.
Erickson, Robert Frederick (J'42) (ABMJYC), 1338 Rosedale Ave., Chicago 40, Ill.
Erickson, Waldemar John (J'45) (EKJNB), Ensign, U.S.N.R.; 6636 N. Glenwood Ave., Chicago 26, Ill.
Ericson, Franklin Rutger ('25; '35) (ERSBJ), Engr., Turbine Div., Gen. Elec. Co., 166 Broad St., Fitchburg, Mass.
Erikson, Arthur ('26) (DLBK), Designing Engr., Am. Smelting & Refining Co., Salt Lake City; *for mail,* 135 S. 6th East St., Salt Lake City 2, Utah.
Erkeneff, Nicholas (J'41) (ABR), Lt., Army Air Forces, 1469th Base Unit, Air Trans. Comd., A.P.O. 947, c/o Postmaster, Minneapolis, Minn.
Erlick, Preston Sumner (J'45) (AEO), c/o Mrs. Archer, 31 Long Dr., Hempstead, L.I., N.Y.
Ermenc, Joseph John (J'34) (EKHF), Prof. Mech. Engrg., Thayer Sch. of Engrg., Dartmouth College, Hanover, N.H.
Ernest, Edward William ('41) (CDJLMY), Supt., Punching Tool & Die Div., Gen. Elec. Co., 1 River Rd., Schenectady 1; *for mail,* 2249 Grand Blvd., Schenectady 8, N.Y.
Ernst, Alfred Fasset ('09; '14) (CKSEL), Assoc., George H. Corey, 20 Exchange Pl., New York 5; *for mail,* Apt. 416B, Fenimore Cooper House, Larchmont Acres, Larchmont, N.Y.

Ernst, Carroll Auld (J'38) (HJY), Asst. to Ch. Engr., Smart Turner Mch. Co. Ltd., 191 Barton St. E.; *for mail*, 233 Victoria Ave. N., Hamilton, Ont., Can.
Ernst, Frederick C. ('31) (SJMLHY), Ch. Engr., Jenkins Bros., 80 White St., New York 13; *for mail*, 199-08—100th Ave., Hollis, L.I., N.Y.
Ernst, Hans ('26; F'44) (MBJACH), Research Dir., Cincinnati Milling Mch. Co., Marburg Ave., Cincinnati 9, Ohio.
Ernst, John Price (J'43) (HKS), Sr. Engr., Kaiser Engrs., Inc., Kaiser Bldg.; *for mail*, Apt. 2, 2654 Harrison St., Oakland 12, Calif.
Ernst, Walter ('28; '35) (HYNOLB), V.P. for Engrg., Commonwealth Engrg. Co., 1771 Springfield, Dayton 3; *for mail*, R.F.D. 2, Mt. Gilead, Ohio.
Eron, Lewis John ('43) (KSH), 195—2nd St. N.; *for mail*, 931 Gardner St., Wisconsin Rapids, Wis.
Erskine, James Harold ('33; '38) (SFC), Supt., Cliffside Steam Sta., Duke Power Co., Cliffside, N.C.
Erskine, Walter H. ('23) (RM), Mfrs. Rep., 2371 Chilcombe Ave., St. Paul, Minn.
Ersted, Gordon Theodore (J'42) (CMJDNY), Mech. Engr., Maint. Engrg., Fisher Body Div., Gen. Motors Corp., 4300 S. Saginaw St., Flint 2; *for mail*, 906 Garland St., Flint 4, Mich.
Ervin, Fred Reid (J'46), Instr., Mech. Engrg. Dept., Duke Univ.; *for mail*, 2111 Club Blvd., Durham, N.C.
Ervin, Thomas Callaway ('27; '35) (CMS), Pres., Lucey Boiler & Mfg. Corp., 1514 Chestnut St., Chattanooga; *for mail*, 1117 E. Brow Rd., Lookout Mt., Tenn.
Erway, Charles Albert (J'43) (SLNCFA), 2nd Asst. Engr., U.S. Lines, Pier 62, New York; *for mail*, 826—43rd St., Brooklyn 32, N.Y.
Erwin, Henry Parsons ('24) (CWMH), Lt. Col., Ord. Reserve; 723—15th St., N.W., Washington 5, D.C.
Erwin, Walter Clark (J'37) (ATHF), Bellevue, Morganton, N.C.
Eschborn, Ralph John (J'44) (ABCKN), Capt., Army Air Forces, Chief, Eng. Unit, Equip. Lab., Engrg. Div., Air Tech. Serv. Comd., TSEPE-6K, Area 3, Wright Field; *for mail*, 104 Oxford Ave., Dayton, Ohio.
Escher, William Franklin ('42; '44) (OCLS), Ch. Mech. Engr. with Albert C. Martin, 233 Higgins Bldg., Los Angeles 12, Calif.
Esdaile, Hector Milton (J'44) (SFKRB), Supt., Combustion Engrg. Corp. Ltd., 540 Dominion Sq. Bldg., Montreal, Que., Can.
Eserkaln, Theodore Fredrick (J'33; '39) (BCHO), Cons. Engr., 2250 N. 62nd St., Wauwatosa 13, Wis.
Eshelman, Clarence M. ('18) (CBYWJ), 4937 Hartwick St., Los Angeles 41, Calif.
Eshelman, Joseph William ('26; '39) (SAFZLK), *Manager*, '39-'42, *Vice-President*, '42-'44; Owner, Jos. W. Eshelman & Co., 1116-21 Martin Bldg., Birmingham 3, Ala.
Eshelman, Rodney L. ('37; '46) (CJKLSM), Area Maint. Supt., E. I. du Pont de Nemours & Co., Box 993, Charleston, W.Va.
Esherick, George, Jr. ('15; '20) (CFKRSZ), Job Engr., Kellex Corp., 233 Broadway, New York; *for mail*, 42 Longview Rd., Port Washington, L.I., N.Y.
Eskin, Samuel G. ('26; '42) (ABL), Dir. of Research, Research Lab., Robertshaw Thermostat Co., 1201 Washington Blvd., Pittsburgh, Pa.; Dir. of Research & Devel., Grayson Heat Control, Ltd., 833 N. Highland Ave., Los Angeles 38, Calif. (*use latter address for mail*).
Esposito, Daniel J. (J'39) (FSZK), Results Engr., United Illum. Co., 1115 Broad St., Bridgeport 1; *for mail*, 126 Savoy St., Bridgeport 6, Conn.
Espy, Melvin Paul (J'34) (AB), Gp. Engr., Engrg. Test Dept., Glenn L. Martin Co., Middle River; *for mail*, Green Spring Ave., Lutherville, Md.
Espy, William Nelson ('21; '35) (ABEKSZ), Prof. Mech. Engrg., Univ. of Ill.; *for mail*, 608 W. Nevada St., Urbana, Ill.
Esselman, Richard B. ('37; '45) (CDMY), Maint. Supt., B. F. Goodrich Co., 5400 E. Olympic, Los Angeles 22; *for mail*, 810 King St., San Gabriel, Calif.
Esselstyn, Horace H. ('09) (CDLMOY), Partner, W. A. & H. H. Esselstyn, 1135 Majestic Bldg., Detroit 26, Mich.
Essex, Thos. J. ('30), Oper. & Maint., Kansas City Power & Light Co.; *for mail*, 4944 Lydia Ave., Kansas City, Mo.
Estabrook, Mansfield ('03; '19) (J), Brown, Wheelock, Harris, Stevens, Inc., 22 E. 40th St., New York 16, N.Y.
Estcourt, Vivian F. ('27; '32) (SCFJHE), Asst. Engr. of Opera., Pacific Gas & Elec. Co., 245 Market St., San Francisco 6, Calif.
Estep, Adrian Clarence ('45), 2514 Calhoun St., Seattle 2, Wash.
Estep, Frank Leslie ('19) (DFJOS), Cons. Engr., 101 Park Ave., New York 17, N.Y.
Estep, Thomas G. ('19) (SFE), Acting Head, Dept. of Mech. Engrg., Carnegie Inst. of Tech., Pittsburgh 13, Pa.
Esterly, John Reese (J'43) (AJNZEM), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 16614 W. Park Ave., Cleveland 11, Ohio.
Estes, Howard M. ('37) (HLJSB), Struc. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail*, 27 Wolcott Rd., Milton 86, Mass.
Estes, Hugh Howard (J'40) (AC), 1st Lt., Army Air Forces, Engrg. Officer, Hdq., South Atlantic Wing, Air Transport Comd., A.P.O. 643, Miami, Fla.
Estey, Charles Gurley (J'42) (ECJMYA), Estey Bros. Co., 1006 Metropolitan Ave., Brooklyn 6, N.Y.
Estrada, Herbert ('40) (FHS), Supt., Economy Div., Philadelphia Elec. Co., Edison Bldg., Philadelphia, Pa.
Etchen, Harold Graham ('38; '45) (CMNJS), Mfg. Engr., Steam Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; for mail, 605—10th Ave., Prospect Park, Pa.
Etlin, Henry Bernard (J'42) (ACM), Pres., Gen. Mgr., H.B. Etlin Mfg. Co., 35 Golden Ave.; *for mail*, 7 Chadwick Ave., Toronto, Ont., Can.
Etter, Lewis Fort (J'35) (MCR), Prod. Control Mgr., Preco, Inc., 960 E. 61st St., Los Angeles; *for mail*, 1484 Cambridge Rd., San Marino 9, Calif.
Ettinger, Jacob (J'45) (CMO), Sales Engr., Empire Design Corp., 1560 Broadway, New York 19; *for mail*, 307 W. 79th St., New York, N.Y.
Ettorre, James Edward (J'44) (NBJML), Design Engr., Bullard Co., 286 Canfield Ave., Bridgeport 2; *for mail*, 187 Berwick Ave., Bridgeport 5, Conn.
Eubank, Carl Neal (A'45) (CS), Dist. Sales Mgr., Mex. Refractories Co., 1731-33 Walnut St., Kansas City 8; *for mail*, 19 E. 68th St., Kansas City 5, Mo.
Eubank, Clifford Joseph, Jr. (J'40) (KOA), Fosdick & Hilmer, 4th & Walnut St., Cincinnati, Ohio; *for mail*, Box 360, Route 4, Erlanger, Ky.
Euvrard, LeRoy Eugene (J'45), Sr. Serv. Engr., Ford Instrument Co., Long Island City; *for mail*, 501 W. 121st St., New York, N.Y.
Evans, Benjamin G. (J'38) (EFS), Lt. (j.g.), U.S.N.R., 333 Angelo Dr., Corpus Christi, Tex.
Evans, Brian Douglas (J'39) (ACJM), Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail*, 48 Thompson Pk., Glen Cove, L.I., N.Y.
Evans, Charles Snyder (J'39) (LKSRDZ), Jr. Mech. Engr., Socony-Vacuum Oil Co., Inc., 37-98 Review Ave., Long Island City 1, N.Y.
Evans, Charles T., Jr. (J'42) (JBDMAL), Ch. Metallurgist, Elliott Co., Jeannette; *for mail*, Box 136, Delmont, Pa.
Evans, Clair O. ('41) (BSK), Asst. Sales Mgr., Tube Div., Phelps Dodge Copper Products Corp., 40 Wall St., New York 5, N.Y.
Evans, Daniel Edward ('33) (CDJM), Chmn., Bd. of Dirs., Evans, Deakin & Co., Ltd., Ryan House Charlotte St., Brisbane, Queensland, Australia.
Evans, David Franklin ('16; '25) (CL), 315 Providence Rd., Media, Pa.
Evans, Edward Benjamin (J'31) (NAJ), Partner, Wood, Arey, Herron & Evans, 2801 Carew Tower, Cincinnati 2, Ohio.
Evans, Francis C. (J'45) (BZJCMY), Engrg. Supvr., Am. Dist. Tel. Co., Inc., 155 Ave. of The Americas, New York 13, N.Y.
Evans, Frederick Huston ('18) (CZNG), Dist. Supt., U.S. Employment Serv., 12 New Market St.; *for mail*, Red Oaks Mill, Poughkeepsie, N. Y.
Evans, George Arminius ('17; '20) (SFEK), Mech. Engr., U.S.A., Army Port & Serv. Comd., A.P.O., c/o Postmaster, San Francisco, Calif.; *for mail*, 2256-B Liliha St., Honolulu 3, T.H.
Evans, Geo. B. ('14), Retired; 275 N. Union Blvd., St. Louis 8, Mo.
Evans, Gordon M. ('20), Box 24, Redford Sta., Detroit, Mich.
Evans, Hugh S., Jr. (J'44) (CHM), Engrg. Trainee, De Laval Steam Turbine Co., Nottingham Way; *for mail*, Apt. 8, 39 N. Hermitage Ave., Trenton 2, N.J.
Evans, James Norton ('28; '34; '35) (ESC), Supt., Fuels & Power Dept., Natl. Tube Co., 28th St.; *for mail*, 2748 S. Jefferson Blvd., Lorain, Ohio.
Evans, John Gow, Jr. (J'42) (SFKERH), Box 210, Va. Poly. Inst., Blacksburg, Va.
Evans, Lawrence Eugene (J'26) (ZFSLHR), Sales, Serv. Engr., Bailey Meter Co., 8527 Rosalie Ave., Brentwood, St. Louis 17, Mo.
Evans, Leigh R. ('18), V.P., Morrison Mch. Products Inc.; *for mail*, 114 Country Club Dr., Elmira, N.Y.
Evans, Martin H. (J'40) (CDL), 1st Lt., Co. C, 41st Armd. Inf. Regt., A.P.O. 252, c/o Postmaster, New York; *for mail*, 245 Beach 138th St., Rockaway Beach, N.Y.
Evans, Melvin J. ('17; '23; '26) (C), Chmn. of Bd., Personnel Dir., Evans Associates, Inc., 307 N. Michigan Ave., Chicago; *for mail*, Brassie Ave., Flossmoor, Ill.
Evans, Myrle Charles ('44) (CDHJMZ), Mgr., Mfg. Research Dept., Internatl. Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Evans, Norman A. ('30; '38) (ACJ), Sales Mgr., Pressed Steel Tank Co., Milwaukee, Wis.
Evans, Philip Dudley (A'44) (KPS), Mgr., Kansas City Office, Chase Brass & Copper Co., Lathrop Bldg., Kansas City 6; *for mail*, 726 E. 67th St., Kansas City 5, Mo.
Evans, Porter, Jr. (J'42) (BJO), Devel. Engr., Elliott Co., Jeannette; *for mail*, Box 306, R.D. 5, Greensburg, Pa.
Evans, Quentin James (J'44) (SAEKRJ), Research Asst., Carbide & Carbon Chemicals Corp., Box 41, Sta. J; *for mail*, 191 Claremont Ave., New York 27, N.Y.
Evans, Richard A. (J'41), 309 Avenue Rd., Toronto, Ont., Can.
Evans, Richard Harold (J'46), Mech. Engr., Tournalayer Div., R. G. Le Tourneau Inc., Longview, Tex.
Evans, Robert G. N. ('42), Pres., Evans Engrg. Co., 2930 W. Burleigh St., Milwaukee 10, Wis.
Evans, Robert Thomas ('40) (CBDKM), Bldg. Constructor, Route 7, Phoenix, Ariz.
Evans, Seth ('29), Elec. Engr., Hughes Tool Co., Houston, Tex.
Evans, Wallace Rockwell, Jr. (J'41) (SBF), Lt., U.S.N.R.; *for mail*, 2128 Chambwood Dr., Charlotte, N.C.
Evans, Walter F. (J'40), Lt., 20th Ferrying Gp., Municipal Airport, Nashville, Tenn.
Evans, Wilfred I. ('30), Ch. Engr., Commonwealth Edison Co., 3501 S. Pulaski Rd., Chicago 23, Ill.
Evans, William David (J'40) (LNW), Tool Designer, Continental Can Co., 601 Myrtle St.; *for mail*, 5414—23rd Ave., S., Seattle 8, Wash.
Evarts, Howard M. ('23; '33) (BES), Mech. Engr., Div. of Bldgs., City of Buffalo, City Hall; *for mail*, 675 Richmond Ave., Buffalo, N.Y.
Evarts, Ralph Eaton ('27; '35) (CMOJZL), Devel. Engr., Santay Corp., 351 N. Crawford St.; *for mail*, 30 W. Chicago Ave., Chicago 10, Ill.
Evashevski, Kenneth (J'40) (HMNK), Liaison Engr., Progressive Welder Co., 3050 E. Outer Dr., Detroit 12; *for mail*, 15516 Troester, Detroit 5, Mich.
Everett, Allen ('42) (CMO), Sr. Assoc., McKinsey & Co., 60 E. 42nd St., New York 17, N.Y.
Everett, C. T. (J'36), 270 N. Treamont St., Indianapolis, Ind.
Everett, Franklin Leland ('26; '37) (BNJZO), Asst. Prof. Engrg. Mechanics, College of Engrg., West Engrg. Bldg., Univ. of Mich.; *for mail*, 1722 Shadford Rd., Ann Arbor, Mich.
Everett, Harold Arthur ('22) (ESBA), Head, Dept. of Mech. Engrg., The Pa. State College, State College, Pa.
Everett, Harvey J. (J'35) (LBKHSJ), Research Engr., Mixing Equip. Co., Inc.; 1024 Garson Ave., Rochester 9, N.Y.
Everett, Russell William ('18; '46) (SKHZLJ), Supt. Constr. & Serv., Foster Wheeler Corp., Carteret; *for mail*, 56 Lincoln Ave. W., Roselle Park, N.J.
Everett, Wilhelm Sydow ('36; '44) (EHKN), Dist. Office, Enterprise Eng. & Fdy. Co., 18th & Alabama, San Francisco; *for mail*, 1502 Beverly Pl., Albany 6, Calif.
Everett, William Bailey (J'45) (CML), Secy., John T. Everett's Sons Co., 632 Goodwyn Inst., Memphis 1; *for mail*, 168 E. Parkway S., Memphis 4, Tenn.
Evers, Lester Arthur (J'41) (SFB), Engr., Jr. Grade, Commonwealth Edison Co., 72 W. Adams St.; *for mail*, 5215 North St. Louis Ave., Chicago, Ill.
Eves, Irwin Gene Chalkley (J'45) (CDO), Sales Engr., Am. Monorail Co., 549 W. Washington Blvd., Chicago 6; *for mail*, Box 318 Y, R.F.D. 1, Melrose Park, Ill.
Evoy, Martin ('26; '33; '35) (CST), 1 Edgehill Rd., Abington, Pa.
Ewald, Harris O. (J'41) (BCM), Engr., Evinrude Motors, 4143 N. 27th St.; *for mail*, 4168 N. 21st St., Milwaukee 9, Wis.
Ewart, Arthur F. ('17; '35), Mgr., Hawaiian Div., Honolulu Iron Works Co.; *for mail*, 2370 Nuuanu St., Honolulu, T.H.
Ewart, Hubert E. ('37) (JLD), Mech. Engr., Blaw-Knox Co., Blawnox; *for mail*, 1700 Grall Ave., Pittsburgh 9, Pa.
Ewart, William Morris ('19; '33) (CMJN), Factory Mgr., Carter Carburetor Corp., 2840 N. Spring St., St. Louis 7, Mo.
Ewer, Roland Gibbs ('19) (KBEHL), Ch. Engr., Air Conditioning & Refrigeration Div., Worthington Pump & Mchy. Corp., 401 Worthington Ave., Harrison; *for mail*, 466 Tremont Pl., Orange, N.J.
Ewert, William A. ('21; '27; '35) (MNCLJ), Cons. Engr., Hotel Baker, St. Charles, Ill.
Exler, Donald C. (J'35) (SHJEFK), Oper. Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Exley, Lucius Morgan ('39) (CSF), Elec. Prod. Mgr., L. I. Ltg. Co., Glenwood Landing, L.I., N.Y.
Exline, Paul Gettis ('37) (BNZ), Sec. Engr., Gulf Research Devel. Co., P.O. Drawer 2038, Pittsburgh 30, Pa.
Extence, Alan Berr (J'42), Lt. (E), H.M.C.S. *Jonquiere*, c/o C.F.M.O., London, England.
Eyler, Kenneth Earl (J'43), 1061 N. Duke St., York, Pa.
Eyre, Thomas T. ('15), 2818 Dumfries Rd., Los Angeles 34, Calif.
Eyres, Stanley A. ('44), 27 Hollister Ave., Utica 3, N.Y.
Eyring, Edward John (J'43) (JKSCHM), 1541 Colonial Terrace, Arlington, Va.

F

Faasse, William (J'42) (SKFEBC), Erection Engr., Babcock & Wilcox Co., Barberton, Ohio; *for mail*, 2948—11th St., Port Arthur, Tex.
Faast, Frank E. (J'45) (CJM), Seaboard Mchy. Corp., 29 Broadway, New York, N.Y.; *for mail*, 40 Lenox Ave., East Orange, N.J.

Faatz, Embry D. ('45) (CAFMEJ), Project Engr., Scintilla Magneto Div., Bendix Aviation Corp., Sidney, N.Y.
Fabel, Donald Criston ('35) (JOCMB), Lt. Col., Ord. Dept., U.S.A.; 2411 Edgerton Rd., Cleveland Heights 18, Ohio.
Fabens, Andrew Lawrie, Jr. (J'39) (MNCJB), Engr., Buckeye Aluminum Co., Spruce St.; *for mail,* 128 W. Henrietta St., Wooster, Ohio.
Faber, Paul V. (J'39) (CJM), Mech. Design Engr., Marine Div., Bendix Aviation Corp., 106 Nostrand Ave., Brooklyn 5; *for mail,* 6832 Harrow St., Forest Hills, N.Y.
Fabera, Wenzel (J'26) (CLS), Engr., Proctor & Gamble Co., 1701 W. 7th St.; *for mail,* 665 Terraine Ave., Long Beach, Calif.
Fabian, Francis G., Jr. (J'38) (ACD), Lt., U.S.N.R., Box 644, Lake Forest, Ill.
Faget, Maxime Allan (J'43), Lt. (j.g.), U.S.N.R., U.S.S. *Guavina,* SS-362, F.P.O., San Francisco, Calif.
Fagg, Leo Wayne ('45) (HBJOMN), Mgr., Field Engrg. Dept., S. M. Jones Co., 518 McBirney Bldg., Tulsa 3; *for mail,* 3007 S. Boston Pl., Tulsa 5, Okla.
Fagin, Sidney (J'45) (BNAEJK), Ord. Dept., U.S.A., Test Engr., Supersonic Wind Tunnels Lab., Aberdeen Proving Ground, Md.; *for mail,* 2196 Matthews Ave., Bronx 60, N.Y.
Fahrlander, Henry William, Sr. ('45) (NBGJRH), Dir. of Stands., Gen. Mchy. Corp.; *for mail,* 112 Elvin Ave., Hamilton, Ohio.
Fahy, James Aloysius ('42) (CDFALO), Pur. Agt., Institutional Commodity Services, Inc., 462 Madison Ave., New York 22; *for mail,* 115 W. 71st St., New York 23, N.Y.
Faig, John Theodore ('97; '05; F'43) (CMS), Pres. of Depts., Ohio Mechanics Inst., 1104 Walnut St., Cincinnati 10, Ohio.
Faile, E. H. ('31), R.F.D. 1, Westport, Conn.
Failmezger, Eugene (J'33), Air Appliance Div., U.S. Hoffman Mchy. Corp., Syracuse; *for mail,* 109 Wellington Rd., Dewitt, N.Y.
Fairbanks, Charles Miguel ('25; '37) (BLJCM), Design Engr., Gen. Elec. Co., 9601 Elmwood Ave., Philadelphia 42, Pa.
Fairchild, Charles Owen ('44) (BKZY), Consultant, 114-74—78th Pl., St. Albans, N.Y.
Fairchild, Frederick P. ('15; '26) (SEFJB), Ch. Engr., Elec. Engrg. Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Fairchild, Sherman M. (A'27), Chmn. of Bd., Fairchild Eng. & Airplane Corp., 30 Rockefeller Plaza; *for mail.* 17 E. 65th St., New York, N.Y.
Fairchild, Wesley Lee (J'45) (KFBCZ), Jr. Engr., Furnace Engrg. Dept., Corning Glass Works, Corning, N.Y.
Faires, Virgil M. ('28; '32) (NSJEKC), Head, Mgmt. Engrg. Dept., Agric. & Mech. College of Tex., College Station, Tex.
Fairfield, John Guthrie ('17; '25; '35) (KBSEFZ), Prof. Heat Engr., Rensselaer Poly. Inst., 110—8th St., Troy, N.Y.
Fairhurst, Kirk S. (J'38), Capt., Army Air Forces, 1636 Camden Ave., Los Angeles 25, Calif.
Fairhurst, Thurstan William ('41), Dir., Ruston & Hornsby, Ltd.; *for mail,* Imperial House, 15-19 Kingsway, London, W.C. 2, England.
Fairley, George G. (J'41) (EABR), Engrg. Draftsman, Phillips Petroleum Co.; *for mail,* 2029 Osage, Bartlesville, Okla.
Fairman, Seibert ('31; '35) (BA), Prof. Engrg. Mechanics, Purdue Univ., West Lafayette, Ind.
Fairman, Stanley Wright ('33; '43) (CMFDSE), Engr., Consumer Products Div., Corning Glass Works; *for mail,* 201 Watauga Ave., Corning, N.Y.
Faison, George Washington, III (J'43) (ACMFKS), 2nd Lt., Army Air Forces, Flight Engr., 680th Sqd., 504th Gp., A.P.O. 336, c/o Postmaster, San Francisco, Calif.; *for mail,* c/o Stuart Moore, Lexington, Va.
Falcon, Joseph Albert (J'43) (SAE), Jr. Mech. Engr., Gibbs & Cox, Inc., 21 West St., New York 6; *for mail,* 83 Van Sicklen St., Brooklyn 23, N.Y.
Fales, Henry Hammett ('13) (STOLDK), Cons. Engr., 408 Grosvenor Bldg., Providence 3, R.I.
Fales, Herbert Gordon ('21; '33) (JA), V.P., Internatl. Nickel Co., Inc., 67 Wall St., New York 5, N.Y.
Fales, James Nelson True ('45) (BEJ), Engrg. Dept., Wolverine Motor Works, Inc., 35 Union Ave., Bridgeport 2, Conn.
Falk, Edward A. (J'38) (AZF), Lt. (j.g.), U.S.N.R., Ship Repair Unit, Naval Shipyard, Mare Island, Vallejo, Calif.
Falk, George Elias (J'34) (NMBJ), Sr. Mech. Engr., Applied Physics Lab., The Johns Hopkins Univ., 8621 Georgia Ave., Silver Spring, Md.
Falk, Harold Sands ('16) (CDJM), Pres., Falk Corp., 3001 W. Canal St., Milwaukee 8, Wis.
Falk, Melvin L. (J'36) (ABHW), Stress Analyst, Beech Aircraft Corp.; *for mail,* Route 4, Wichita, Kan.
Falkner, Jefferson Cameron ('23; '35) (BCFJKM), Mgr., Elec Prod. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Falkovich, Oleg C. ('41) (KBAHNW), Research Engr., Bonneville Power Admin., Box 3537, Portland, Ore.
Falla, Fernando ('23), Spec. Designer, Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.; *for mail,* 59 Elm St., Maplewood, N.J.
Fallon, John (J'10) (JFCKGZ), Chmn., Managing Dir., Incandescent Heat Co., Ltd., Cornwall Rd., Smethwick, Birmingham; *for mail,* 82 St. Bernard's Rd., Olton, Warwickshire, England.
Falls, Eugene K. (J'32) (SEBC), Asst. Prof. Mech. Engrg., Univ. of Rochester, River Campus, Rochester 3, N.Y.
Falvey, Jeremiah Alfred (J'41) (BASHCF), Design Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 815 Summit Ave., Prospect Park, Pa.
Fambro, George W. (J'41) (ABC), Draftsman, Glenn L. Martin Co.; *for mail,* Cent. Y.M.C.A., 24 W. Franklin St., Baltimore, Md.
Fangboner, Harold Frank (J'37) (NMHB), Project Engr., John Bean Mfg. Co., 735 E. Hazel, Lansing 4, Mich.
Fangemann, Wm. Howard (J'36) (BGM), Charge Engrg. Records, Intertype Corp., 360 Furman St.; *for mail,* 300 Sterling St., Brooklyn, N.Y.
Fanjul, Higinio Charles (J'35) (CDM), Asst. Mgr., Cespedes Sugar Co., Central Cespedes, Camagüey, Cuba.
Fankhauser, Arnold Franklin Eaton (J'43) (CDOMAJ), Design Engr., Plant Engrg. Dept., Gen. Elec. Co., 1635 Broadway; *for mail,* 1020 McKinnie Ave., Ft. Wayne, Ind.
Fannon, William Alsop ('07) (DFKNSW), Pres., Treas., Fannon Trading Co., Inc., 206 W. College Ave., Appleton, Wis.
Fanshier, Chester ('43) (CGZYJB), Pres., Metal Goods Mfg. Co., 106-110 S. Park, Bartlesville, Okla.
Farbar, Leonard (J'36) (HK), Process Engr., Avon Refinery, Tide Water Associated Oil Co.; *for mail,* 10 Como St., Daly City, Calif.
Farber, Erich Alexander (J'43) (MBK), Lowry Hall, Columbia, Mo.
Fardelmann, John Henry ('18; '22; '27) (COS), Mech. Engr., M. H. Treadwell Co., 140 Cedar St., New York 6, N.Y.; *for mail,* 217 Alexander Ave., Montclair, N.J.
Faridany, H. P. (J'38) (CHS), Pipe Line Supt., Kermanshah Petroleum Co. Ltd., Kermanshah, Iran.
Farin, William George (J'45) (KLP), Jr. Engr., E. I. du Pont de Nemours & Co., Box 993, Charleston 24; *for mail,* 1524 B Washington E., Charleston 1, W.Va.
Farintosh, Henry Ernest (J'44) (ASBCMJ), Canadian Natl. Carbon Co. Ltd., Davenport Rd., Toronto; *for mail,* 11-A Silverbirch Ave., Toronto 8, Ont., Can.
Farkas, Thomas Paul (J'42) (BCLMZ), Mech. Engr., Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14; *for mail,* 34-40—79th St., Jackson Heights, L.I., N.Y.
Farley, James Joseph (J'40) (ANEBM), Aero. Engr., Naval Air Matl. Center, Naval Base, Philadelphia; *for mail,* 1623 N. 60th St., Philadelphia 31, Pa.
Farmer, Edward H. ('41) (ACMLDJ), Supt. Fabrication, Lockheed Aircraft Corp.; *for mail,* 4366 Clybourn Ave., Burbank, Calif.
Farmer, F. Malcolm ('15), V.P., Cons. Engr., Elec. Testing Labs., 79th St. & East End Ave., New York, N.Y.
Farmer, George Coburn ('44) (BCR), Commercial Engr., Westinghouse Air Brake Co., Wilmerding, Pa.
Farmer, John Taylor ('40) (SFEHLC), Mech. Engr., Montreal Engrg. Co., Ltd., 244 St. James St., Montreal, Que., Can.
Farmer, William Crawford ('45) (ACBJM), Lt. Col., Plans & Policies Officer, Ord. Dept., U.S.A., Rm. 4 E 372, The Pentagon, Washington 25, D.C.
Farnell, George (J'44) (EKJF), Mech. Engr., Research Project, Mass. Inst. of Tech., 77 Massachusetts Ave.; *for mail,* 46 Massachusetts Ave., Cambridge 39, Mass.
Farnham, Duane Wesley (J'34) (EFLSKA), 2621 E. 20th St., Tulsa, Okla.
Farnham, George Wright ('16; '25; '35) (CAMBJN), V.P., Irwin-Farnham Publ. Co., 332 S. Michigan Ave., Chicago 4, Ill.
Farnham, Roger Durfee (J'42), Lt. (j.g.), U.S.N., T.T.S.A., Naval Air Sta., Banana River, Cocoa, Fla.
Farnham, Walter E. ('18; '35), Prof. Graphics, Head of Dept., Engrg. Sch., Tufts College, Medford, Mass.
Farnsworth, Augustus Page ('43) (COMA), Partner, Coverdale & Colpitts, 120 Wall St., New York 5, N.Y.
Farnsworth, Paul L. (J'41) (ABE), 3712 Highland Ave., Drexel Hill, Pa.
Farquhar, Bannister Wells ('40) (JCSKB), Engr., Gulf Oil Corp.; *for mail,* 4525 Alamosa Dr., Port Arthur, Tex.
Farquhar, Francis ('07) (C), V.P., Chmn. of Bd., A. B. Farquhar Co., 142 N. Duke St.; *for mail,* Box 1432, York, Pa.
Farquhar, Lloyd Carleton ('19; '35) (CJR), Works Mgr., East St. Louis Works, Am. Steel Fdys., 2039 E. Broadway, East St. Louis, Ill.
Farr, Arthur V. ('37) (Y), Pres., Farr Engrg. Co., 300 E. 43rd St., New York 17, N.Y.; *for mail,* 44 Clinton Ave., Maplewood, N.J.
Farr, Eric Martin (J'45) (FJABSZ), Test Engr., Wright Aero. Corp., Beckwith Ave., Paterson; *for mail,* Packanack Lake, N.J.
Farr, Morrill Spencer (J'35) (MJBKLY), Asst. Gen. Mgr., Secy., Treas., Farr Co., 2615 Southwest Dr., Los Angeles 43, Calif.
Farrar, D. Franklin, Jr. (J'40) (ABW), Design Draftsman, Plant Engrg. Dept., Nashville Div., Consltd. Vultee Aircraft Corp., Couchville Pike; *for mail,* 4109 Belmont Blvd., Nashville 4, Tenn.
Farrell, Eugene Francis (J'36) (BKE), Devel. Engr., Detroit Gear Aircraft Parts Div., Borg-Warner Corp., 12345 Kercheval St., Detroit; *for mail,* 1602 Anita St., Grosse Pointe Woods 30, Mich.
Farrell, Frederick L. ('13; '21; '35) (BEFKSW), Dist. Mgr., Combustion Engrg. Co., Inc., 80 Federal St., Boston; *for mail,* 4 Bay State Rd., Belmont 78, Mass.
Farrell, Jas. A., Jr., (J'24), Pres., Am. South African Line, Inc., 26 Beaver St., New York, N.Y.
Farrell, Joseph Milton ('33; '35) (AEMOCN), Classification Coder, War Assets Corp., Pratt & Whitney Plant, Kansas City, Mo.; *for mail,* 121 N. Yale St., Wichita 8, Kan.
Farrell, Raymond Patrick ('44) (CJM), Secy., Treas., Harper Packing Co., 312 Patterson, Chester; *for mail,* 821 Kerper St., Philadelphia 11, Pa.
Farrington, Stephan G. ('41) (ZSLMCO), V.P., Gen. Mgr., Foster Engrg. Co., 109 Monroe St., Newark; *for mail,* 35 Beverly Rd., West Orange, N.J.
Farrington, Stephen John, Jr. (J'42) (CMD), Methods Engr., De Laval Steam Turbine Co., Trenton 2; *for mail,* 834 Berkeley Ave., Trenton 8, N.J.
Farris, Marshall Elmer ('35) (HMNRS), Dean, College of Engrg., Univ. of New Mex., Albuquerque, New Mex.
Fassbender, Walter J. (J'22) (SBJKHF), Designing Engr., Yarnall-Waring Co., 102 E. Mermaid Lane, Philadelphia 18; *for mail,* 7636 New 2nd St., Philadelphia 26, Pa.
Fassett, Donald G. (J'37) (CDNW), Asst. Engr., Mch. Design, Ederer Engrg. Co., 2931—1st Ave. S., Seattle 4; *for mail,* Box 778, R.F.D. 1, Port Blakely, Wash.
Faulkner, C. L. (J'36) (LS), Soybean Mill, Swift & Co.; *for mail,* 52 S. O'Neal St., Frankfort, Ind.
Faulkner, David S. ('12; '13; '35) (CEJ), V.P., Natl. Supply Co., Torrance, Calif.
Faus, Herbert Williams ('41) (RFJSC), Engr. M.P., N.Y. Cent. System, 466 Lexington Ave., New York 17, N.Y.
Faust, H. M. ('30; '35) (FSEC), Research Engr., Mgr. Coal Devel., N.Y. Coal Co., 150 E. Broad St., Columbus 15; *for mail,* 2873 E. Cleft Dr., Columbus 8, Ohio.
Faust, Per Alexander ('21) (BCDFKN), M.M., Maint. Engr., Internatl. Smelting & Refining Co., Inspiration; *for mail,* Box 1057, Miami, Ariz.
Fava, Albert Anthony ('45) (SFLMD), Ch. Engr., Atlantic Gelatin Co., Inc., Hill St., Woburn, Mass.
Faville, Frederick Alden ('44), Pres., Faville-Le Vally Corp., 105 W. Adams St., Chicago 3, Ill.
Fax, David Hirsch (J'38) (BSKNC), Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 2639 N. 46th St., Milwaukee 10, Wis.
Fay, Charles H. ('27) (FHS), Cons. Engr., 216 Glen Ave., Millburn, N.J.
Fay, John A. (J'40) (ACM), Jr. Prod. Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 6449 Wilkinson, North Hollywood, Calif.
Fay, Joseph Edmund (J'39) (C), Mgr., U.S. Employment Serv.; *for mail,* 211 Wyoming Ave., West Pittston, Pa.
Fay, Oscar Karl ('43) (CKZSGF), Prod. Supt., Twin Branch Plant, Ind. & Mich. Elec. Co., Mishawaka, Ind.
Fay, Robert Clovis (J'45) (ENSBJK), Ensign, U.S.N.R., PC-1147, F.P.O., San Francisco, Calif.
Fayerweather, Frederick Oakley, Jr. (J'38) (AC), Serv. Rep., Military Liaison, Wright Aero. Corp., 132 Beckwith Ave., Paterson 3, N.J.
Faymonville, Philip R. ('18; '25; '35) (CYD), Col., Gen. Staff Corps, Hdq., 4th Serv. Comd., 526 Old Post Office Bldg., Atlanta 3, Ga.
Feagles, Ralph L. ('12) (BCDM), Pat. Engr., Exec., Black Sivalls & Bryson, Inc., 2131 Westwood Ave., Oklahoma City, Okla.
Fearnside, Thomas Astley ('44) (EKORS), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 33 Wall St., Wellesley 81, Mass.
Fechheimer, Carl J. ('27) (KHB), 1930 N. Prospect Ave., Milwaukee 2, Wis.
Fedde, Arnold M. (J'33), 454—9th St., Brooklyn 15, N.Y.
Fedenia, John N. (J'40), Ensign, U.S.N.R., 2508 W. National Ave., Milwaukee 4, Wis.
Federici, Frank (A'44) (CLO), Dir., Associated Indus. Consultants, Broad St. Bank Bldg., Trenton 8, N.J.
Fee, Harold Rollins ('21; '28; '35) (CES), Office Mgr., Ingersoll-Rand Co., 11 Broadway, New York 4; *for mail,* 17 Wyndmere Rd., Mt. Vernon, N.Y.
Feeley, Edward Joseph, Jr. (J'41) (ABH), 30 Rhode Island Ave., N.W., Washington, D.C.
Fegel, Arthur C. ('41) (WMCJ), Mfg. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 9 English Village, Cranford, N.J.
Feicht, Edward R. ('07; '17) (SFOT), Head, Firm of Edward R. Feicht, Cons. Engrs., 111 Old Lancaster Rd., Bala-Cynwyd, Pa.
Feicht, Edward R., Jr. (J'41) (AMC), 1st Lt., Army Air Forces, Air Tech. Serv. Comd., Wright Field, Dayton, Ohio; *for mail,* 111 Old Lancaster Rd., Bala-Cynwyd, Pa.

Feiereisen, William John (J'43) (ASEFKO), U.S. Navy, Hdq. Sqd., Fleet Air Wing 3, F.P.O., New York, N.Y.; *for mail,* 207 James St., Beaver Dam, Wis.
Feige, William Frederick (J'35) (NBHJC), Designing Engr., H. R. Krueger Co., 1469 E. Grand Blvd., Detroit; *for mail,* 26395 Humber St., Huntington Woods, Mich.
Feiker, Fred'k M. (A'12) (CT), Dean, Sch. of Engrg., The George Washington Univ., Washington, D.C.
Feil, George William, Jr. (J'44) (NJBAZ), Research Engr., Aluminum Research Labs., Aluminum Co. of Am., New Kensington, Pa.
Fein, Mitchell (J'45), Dir. of Engrg., Consltd. Mgmt. Consultants, 521—5th Ave., New York 17, N.Y.
Feinberg, Milton (J'42) (AMHBS), 1451 S. Beverly Dr., Los Angeles 35, Calif.
Feiner, Hyman L. (J'40) (CTMAGL), Ch. Engr., Consultant, S. Liebovitz & Sons, Inc., 75 Leonard St., New York 13, N.Y.; *for mail,* 2312 N. 2nd St., Harrisburg, Pa.
Feiner, Mark Anthony ('42) (CMLDJN), Secy., P. Feiner & Sons, Inc., 522 W. 45th St., New York 19, N.Y.
Feinstein, Lester (J'40) (BJM), 226 B. 141st St., Belle Harbor, L.I., N.Y.
Feitl, Rudolph Raymond (J'43) (BN), Engr., Otis Elevator Co., 260—11th Ave., New York, N.Y.; *for mail,* 5405 Bergenline Ave., West New York, N.J.
Fehr, Roy Benner ('14; '19; '23) (MNJBEK), Dir., Engrg. & Sales, Schaefer Elec. Products Co., 5680—12th St., Detroit 8; *for mail,* 704 Forrestdale Rd., Royal Oak, Mich.
Felber, George S. (J'40) (JLM), Specifications Engr., Am. Can Co., 320 Elizabeth Ave., Newark 8; *for mail,* 636 Buchanan St., Hillside, N.J.
Felberg, Leonard (J'37) (AEM), Foreman, Republic Aviation Corp., Farmingdale; *for mail,* 942 E. 14th St., Brooklyn 30, N.Y.
Felch, Robert Irving (J'41) (CMSJHL), Systems Analyst, Indus. Engrg. Dept., RCA-Victor Div., Radio Corp. of Am., Camden, N.J.; *for mail,* 8617 Evergreen Ave., Chestnut Hill, Philadelphia 18, Pa.
Feld, Louis (J'41) (SHNOEC), Mech. Engr., Centrifugal Engrg. Div., Worthington Pump & Mchy Corp., Harrison; *for mail,* 390 Hawthorne Ave., Newark, N.J.
Feldman, Abram M. ('00) (O), Retired; 320 Central Park W., New York 25, N.Y.
Feldman, Morton Marvin (J'43) (BKS), 938 S. Townsend St., Syracuse, N.Y.
Feldman, Nathan R. (J'42) (LMSJZ), Sr. Engr., Research & Devel., Philco Corp., C & Tioga Sts., Philadelphia 34; *for mail,* 926 Tabor Rd., Philadelphia 41, Pa.
Felgar, James Huston ('16) (CGKL), Dean Emeritus, College of Engrg., Univ. of Okla., Norman, Okla.
Feliu, Carlos J. (J'39) (CSFLDZ), Mgr., Indus. Mchy. Div., Garcia Commercial Inc., Stop 6½, Fernandez Juncos Ave., San Juan; *for mail,* Box 363, 13 Prince St., San Juan 2, P.R.
Felix, Samuel Palmer, Jr. (J'40) (CJM), Asst. Supt., North Plant, De Laval Steam Turbine Co., Trenton 2, N.J.; *for mail,* The Molecule-Yardley Rd., Morrisville, Pa.
Felker, George F. ('13; '15; '18) (MS), V.P., Crosby Steam Gage & Valve Co., Rm. 1728, 165 Broadway, New York 6, N.Y.
Fell, Hugh Parrish ('15; '19) (LWMDS), Prod. Mgr., Casein Div., Borden Co., 350 Madison Ave., New York 17, N.Y.
Fellers, Clark I. (J'41) (MNLOCB), Plant Engr., Navy Ord. Div., Eastman Kodak Co., 50 Main St. W., Rochester 4; *for mail,* 322 Lake Ave., Rochester 6, N.Y.
Fellinger, Robert Cecil (J'44) (NSJMBZ), 320 Lewis St., Burlington, Iowa.
Fellows, Julian Robert ('41) (FKMJ), Prof. Mech. Engrg., Univ. of Ill., Urbana, Ill.
Fellows, Olin Burt ('19; '35) (CMB), Pres., Ideal Wrapping Mch. Co., 81 Sprague Ave., Middletown, N.Y.
Felt, Wright L. ('38) (CSEFH), Regional Engr., Fed. Works Agency, 1206 Santee, Los Angeles 1; *for mail,* 256 El Camino Dr., Beverly Hills, Calif.
Felten, Jos. M. ('27) (BCL), R.F.D. 1, Kingston, Ga.
Felton, George W. ('21) (CJM), 252 Commonwealth Ave., Boston 16, Mass.
Fendel, Fred A. (J'41) (NCYAJ), Lt. (j.g.), U.S.N.R., Staff, Fleet Air Wing 8, Naval Air Sta., Alameda, Calif.; *for mail,* 151 Nixon Ave., Staten Island 4, N.Y.
Fendrich, V. William (J'40) (CGR), Assoc. Inspr. of Naval Matls., 30 Church St., New York 7; *for mail,* 474—8th St., Brooklyn, N.Y.
Feng, Tze-Pei ('45) (DLO), c/o F. D. Loo, Bank of Kiangsu, 371 Kiangse Rd., Shanghai, China.
Fennel, Charles K. (J'40) (CDL), Sales Engr., Internatl. Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Fennell, Arthur Richard, Jr. (J'38) (HJM), Sr. Draftsman, Design Gages, Greenfield Tap & Die Corp., Sanderson St.; *for mail,* 42 Abbott St., Greenfield, Mass.
Fenning, James F. ('45), 515 Jackson Ave., Bridgeport, Conn.
Fenton, Clifford Monroe ('45) (BCDLNS), Design Supvr., De Laval Steam Turbine Co., Nottingham Way, Trenton 2; *for mail,* 1928 Pennington Rd., Trenton 8, N.J.
Fenwick, Harold Helvie ('29) (GCA), Assoc. Prof., Univ. of Louisville, 3rd & Eastern Pkwy., Louisville 8, Ky.
Ferar, Robt. (J'34), Engr., Testing Lab., U.S. Radiator Co., 127 N. Campbell; *for mail,* 2689 Boston Blvd., Detroit, Mich.
Ferguson, Allan R. (J'34) (BAMC), Mgr., Aircraft Devices Div., Foote Bros. Gear & Mch. Corp., 5225 S. Western Blvd., Chicago; *for mail,* 338 N. Ashland Ave., La Grange, Ill.
Ferguson, Cedric Keith (J'43) (KSH), Lt. (j.g.), S-E, U.S.N.R., Engrg. Exper. Sta., Annapolis; *for mail,* 418 Severn Ave., Eastport, Md.
Ferguson, Donald Adams (J'45) (AES), Ensign, U.S.N.; *for mail,* 428 Yale Ave., New Haven 15, Conn.
Ferguson, Donald McHardy (J'39) (MCD), Methods & Stands. Engr., E. D. Jones & Sons Co., Depot St., Pittsfield, Mass.
Ferguson, Hardy Smith ('99; F'41) (HLS), Partner, Hardy S. Ferguson & Co., 200—5th Ave., New York 10, N.Y.
Ferguson, Harry J. (J'41) (CHK), Plant Engr., Resisto Pipe & Valve Co., 262 Bridge St., East Cambridge; *for mail,* 243 Metropolitan Ave., Roslindale, Mass.
Ferguson, James Rex (J'43) (AKNHEW), Engrg. Designer, N. Am. Aviation, Inc., 5601 Imperial Way, Inglewood; *for mail,* 848 S. Oxford Ave., Los Angeles 5, Calif.
Ferguson, John Emil (J'40) (MC), Mch. Designer, Marathon Corp., Menasha; *for mail,* 508 Oak St., Neenah, Wis.
Ferguson, John F. ('31) (OSRLKN), Prin., Ferguson Engrs., 17 Queen St. E., Toronto 1, Ont., Can.
Ferguson, Louis A., Jr. ('21; '35), Customer's Serv. Mgr., Commonwealth Edison Co., 72 W. Adams St., Chicago, Ill.
Ferguson, Morris S. (J'44) (DEJNR), Engr., Indus. Hygiene, Dust & Fume Control, Am. Smelting & Refining Co., Garfield; *for mail,* Apt. 10, 153—2nd Ave., Salt Lake City 3, Utah.
Ferguson, Ray T. (J'38) (JSL), Ch. Engr., Graham Mfg. Co. Inc., 415 Lexington Ave., New York 17, N.Y.
Ferguson, Reginald ('29) (MJCNFA), Dir., Charge of Prod., F. Perkins (Diesels) Ltd., Queen St., Peterborough, Northants, England.
Ferguson, Robert Bruen ('19; '26) (LKNCDS), R.F.D. 1, Glassboro, N.J.
Ferguson, Robert Ireland (J'39) (JMBCND), Tool Engr., Greenfield Tap & Die Corp., Sanderson St., Greenfield, Mass.; *for mail,* 5202 W. Roosevelt Dr., Milwaukee 10, Wis.
Ferguson, William ('32) (KS), Asst. Supvg. Engr., Travelers Ins. Co., 700 Main St., Hartford 15, Conn.
Ferguson, William Burder ('43) (ACW), 52 Garden Pl., Brooklyn 2, N.Y.
Fernald, Ernest Mercer ('18; '28) (BKSEFO), Prof. Mech. Engrg., Lafayette College; *for mail,* R.D. 2, Easton, Pa.
Fernald, Henry B., Jr. (J'32) (ABYSJ), Metal. Coordination Supvr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* 81 Gordonhurst Ave., Upper Montclair, N.J.
Fernandez, A. (J'40), Cia. Azucarera America, Central America, Oriente, Cuba.
Fernow, Bernhard Edward ('28) (EFHKS), Prof. & Head, Mech. Engrg. Dept., Clemson College, Clemson, S.C.
Fernstrom, F. S. ('17; '31; '35) (CDJMY), Mgr., Spec. Serv. Dept., Ernst & Ernst, 123 S. Broad St., Philadelphia; *for mail,* 5515 Wissahickon Ave., Philadelphia 44, Pa.
Ferrara, Norman ('45) (BCDJLS), Head Gen. Engrg. Dept., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City 2; *for mail,* Box 683, New Brunswick, N.J.
Ferrari, Frank A. (J'38) (SEFR), Engr., Maint. Dept., Symington-Gould Corp., 20 Symington Pl., Rochester 11; *for mail,* 44 Madison St., Rochester 8, N.Y.
Ferrari, Luigi Mario (J'38) (MCBD), Asst. Ch. Engr., Indus. Engr., Pressuretube, Inc., 609 W. 134th St., New York 31; *for mail,* 3354 Radcliff Ave., Bronx 67, N.Y.
Ferrary, Ferdinand F. (J'37) (LM), Mech. Engr., Ch. of Layout & Design, Ferrary Mar. Repairs, 278 Pearl St., New York; *for mail,* 538 E. 37th St., Brooklyn, N.Y.
Ferré, Luis Alberto ('40) (BCK), V.P., Charge Prod., Porto Rico Iron Works, Inc., Real St., Playa Ponce, P.R.
Ferretti, Alfred John ('17; '24; '30) (BFK), Prof. Mech. Engrg., Northeastern Univ., 360 Huntington Ave., Boston 15, Mass.
Ferris, Charles E. ('04; H'42), Dean of Engrg., Univ. of Tenn.; *for mail,* 2551 Kingston Pike, Knoxville, Tenn.
Ferris, Edwin Alden ('24; '42) (SFKC), Asst. Mgr., Charge of Serv., Serv. & Erection Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 109 Hobart St., Ridgefield Park, N.J.
Ferris, Howard E. (J'39) (EFCLS), V.P., Charge Operas., S. Atlantic Gas Co., 114 Barnard St., Savannah, Ga.
Ferris, John P. ('26; '35) (CLWMNB), Dir., Commerce Dept., Tenn. Valley Authority, Knoxville; *for mail,* 11 W. Circle Rd., Norris, Tenn.
Ferris, John Riker (J'40) (AEBHFJ), Sr. Test Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* 166 Orange Rd., Montclair, N.J.
Ferris, John Thomas ('21; '30) (CDM), Ch. Indus. Engr., Graver Tank & Mfg. Co., Inc. & Phoenix Mfg. Co., 4809 Todd Ave., East Chicago; *for mail,* 224 Dyer Blvd., Hammond, Ind.
Ferris, Roy Merk (J'45) (MNOL), Asst. Zone Engr. Magnolia Petroleum Co. (Socony-Vacuum Oil Co., Inc.); *for mail,* Box 668, Beaumont, Tex.
Ferris, Walter ('07) (BGHLSY), V.P., Dir. of Engrg., Oilgear Co., 1403 W. Bruce St., Milwaukee 4, Wis.
Ferry, John M. ('24; '35) (OSF), Bldg. Engr., N.Y. Tel. Co., 140 West St., New York 7; *for mail,* 201 W. 79th St., New York 24, N.Y.
Ferry, Ralph M. ('12; '21) (ACJM), Mgr., Tenn. Operas. Exec., Aluminum Co. of Am., Box 465, Alcoa, Tenn.
Fersing, Leif ('30; '35), Research Engr., Jones & Lamson Mch. Co.; *for mail,* 81 Wall St., Springfield, Vt.
Fertig, Edward John (J'30) (SFAJC), Diamond Power Specialty Corp., 271 Madison Ave., New York, N.Y.; *for mail,* 266 Grand Ave., Leonia, N.J.
Fertig, John Landon ('42) (RECF), Gen. Mech. Inspr., Atchison, Topeka & Santa Fe Ry. Co., Motive Power Bldg., Topeka, Kan.; *for mail,* 705 Belleview Ave., La Junta, Colo.
Fessenden, Edwin Allan ('08; '14) (SKBF), Retired; 140 Oakwood Ave., Troy, N.Y.
Fetscher, John Joseph ('41; '44) (KSENHO), Engr., George G. Sharp, Naval Arch., 30 Church St., New York 7, N.Y.
Fetske, William August (J'42) (MNCE), 10 West End Pl., Elizabeth 2, N.J.
Fetters, George Henry (J'36) (LNSBD), Engr., Bristol Processing Corp., Beaver Dam Rd., Bristol; *for mail,* 400 S. Bellevue Ave., South Langhorne, Pa.
Feuchter, Robert John ('27; '35) (YHO), Partner, Cranston Molding Co., 16 Manhasset, Cranston; *for mail,* c/o R. L. Jacob, 124 Wentworth Ave., Providence 5, R.I.
Few, E. Liddon ('34), Wick Cottage, 12 Westleigh Ave., Putney, S.W. 15, England.
Feyling, Per L. F. ('35), Wks. Mgr., Whitehead Metal Products Co. of N.Y., Inc., 235 Bridge St., Cambridge; *for mail,* 140 Gilbert Rd., Belmont, Mass.
Fezandié, Eugène Hector ('22; '26; '30) (ABEFKO), Assoc. Prof., Stevens Inst. of Tech., Hoboken, N.J.
Fiala, Francis William (J'38) (NBHYJ), Asst. Ch. Engr., Thomas C. Wilson, Inc., 21-11—44th Ave., Long Island City 1; *for mail,* 186-32 Ilion Ave., St. Albans 12, N.Y.
Fiala, Sigmund N. (J'25), Engrg. Dept., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.; *for mail,* 171 Satterthwaite Ave., Nutley, N.J.
Fickes, Alfred Comp ('44) (CJLEMS), Asst. Sales Mgr., Buffalo Tank Corp., 744 Broad St., Newark 2, N.J.; *for mail,* 83 Norwood Ave., Staten Island 4, N.Y.
Fidelius, Walter R. (J'24) (CMJL), Asst. Ch. Engr., Optimus Equip. Co., Matawan; *for mail,* 69 W. Front St., Red Bank, N.J.
Fidler, Isaac ('40) (EST), Cons. Engr., 217 Willowbrook St., High Point, N.C.
Fiedler, Fred Stanton (J'45) (ACEFHS), Asst. Engr., Southwest Sales Constr., Union Oil Co. of Calif., 617 W. 7th St., Los Angeles 14; *for mail,* 710 S. Walnut, Inglewood, Calif.
Fiege, Henry J. ('20; '27) (CMDLJ), Gen. Foreman, Waterbury-Farrel Fdy. & Mch. Co., 453 Bank St., Waterbury 10; *for mail,* 43 Birchwood St., Waterbury 66, Conn.
Field, Crosby ('15; '16; '21; F'38) (BYLJKM), Pres., Flakice Corp., 360 Furman St., Brooklyn 2, N.Y.
Field, Emmet J. (J'33) (CMDNWJ), Asst. Branch Mgr., Internatl. Harvester Co., 23—6th St., S.E.; *for mail,* 84 Linden Dr., Mason City, Iowa.
Field, Ernest George ('19; '24; '30) (CDJMT), Prin., E. G. Field & Associates, 1101 Hurt Bldg., Atlanta 3; *for mail,* 670 Park Dr., N.E., Atlanta, Ga.
Field, John ('38), c/o W. Corlett, 39 Grosvenor St., Middle Brighton, Victoria, Australia.
Field, Kenneth Adelbert (J'42) (ESH), Design Draftsman, Moore Turbine Div., Worthington Pump & Mchy. Corp.; *for mail,* 47 Elm St., Wellsville, N.Y.
Field, Michael (J'38) (BJHA), Sr. Tech. Engr., Research Dept., Cincinnati Milling Mch. Co.; *for mail,* 6408 Evelyn Dr., Mt. Washington, Cincinnati 30, Ohio.
Field, R. Anthony (J'44) (CDM), Indus. Engr., Eastman Kodak Co., Kodak Park, Rochester 13, N.Y.; *for mail,* Box 6836, Cleveland, Ohio.
Field, Russell Watts, Jr. (J'40) (CM), 2nd Lt., Army Air Forces, U.S.A., Pilot's Rm., Army Air Field, Harlingen, Tex.; *for mail,* 39 E. Manning St., Providence 6, R.I.
Field, William Thompson ('31) (HORS), Maj., Corps of Engrs., U.S.A., Repairs & Utilities Branch, 6th Serv. Comd., 20 N. Wacker Dr., Chicago; *for mail,* 1466 Lake Ave., Wilmette, Ill.
Fielder, Harold Sydney ('44) (NCBKHJ), Mgr. of Engrg., Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6; *for mail,* 1111 Lake St., Evanston, Ill.

Fields, David Eugene ('43) (CEKLM), V.P., Sales Mgr., Tulsa Boiler & Mchy. Co., Box 1978, Tulsa 1; *for mail,* 2304 S. Lewis Ave., Tulsa 5, Okla.
Fields, Paul William (J'41) (CDMA), 56 Heiskell Ave., Wheeling, W.Va.
Fieman, Leonard (J'39) (AKO), Asst. Gp. Engr., Boeing Aircraft Co., 7755 E. Marginal Way, Seattle 14; *for mail,* 4550—18th Ave., Seattle 5, Wash.
Fierro, Santos ('33; '35), Ave. Juarez 2718, Chihuahua, Chih., Mex.
Fieweger, George (J'43) (A), Ensign, U.S.N., Salvage & Supply Officer, U.S.S. *Vent,* ARS-29, F.P.O., San Francisco, Calif.; *for mail,* 4272 Holly Ave., St. Louis 15, Mo.
Figenschou, Rasmus Aage Hagen (J'45) (BOZ), Rep., Bergens Telefon Compagni, Bergen, Norway; *for mail,* 3 E. 75th St., New York 21, N.Y.
Filbert, Howard C., Jr. (J'46) (CMY), Naval Ord. Lab., Naval Shipyard, Washington, D.C.; *for mail,* 5002 Denmore Ave., Baltimore 15, Md.
Files, Carl Winfred (J'45), Prof., Mech. Engrg. Dept., Tex. Agric. & Mech. College, College Station, Tex.
Filippone, Francis S. (J'39) (CDJ), Asst. Prod. Mgr., Heintz Mfg. Co., Front St. & Olney Ave., Philadelphia 20, Pa.; *for mail,* 212 Lyons Ave., Newark 8, N.J.
Fillman, Charles W. ('41; '43) (CMABDZ), Box 1855, Miami, Fla.
Filter, Curtis F. (J'46) (ABM), Mech. Engr., Mich. State Highway Dept.; *for mail,* 522 Washtenaw, Lansing, Mich.
Fimbel, Paul N. (J'42) (MWDCB), Prod. Engr., Overhead Door Co., Hillside 5; *for mail,* 1926 Oakwood Pkwy., Union, N.J.
Finch, Frank Richard ('20) (GBNWD), Prof. Mechanism & Drawing, College of Engrg., Univ. of Mich.; *for mail,* 1619 S. University Ave., Ann Arbor, Mich.
Finch, Rogers Burton (J'44) (TCL), Textile Div., Mass. Inst. of Tech., Cambridge, Mass.
Finch, Stanley B. ('43) (CMS), East. Devel. Engr., Pabst Brewing Co., 221 N. La Salle St., Chicago 1, Ill.; *for mail,* 14 Garfield Pl., Poughkeepsie, N.Y.
Finch, Volney Cecil ('28) (AEHKNS), Prof. Mech. Engrg., Stanford Univ.; *for mail,* 639 Mirada Ave., Stanford University, Calif.
Finck, Herbert Ferdinand ('42) (FHS), Power & Hyd. Engr., Ecusta Paper Corp., Pisgah Forest; *for mail,* 418 E. Main St., Brevard, N.C.
Fincke, Donald M. ('23; '30; '35) (CMJD), Asst. Works Mgr., Underwood Corp., 581 Capitol Ave., Hartford 6, Conn.
Findlater, Stevenson ('27; '34; '35) (BDFHKM), Asst. to Pres., Natl. Tube Co., 1802 Frick Bldg., Pittsburgh 30; *for mail,* 39 Irwin Dr., Pittsburgh 10, Pa.
Findley, William Nichols (J'37) (BYHWAN), Asst. Prof. Theoretical & Applied Mechanics, 302a Talbot Lab., Univ. of Ill., Urbana, Ill.
Fine, Bernard M. ('14; '19; '35), Midvale Co., Nicetown; *for mail,* Apt. 20, Bel-Air, 427 W. Chelten Ave., Philadelphia 44, Pa.
Fine, Lewis ('20; '35) (EJM), Supt., Mech. Repair Shops, Bethlehem Steel Co.; *for mail,* 1325 Church St., Bethlehem, Pa.
Fine, Maurice Edward ('36; '45) (KOS), Mech. Engr., Charge Air Conditioning, Giffels & Valle, Inc., 1000 Marquette Bldg., Detroit; *for mail,* 4625—2nd Blvd., Detroit 1, Mich.
Finger, Harold Ben (J'44) (AHBKN), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 4371 W. 143rd St., Cleveland 11, Ohio.
Finger, Karl E. (J'41) (BJL), Lt., 1107 Talley Rd., Wilmington 168, Del.
Fingerhut, Solomon (J'45), Aircraft Engr., Designers for Industry, Inc., Cleveland; *for mail,* 3294 Clarendon Rd., Cleveland Heights 18, Ohio.
Fingeroot, Ben (J'40), Instr. on Aircraft Engines, Chanute Field, Ill.
Finigan, John (J'39), Aviation Cadet, Aviation Cadet Det., Class 41-3A, Chanute Field, Ill.
Fink, Ferdinand ('27), Supvg. Engr., Mountain Ice Co., 100 Sylvan Ave., Newark, N.J.
Fink, George Edward ('35) (CS), Asst. Gen. Supt., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 88-23—238th St., Bellerose 6, N.Y.
Fink, George Rupert ('43), Pres., Natl. Steel Corp., Tecumseh Rd., Ecorse, Detroit, Mich.
Fink, Milton (J'36), Engr., Westinghouse Elec. Elev. Co., 150 Pacific Ave., Jersey City, N.J.; *for mail,* 65-36 Wetherole St., Forest Hills, N.Y.
Finke, F. W., Jr. (J'40), 315 E. 238th St., New York, N.Y.
Finke, Howard E. (J'42) (AMJ), Engr., Gen. Elec. Co., 62 Tremont, Everett; *for mail,* 282 Puritan Rd., Swampscott, Mass.
Finkel, Jacob Jay ('17; '17; '35) (CMO), Gen. Mgr., Serv. Engr., Harbor Inn Serv. Inc., 116th & Beach Channell Dr., Rockaway Beach; *for mail,* 221—136th St., Belle Harbor, N.Y.
Finkel, Julian B. (J'37) (NJMDC), Asst. Plant Engr., Tool Design, J. S. Thorn Co., 20th & Allegheny Ave., Philadelphia 32, Pa.
Finkl, Charles William (J'43) (SHFEAL), Plant Engr., A. Finkl & Sons Co., 2011 Southport Ave., Chicago 14, Ill.
Finkle, Earl (J'41) (NMBEJY), Design Engr., Naval Ord. Lab., Naval Shipyard, Washington 25; *for mail,* 3209—19th St., N.W., Washington 10, D.C.
Finkle, Frederick Cecil ('14) (HSOCE), Cons. Engr., F. C. Finkle Engrg. Offices, 805 N. Crescent Dr., Beverly Hills, Calif.
Finlay, Robert B. (J'44) (ACP), U.S.N.; 345 Pondfield Rd., Bronxville, N.Y.
Finlay, Walter Stevenson, Jr. ('16; F'36) (OS), *Manager,* '22, *Vice-President,* '22-'24; Exec. V.P., J. G. White Engrg. Corp., 80 Broad St., New York 4, N.Y.
Finlayson, James Colin (J'41) (AHMCD), Flight Lt., Aero. Engr., Royal Canadian Air Force, Test & Devel. Estab., R.C.A.F. Sta., Rockcliffe, Ont., Can.
Finn, George Andrew, Jr. (J'45), Lt. (j.g.), U.S.C.G.; *for mail,* 26 Valleywood Rd., Cos Cob, Conn.
Finnegan, Joseph Bernard ('20) (AFWY), Prof. Fire Protection Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.
Finnegan, Thomas Joseph ('44) (SJFKZ), Chem. Engr., Buffalo-Niagara Elec. Corp., 300 Elec. Bldg., Buffalo 3, N.Y.
Finneran, J. E., Jr. (J'36) (CJM), Die & Tool Designer, Superior Die Tool & Mch. Co., 115 W. Frankfort St.; *for mail,* 89 S. Dawson Ave., Columbus, Ohio.
Finnerty, Frank C. (J'40) (CMYO), Indus. Engr., Corrigan, Osburne & Wells, Inc., 60 E. 42nd St., New York 17, N.Y.; *for mail,* 79 E. 1st St., Clifton, N.J.
Finnerty, Frank Joseph (J'42) (CLMYB), Hanford Engr. Works, Hanford, Wash.; *for mail,* 521-B E. 85th St., New York, N.Y.
Finney, Burnham (A'41) (CDM), Editor, *American Machinist,* McGraw-Hill Publ. Co., 330 W. 42nd St., New York 18, N.Y.
Finney, Wallace R. ('21; '28) (EFMO), Pipe Line Adviser, Standard Oil Co. of N.J., 30 Rockefeller Plaza, New York 20, N.Y.
Finnie, Jack Douglas (J'45) (CFP), Shell Oil Co., 116—8th Ave. W., Calgary, Alta., Can.
Finster, George Charles (J'36) (BCM), 215—52nd St., West New York, N.J.
Fiore, Hannibal Salvatore (J'43) (ACMKHF), Ensign, U.S.N.R., Naval Ord. Lab., Naval Gun Factory, Washington 25, D.C.; *for mail,* 311 Fairview Ave., Fairview 4, N.J.
Firey, Joseph Carl (J'41) (AEFSZ), Lt. (j.g.), U.S.N.R., U.S.S. *Nereus,* AS-17, F.P.O., San Francisco, Calif.; *for mail,* 6226 Ravenna Ave., N.E., Seattle 5, Wash.
Firing, Wilhelm ('28) (EKS), Ch. Engr., Thunes Mek, Verksted, Box 225, Oslo, Norway.
Firth, David ('44), Dir. of Engrg., Dodge Mfg. Corp., 500 S. Union St., Mishawaka, Ind.
Fisch, Albert, Jr. (J'40) (DKLSZ), Plant Engr., Publicker Alcohol Industries, 1429 Walnut St., Philadelphia; *for mail,* 6814 Large St., Philadelphia 24, Pa.
Fisch, Jacob ('30; '36) (BCHN), Ch. Design Engr., S. Morgan Smith Co., York, Pa.
Fisch, Roland Vincent (J'44) (HMS), Mech. Engr., Equip. Designer, Ohio Rubber Co., Ben Hur Ave., Willoughby, Ohio.
Fischbach, Joseph Winston (J'39) (BKHNAJ), Ord. Engr., Rocket Sec., Ballistics Research Lab., Aberdeen Proving Ground; *for mail,* 235 Seneca Ave., Havre de Grace, Md.
Fischer, Adelbert F., Jr. (J'38), 122 Elmer St., Westfield, N.J.
Fischer, F. K. ('40) (SKH), Cent. Sta. Steam Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13, Pa.
Fischer, Howard Louis ('30) (YABCGO), Pat. Atty., Engr., 1286 University Ave., St. Paul 4, Minn.
Fischer, Hugo Edward (J'38) (NDEBH), Draftsman, Keystone Steel & Wire Co., South Bartonville; *for mail,* 215 Marquette Ave., Peoria 6, Ill.
Fischer, Kermit ('30; '36) (ZLHCBA), Pres., Fischer & Porter Co., County Line Rd., Hatboro 1; *for mail,* Dog Tail Rd., R.D. 1, Doylestown, Pa.
Fischer, Leander J. (J'37), Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 36 Clifton Ave., Marblehead, Mass.
Fischer, Robert John (J'43) (ABS), Pvt., Army Ground Forces, Replacement Depot 1, Ft. George G. Meade, Md.
Fischer, Udo Wilfried (J'39) (CMDLKA), V.P., Schutte & Koerting Co., 12th & Thompson Sts., Philadelphia 22; *for mail,* 6904 Wissahickon Ave., Philadelphia 19, Pa.
Fischer, Walther Carl ('37) (ESK), Installation Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7, Wis.
Fischman, Samuel O. (J'33) (JWMC), Pres., Century Equip. Co., 2240 W. Ogden Ave., Chicago 12, Ill.
Fish, Edwards Russell ('14; F'36) (SFOKCE), *Manager,* '23-'26, *Vice-President,* '26-'28; Retired; 30 Laurel Ave., Windsor, Conn.
Fish, Floyd Hamilton, Jr. (J'43) (BKN), 1st Lt., Antiaircraft Radar Officer, Instr., Army Univ. Center, Oahu, T.H., A.P.O. 957; *for mail,* Box 702, Blacksburg, Va.
Fish, James Franklin (J'43) (A), Field Rep., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8, Conn.; *for mail,* Putnam Ave., Greenville, R.I.
Fish, Maurice J. ('45) (BKA), Adv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 83-43—118th St., Kew Gardens, N.Y.
Fishbein, John Jack (J'44) (AEF), 1533 Upshur St., N.W., Washington 11, D.C.
Fisher, A. W. (J'39), Aviation Cadet, Aviation Cadet Det., Turner Field, Albany, Ga.
Fisher, Benjamin James, Jr. (J'37) (HMZJ), 800 Whitesboro St., Utica 4, N.Y.
Fisher, David Alexander (J'31) (EKSFZ), Asst. Prof. Mech. Engrg., Engrg. Sch., Tufts College, Medford; *for mail,* 25 Sagamore Park, West Medford, Mass.
Fisher, David Francis (J'32) (MJC), Foreman, Am. Steel & Wire Co., 80 Hamilton Ave., Trenton; *for mail,* 101 Jackson St., Trenton 9, N.J.
Fisher, Elbert Clement ('13) (BDFSZ), Engr., Agar Container Div., Internatl. Paper Co., Whippany, N.J.
Fisher, Evan Dearborn (J'44), 204 Cedar Ave., Takoma Park, Md.
Fisher, Francis Powell ('30) (EFKO), Partner, Davis & Fisher, Engrs., 302 Grain Bldg., Saskatoon, Sask., Can.
Fisher, George H. B. (A'26) (BEH), Secy., Engrg. Dept., Canadian & Gen. Finance Co., Ltd., 25 King St. W., Toronto, Ont., Can.
Fisher, George Kenneth (J'36) (HKS), Sheffler-Gross Co., Drexel Bldg., Philadelphia 6, Pa.
Fisher, Harland Stanley (J'42) (NMDBCJ), Devel. Engr., E. I. du Pont de Nemours & Co., Buffalo Ave.; *for mail,* 2480 Linwood Ave., Niagara Falls, N.Y.
Fisher, Harold Walling (J'45) (JRLA), Devel. Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 27 Lafayette St., Schenectady, N.Y.
Fisher, Harry Nelson (J'43) (AEHGN), Checker, Engrg. Design, Jacobs Aircraft Eng. Co., Queen & Baley Sts., Pottstown; *for mail,* 233 S. Reading Ave., Boyertown, Pa.
Fisher, Henry Donald ('07; '14) (SWOFLE), Mech. Engr., Westcott & Mapes Inc., 109 Church St., New Haven 5; *for mail,* 404 Yale Ave., New Haven 15, Conn.
Fisher, Howard C. ('12; '19; '20) (C), Pres., Treas., Cent. Engrg. & Constr. Co., 210 Main St., Pawtucket, R.I.
Fisher, James Franklin (J'33) (RMS), 676 N. Pearl St., Albany 4, N.Y.
Fisher, John Crocker (J'46) (BJ), Instr., Rm. 1-313, Mass. Inst. of Tech., Cambridge 39, Mass.
Fisher, Philip Stephen, Jr. (J'44) (ABCM), Lt. Comdr., U.S.N.R., 10A-45P-A5L, Bldg. 679, Naval Air Sta., Pensacola, Fla.
Fisher, Raymond Moore, Jr. (J'45) (SBAONC), Test Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 730 Union St., Schenectady 5, N.Y.
Fisher, Richard Henry (J'42) (AKEFM), Sr. Test Engr., Wright Aero. Corp., Beckwith Ave., Paterson 3; *for mail,* 21 N. 13th St., Haledon, Paterson 2, N.J.
Fisher, Russell Todd ('23; '35) (TDC), Pres., Secy., Natl. Assn. of Cotton Mfrs., 80 Federal St., Boston 10, Mass.
Fisher, Samuel Scott (J'40) (RNJCMF), Engr. Draftsman, Am. Car & Fdy. Co., Arch St., Milton; *for mail,* R.F.D. 1, Allenwood, Pa.
Fisher, W. H. (J'38) (FRS), Norfolk & West. Ry. Co., 1105 Reynolds Bldg., Winston-Salem, N.C.
Fisher, Warren Howard (J'43) (NBCESM), Mech. Engr., West. Union Tel. Co., 60 Hudson St., New York, N.Y.; *for mail,* 424 Franklin Ave., Hasbrouck Heights, N.J.
Fisher, William John ('19; '33) (CDHJLM), Pres., Gen. Mgr., A.B. Farquhar Co.; *for mail,* 106 N. Marshall St., York, Pa.
Fisher, William Wallace ('10; '21; '21) (CM), Mgr., Listing Mch. Div., Monroe Calculating Mch. Co., 555 Mitchell St., Orange, N.J.
Fishkin, Charles Ira (J'41) (S), Natl. Aniline Div., Allied Chem. & Dye Corp., Box 975, Buffalo 5, N.Y.
Fishkin, Jack (J'45) (ELY), Lt. (j.g.), U.S.N.R., Box 21, Navy 121, F.P.O., New York, N.Y.; *for mail,* 204 Smith St., Perth Amboy, N.J.
Fishman, Bernard (J'42) (MACNH), Ch. Engr., Star Auto Radio Accessories, Inc., 450 W. 12th St., New York; *for mail,* 1057—58th St., Brooklyn 19, N.Y.
Fisk, Gustaf Leonard ('15; '19) (J), 271 Central Park West, New York 24, N.Y.
Fiske, James M. (J'39), Ch. Engr., Food Mchy. Corp., Lakeland; *for mail,* 603 S. Orleans, Tampa, Fla.
Fissore, Oscar Francis (J'40) (EFHKS), Mar. Engr., with George G. Sharp, Naval Arch., 30 Church St., New York 7; *for mail,* 1531 Richmond Rd., Staten Island 4, N.Y.
Fitch, Roland Charles ('37; '43) (SFKHC), Results Engr., Twin Branch Plant, Ind. & Mich. Elec. Co., Mishawaka; *for mail,* 140 E. Farneman Ave., South Bend 14, Ind.
Fitch, William Kountz ('28) (HS), V.P., Dravo Corp., 300 Penn Ave., Pittsburgh 22, Pa.
Fitts, James Logan ('13) (BEHKSZ), Cons. Mech. Engr., Warren Webster & Co., 17th & Federal Sts., Camden; *for mail,* 6175 Cedar Ave., Merchantville, N.J.
Fitz, Ervin Moul ('01; '16) (RSNCME), Retired; 58 South St. E., Worthington, Ohio.
Fitze, Maurice Edward (J'26) (SFZK), Sr. Test Engr., Power Plants, Wis. Elec. Power Co., 231 W. Michigan St., Milwaukee 1; *for mail,* 10841 W. Luther Ave., Hales Corners, Wis.
Fitzgerald, Charles ('18; '22) (HECKAY), Asst. to V.P., Pipe Line Dept., Sinclair Refining Co., 901 Fair Bldg., Ft. Worth 2, Tex.

Fitz-Gerald, Gerald ('17; '25) (FZETLK), Dist. Mgr., Maxon Premix Burner Co., 50 Church St., New York 7, N.Y.; *for mail,* 3 Aberdeen Pl., Radburn-Fairlawn, N.J.

FitzGerald, J. Morgan ('27) (FHRS), 354 Berkley St., Philadelphia 44, Pa.

Fitzgerald, James ('42) (SHKCBD), Comdr., Civil Engr. Corps, U.S.N., Washington, D.C.; *for mail,* 284 Highland Ave., Ravenna, Ohio.

Fitzgerald, John ('43) (SFCZ), Ch. Engr., Newport Elec. Corp., 159 Thames St., Newport, R.I.

Fitzgerald, Michael J. (J'42) (BHJ), Tool Engr., Collins Radio Co., 855—35th St., N.E.; *for mail,* 123—36th St. Dr., S.E., Cedar Rapids, Iowa.

Fitzgerald, Ray Munn, Jr. (J'44) (BCE), Mech. Engr., Styrene Div., Dow Chem. Co., Velasco; *for mail,* 1519 W. 7th St., Freeport, Tex.

Fitzgerald, Thomas Witt, Jr. (J'43) (ANMCDL), Mch. Designer, Eastman Kodak Co., Kodak Park, Rochester 18; *for mail,* 235 S. Union St., Spencerport, N.Y.

Fitzhugh, Robert R. (J'36) (BKS), c/o Nicaro Nickel Co., Nicaro, Oriente, Cuba.

Fitzke, William Otto (J'41) (SJH), Power Supt., Republic Steel Corp., 8th St., N.E., Canton 3, Ohio.

Fitzpatrick, Frank Richard ('19) (CRSF), Asst. to Pres., Superheater Co., 60 E. 42nd St., New York 17, N.Y.

Fitzpatrick, John R. (J'41), Apprentice Seaman SV/7-G, Midshipmen's Trng. Sch., Notre Dame Univ., South Bend, Ind.

Fitzpatrick, John T. (J'43), 76 Franklin St., Morristown, N.J.

Fitzsimmons, A. M. R. ('34; 35) (C), 101 Northbrook Lane, Bethesda 14, Md.

Fitzsimmons, Joseph Harold (J'33) (SKY), Design Engr., Fed. Shipbldg. & Dry Dock Co., 744 Broad St., Newark, N.J.; *for mail,* 1181 Sheepshead Bay Rd., Brooklyn, N.Y.

Fitzsimmons, Saml. D. ('24), Ft. Myers, Fla.

Fixman, Carl M. (J'34) (EKSB), Mech. Engr., Public Works Div., Naval Shipyard, Mare Island; *for mail,* 927 Route Blvd., Albany 6, Calif.

Flack, Alonzo ('15; '35) (CDLM), Chmn., Treas., Mgmt. Counselor, Emerson Engrs., 30 Rockefeller Plaza, New York 20, N.Y.

Flad, Edward ('91) (O), Cons. Engr., Mem. Miss. River Comm., 828 U.S. Court House, St. Louis 1, Mo.

Flagg, Chas. N., Jr. ('15), Pres., C. N. Flagg & Co. Inc., 79 Griswold St., Meriden, Conn.

Flagge, Andrew (J'46) (JLM), Box 5651, Route 2, Des Plaines, Ill.

Flagle, Charles Denhard (J'40) (ABC), Design Engr., Westinghouse Elec. Corp., Lester, Pa.

Flaherty, Robert (J'41) (GJN), Design Engr., Miller Ptg. Mchy. Co., 1117 Reedsdale St., Pittsburgh 12; *for mail,* 49 Riverside Gardens, Pittsburgh 2, Pa.

Flanagan, James Henry (J'44) (AJECM), Lt. (j.g.), U.S.N.R.; *for mail,* 12 Lambert Rd., Belmont 78, Mass.

Flanagan, Robert G. (J'38), Tennant Co., 2905 Gulf Bldg., Houston, Tex.

Flanagan, Walter N. ('17; '20; '30) (JSFCED), Engrg. Consultant, Carnegie-Ill. Steel Corp., Carnegie Bldg., Pittsburgh 30, Pa.

Flanders, Ralph Edward ('08; '14; F'36) (NCR BML), *Manager,* '26-'29, *Vice-President,* '29-'31, *President,* '35; *Worcester Reed Warner Medallist,* '34; Pres. on Leave, Jones & Lamson Mch. Co. & Bryant Chucking Grinder Co.; *for mail,* Box 477, Springfield, Vt.

Flar, Henry (J'43) (MCSLDJ), Expediting Engr., M. W. Kellogg Co., Foot of Danforth Ave.; *for mail,* 307 Chapel Ave., Jersey City 5, N.J.

Flateboe, E. I. ('14; '35), Gen. Mgr., Sumner Iron Works, Box 1006, Everett, Wash.

Flater, Harold ('22), Works Mgr., Internatl. Harvester Co., Norrköping, Sweden.

Flather, Frederick Arthur ('92) (TCMHSD), Treas.-Mgr., Boott Mills, Milk St., Boston; *for mail,* Mansur St., Lowell, Mass.

Flatow, Walter W. (J'45) (CJLMN), 914—1st Ave., Spokane 6, Wash.

Flavin, Edward J. (J'36) (KMB), Mech. Engr., Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.

Flaws, David Banks (J'37) (LJ), Planning Engr., Lummus Co., 420 Lexington Ave., New York 17; *for mail,* 1514 Metropolitan Ave., New York 62, N.Y.

Fleet, Samuel L. ('19; '35) (O), Engr., Estimator, S. L. Fleet Co., Inc., 51 Madison Ave., New York 10, N.Y.

Fleischer, Irving (J'44), Hyd. Engr., Greer Hydraulics, Inc., 1776 Broadway; *for mail,* 108 S. Village Ave., Rockville Centre, N.Y.

Fleischer, Thomas (J'42) (JSM), Lt. (j.g.), U.S.N.R., Engrg. Officer, LSM-303, F.P.O., New York, N.Y.; *for mail,* 1141 N. New St., Bethlehem, Pa.

Fleischmann, Walter Ludwig ('44) (JBSZCK), Metal. Engr., Turbine Generator Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.

Fleisher, Walter L. (J'42) (BFKOTY), Pres., Air & Refrigeration Corp., 475—5th Ave., New York 17, N.Y.

Fleming, Sir Arthur P. M. ('28), Dir., Metro.-Vickers Elec. Co., Ltd., Trafford Park, Manchester 17, England.

Fleming, Burritt Gray ('31; '45) (ABCDFH), Field Engr., Wright Aero. Co., Cincinnati 15; *for mail,* 6618 Elm St., Mariemont, Cincinnati 27, Ohio.

Fleming, David H., Jr. (J'41) (BJM), Devel. Engr., Linde Air Products Co., 646 Freylinghuysen Ave., Newark; *for mail,* 31 E. Highland Ave., Orange, N.J.

Fleming, John Taylor (J'28) (CJM), Asst. to Eng. Dir., John Brown & Co. Ltd., Engrs. & Shipbldrs., Clydebank, Glasgow; *for mail,* 15 Westbourne Dr., Bearsden, Scotland.

Fleming, Laurence Trent ('20; '35) (ROCB), Chief Rail Branch, Research & Devel. Div., Office of Chief of Trans., War Dept., The Pentagon, Washington 25, D.C.; *for mail,* 105 La Fonda Ave., Box 86, Route 1, Santa Cruz, Calif.

Fleming, Robert Martin ('43) (SKFC), Power Dept. Head, Diamond Alkali Co., Painesville, Ohio.

Fleming, Wills M. ('05; '09) (HM), Ch. Engr., Reciprocating Pump Engrg. Div., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 51 Bennett Ave., Arlington, N.J.

Fleser, Russell Harrison, Jr. (J'42) (ABHMSK), Aircraft Draftsman, Douglas Aircraft Co., Inc., Lakewood & Carson, Long Beach; *for mail,* 440 N. Citras Ave., Los Angeles 36, Calif.

Flesher, Marion George ('28; '35) (SFEKJO), Engr., Westinghouse Elec. Corp., 20 N. Wacker Dr., Chicago; *for mail,* 11721 Longwood Dr., Chicago 43, Ill.

Fletcher, Edwin Hubert (J'45) (ENMJHC), Engrg. Draftsman, Bethlehem Steel Co., 3075 Richmond Terrace, Staten Island 3; *for mail,* 52 Delafield Ave., Staten Island 1, N.Y.

Fletcher, James ('24), Engr., Babcock & Wilcox Co., Barberton, Ohio.

Fletcher, James Loren ('32; '41) (CKBYJM), Dir. of Devel., Carrier Corp., 300 S. Geddes St., Syracuse 1, N.Y.

Fletcher, Leonard J. ('41) (CEM), Trng. Library, Bldg. X (Librarian), Caterpillar Tractor Co., Peoria 8, Ill.

Fletcher, Northrop Rogers ('24; '30; '35) (EFJ NOS), Mech. Engr., Asst. Diesel Consultant, Ebasco Services Inc., Rm. 1100, 2 Rector St., New York 6, N.Y.

Fletcher, Ralph Leslie ('26) (CFSK), V.P., Providence Gas Co., 100 Weybosset St., Providence 1, R.I.

Fletcher, Thomas Frederic ('45) (FKS), Supvg. Engr., Cía. Swift de La Plata, Puerto La Plata, F.C.S., Argentina, S.A.

Flickinger, John Hodgdon (J'46) (CM), Mfg. Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia; *for mail,* 814 N. 63rd St., Philadelphia 31, Pa.

Fliendner, Angus T. ('38) (NBMJH), Design Engr., Cleveland Tractor Div., Oliver Corp., 19300 Euclid Ave., Cleveland; *for mail,* Erie Rd., Willoughby, Ohio.

Flink, Axel A. (J'35) (CMO), Cons. Engr., Prod. Mgmt. Engrg. Associates, Inc., Rm. 375, Russ Bldg., San Francisco; *for mail,* 2708 Via Campesina, Palos Verdes Estates, Calif.

Flinn, A. V. (DHMBFG), Managing Dir., Adamson-Alliance Co., 165 Fenchurch St., London, E.C. 3; *for mail,* Doveleys Rd., Pendleton, Manchester, England.

Flinn, William Stuart (J'45), Engrg. Research & Devel. Dept., Elliott Co., Jeannette; *for mail,* 25 Hazel Dr., Pittsburgh 16, Pa.

Flinner, A. O. ('29; '35; '35) (EFS), Assoc. Prof. Mech. Engrg., Kan. State College of Agric. & Applied Sci., Manhattan, Kan.

Flint, Amos Henry ('42) (EGLOSZ), Ch. Engr., Corn Products Refining Co., 333 N. Michigan Ave., Chicago 1, Ill.

Flint, Charles Kimball ('27) (CS), V.P., Gen. Mgr., Kodak Park Works, Eastman Kodak Co., Rochester 13, N.Y.

Flint, Eugene Marcus ('41) (GCMY), Maint. Engr., Rand McNally & Co., 536 S. Clark St., Chicago 5; *for mail,* 1931 S. 10th Ave., Maywood, Ill.

Flint, Thomas (J'40) (NBJMDG), Ch. Engr., J. W. Greer Co., 119 Windsor St., Cambridge; *for mail,* Simon Willard Rd., Concord, Mass.

Flitner, David Perkins (J'45), U.S. Navy; Greenville, N.H.

Flood, Henry, Jr. ('16; '35), Apt. 407, 36 Riverside Ave., Red Bank, N.J.

Floodeen, Eddy ('24; '35) (ONMKBJ), Cons. Engr., 1504 Smith Tower, Seattle 4, Wash.

Floor, Erik ('45) (HOS), Erik Floor & Associates, 139 N. Clark St., Chicago 2, Ill.

Floor, Urban (J'45) (HBEANS), Engr., Woodward Governor Co., 5001 N. 2nd Ave., Rockford, Ill.

Floreen, Edward Dore (J'41) (MCNKDF), Design Engr., Rudy Furnace Co.; *for mail,* 106 Green, Dowagiac, Mich.

Florek, Ralph Stephen (J'41) (CMY), Ch. Engr., Moore Enameling & Mfg. Co., West Lafayette, Ohio.

Florin, Urban Albert (J'43) (ASJBHK), Engrg. Apprentice, Indus. Engrg. Dept., E. J. Brach & Sons, 4600 W. Kinzie, Chicago; *for mail,* 3912 N. Kenneth, Chicago 41, Ill.

Florschutz, Fritz Edmund ('44) (NJBCMO), Chief, Drawing Office, Switch Gear Div., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 12 Hawthorne Rd., Pittsburgh 21, Pa.

Flounders, James McCay (J'41) (YCM), Sales Engr., B. F. Goodrich Co., 500 S. Main St., Akron, Ohio.

Flower, Ad ('24) (SFKECO), Sales Mgr., New York Office, Edge Moor Iron Works, Inc., 7204 Empire State Bldg., New York 1, N.Y.; *for mail,* 819 N. Harrison St., Wilmington 32, Del.

Flowers, H. Fort ('39) (DMR), Pres., Gen. Mgr., Differential Steel Car Co., Findlay, Ohio.

Flygare, Carl G., Jr. (J'41), 116 Forest St., Worcester 5, Mass.

Flyke, Milton Joseph (J'45) (SNKMCH), 606 S. Scoville Ave., Oak Park, Ill.

Flynn, Charles Andrew ('16) (BCDMOZ), V.P., Treas., Flynn-Hill Elevator Corp., 6 Howard St., New York 13, N.Y.

Flynn, Edward David ('38) (LHCY), Ch. Engr., Oliver United Filters, Inc., 2900 Glascock St., Oakland 1, Calif.

Flynn, John Bernard ('44) (BCHJMY), Exper. Engr., Fisher Body-Ternstedt Div., Gen. Motors Corp., 6307 W. Fort St., Detroit 9; *for mail,* 2253 Gladstone Ave., Detroit 6, Mich.

Flynn, John V. (J'40) (CFS), 49 Dixie Dr., Penns Grove, N.J.

Flynn, Michael Henry ('13) (CMNDB), Methods & Layout Engr., Royal Typewriter Co., Inc., 150 New Park Ave., Hartford 6; *for mail,* 113 Whitney St., Hartford 5, Conn.

Flynn, Roland W. ('41) (EJLSBM), Div. Lub. Engr., Gulf Oil Corp., 17 Battery Pl., New York 4, N.Y.

Flythe, John Thomas, Jr. (J'43) (JNMC), Jr. Design Engr., Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 5; *for mail,* 2703 Chamberlayne Ave., Richmond 22, Va.

Foard, Castie W. ('36) (BAZM), Physicist, Devel. Dept., Camera Works, Eastman Kodak Co., 333 State St., Rochester 4, N.Y.

Fobian, George Willard (J'42), Design Engr., Lister-Blackstone, Inc., 1706 S. 68th, West Allis; *for mail,* 129 N. 91st Pl., Milwaukee, Wis.

Fobian, Robert John (J'40) (HMCAEL), Mech. Engr., Devel. Work, Oilgear Co., 1403 W. Bruce St., Milwaukee 4; *for mail,* 1733 S. 76th St., West Allis 14, Wis.

Fodor, Nicholas ('24; '30) (ERHAO), Pres., Ch. Engr., Diesel Engrg. & Mfg. Corp., 200 N. Laflin St., Chicago 7; *for mail,* 835 Greenwood Ave., Wilmette, Ill.

Foell, Charles Francis (J'41) (ERS), Tech. Writer, Editor, Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4; *for mail,* 32-18 Bell Blvd., Bayside, N.Y.

Fogarty, Wm. Bailey ('45), Comdr., U.S.N., Retired; 624 Lincoln Ave., Cincinnati, Ohio.

Fogel, Jerome J. (J'40) (MCJD), Prod. Engr., Automotive Specialty Co., Inc., 382-386 Jefferson St., Brooklyn 6; *for mail,* 105 Buckingham Rd., Brooklyn 26, N.Y.

Fogelson, Emile ('27) (NAJDSY), Charge Mch. Design, Prod. Methods, Inc., 48 E. 43rd St., New York 17; *for mail,* 410 Central Pk. W., New York 25, N.Y.

Fogelsonger, Robert B. (J'40) (MSC), 100 Maple St., Clarence Center, N.Y.

Fogg, Oscar H. ('07; '13), Chmn. of Exec. Com., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.

Fogg, William Raymond ('21) (OL), Arch. Engr., Partner, Ballinger Co., 105 S. 12th St., Philadelphia 7; *for mail,* 41 W. Stratford Ave., Lansdowne, Pa.

Fogler, Ben Baker ('12; '18; '22) (OLT), Sr. Engr., Arthur D. Little, Inc., 30 Memorial Dr., Cambridge 42, Mass.

Fogwell, J. Wray (J'40) (ABJ), Mech. Engr., Cuno Engrg. Corp., 80 S. Vine St., Meriden; *for mail,* 47 Berkley Court, Southington, Conn.

Foley, Edward Michael (J'41) (SLKE), Lt. (j.g.), U.S.S. *Midway,* F.P.O., New York, N.Y.; *for mail,* 3443 N.E. 35th Ave., Portland, Ore.

Foley, Glenroy B. ('28; '40) (C), Mech. Engr., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* Box 104, Ridlonville, Me.

Foley, Jas. A. ('29), Mech. Bridge Engr., Hudson County Engrg. Dept., Court House; *for mail,* 439 Fairmont Ave., Jersey City, N.J.

Foley, Matthew J. (J'40), Jr. Mech. Engr., with W. F. Tuttle, Works Engr., Am. Rolling Mill Co.; *for mail,* 2367 Tytus Ave., Middletown, Ohio.

Foley, Raymond B. ('45) (DHKS), Supvr., Dust Collector Research, Am. Blower Corp., Detroit 32, Mich.

Foley, W. S. ('37) (HECJ), Asst. Engr., Distribution & Transmission, Bur. of Water, City of Philadelphia, 829 City Hall Annex, Philadelphia; *for mail,* 750 Adams Ave., Philadelphia 24, Pa.

Foley, Walter Joseph ('10; '23) (HDO), Dredging Engr., Stand. Dredging Corp., 800 Central Bldg., Los Angeles 14, Calif.

Follari, Salvatore (J'46), Petroleum Heat & Power Co., Stamford, Conn.

Folmsbee, Clyde Henry ('39) (RCBMKH), Asst. Mech. Engr., Am. Car & Fdy. Co., Oak & 9th Sts., Berwick; *for mail,* 344 E. Main St., Bloomsburg, Pa.

Folsom, Richard Gilman ('29; '40) (HABZ), Assoc. Prof. Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.

Foltz, Raymond D. ('21; '25; '35) (CFJKMS), Pres., M. H. Detrick Co., 21 West St., New York 6, N.Y.

Folz, Joseph James (J'40) (LKSFBC), Process Engr. (Project), E. I. du Pont de Nemours Co., 9498 Nemours Bldg., Wilmington, Del.

Fomilyant, Anatol A. ('45) (CBJLMH), Partner, All Am. Fabricators, 2201 E. 51st St., Los Angeles 11; *for mail,* 3554 Mountain View Ave., Pasadena 8, Calif.

Fonda, Frederick Anthony (J'43) (CFS), Lt. (j.g.), U.S.N.R., U.S.S. *Greene*, F.P.O., San Francisco, Calif.
Foody, John Joseph ('44) (SEKHCO), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 6; *for mail*, 3321 Bruckner Blvd., New York 61, N.Y.
Foord, James L. ('15) (R), R.D. 5, Canoga Rd., Auburn, N.Y.
Foote, Avery George ('43), Regional Engr., Frigidaire Div., Gen. Motors Corp., 1250—53rd St., Oakland 8; *for mail*, 365 Vermont Ave., Berkeley 7, Calif.
Foote, Earl Eastman ('43) (CKS), Ch. Engr., Consumers Cent. Heating Co., 108 E. 11th St.; *for mail*, 3412 N. 28th St., Tacoma, Wash.
Foote, Frederick David (A'36) (RAW), Partner, Alloys Devel. Co., 1102 Park Bldg., Pittsburgh 22, Pa.
Foote, Lawrence Rossiter (J'43) (BMOSJA), Manhattan Project, Det. 2, Hanford Engr. Works, Richland, Wash.
Foote, Leonard ('22; '28), 1025 Lincoln Ave., Palo Alto, Calif.
Foote, Samuel David ('46), Res. Engr., British Am. Oil Co., Ltd., Box 279, Calgary, Alta., Can.
Foote, William Russell (J'37) (AB), Engr., Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.
Forbes, F. Powell ('38; '45) (WCS), Mgr., Engrg. & Fabrication Div., Weyerhaeuser Timber Co., Box 629, Newark 1, N.J.
Forbes, James Bellingham ('35; '35) (LSCYW), Alco Products Div., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Forbes, John A., Jr. (J'39) (AHS), Engr., Hercules Powder Co., Radford; *for mail*, 510 Progress St., Blacksburg, Va.
Forbes, John Dubler (J'41) (WCNM), Pres., Triboro Woodworking Mchy., Inc., 147 Grand St., New York 13, N.Y.
Forbes, Robert Thomson ('21) (FKS), New England Rep., Green Fuel Economizer Co., Peabody Engrg. Co., Am. Engrg. Co., Buell Engrg. Co., Marley Co., Equipoise Controls, 201 Devonshire St., Boston 10; *for mail*, 448 Woodward St., Waban 68, Mass.
Forbes, William Gordon ('45), Lub. Engr., Tide Water Associated Oil Co., 17 Battery Pl., New York 4; *for mail*, 215-37—43rd Ave., Bayside, L.I., N.Y.
Ford, Albert D., Sr. ('38) (EFKS), Head, Mech. Engrg. Dept., Univ. of New Mex., Albuquerque, New Mex.
Ford, Albert Duane, Jr. (J'42) (EKS), Westinghouse Elec. Corp., 309 S.W. 6th Ave., Portland 4, Ore.
Ford, Arthur Samuel ('18; '35), Dir., Directorate of Tech. Practice, Dept. of Munitions, "Burke House", 340 Collins St., Melbourne, C. 1, Australia.
Ford, Edward Eugene (J'41) (CMT), Engrg. Dept., Fulton Bag & Cotton Mills, 170 Blvd., S.E.; *for mail*, 1230 Briarwood Dr., N.E., Atlanta, Ga.
Ford, H. Stanley ('21; '35) (KSFCEL), V.P., Gen. Mgr., Bigelow-Liptak Corp., 2842 W. Grand Blvd., Detroit 2, Mich.
Ford, Harold Percy ('24) (OSCJND), Mech. Engr., U.S. Engrs., Textile Tower, Seattle; *for mail*, 3312 Empire Way, Seattle 44, Wash.
Ford, Henry ('15), *Holley Medallist*, '36; Founder, Ford Motor Co., Detroit, Mich.
Ford, John Henry ('39) (BFKS), Engrg. Dept., Container Corp. of Am., 111 W. Washington St., Chicago 2, Ill.
Ford, Louis R. ('20), Editor, *Motorship*, Diesel Publications, Inc., 192 Lexington Ave., New York 16, N.Y.
Ford, Norman Michael (J'42) (NESM), Devel. Engr., Schirgun Corp., 136 W. 52nd St., New York; *for mail*, 1219 43rd St., Brooklyn 19, N.Y.
Ford, Robt. E. ('19) (BHM), Partner, Luther Ford & Co., 100 N. 7th St.; *for mail*, 2540 Humboldt Ave. S., Minneapolis, Minn.
Ford, Thomas Henry (J'45) (KSHBNE), Mech. Engr., with E. E. Ashley, Cons. Engr., 10 E. 40th St., New York; *for mail*, 61-59—78th St., Elmhurst, L.I., N.Y.
Ford, William Blanchard, Jr. (J'44) (HKS), Test Engr., Ingersoll-Rand Co., Morris St.; *for mail*, Box 12, c/o Shipman, Phillipsburg, N.J.
Ford, William F. (A'45) (ABEHNO), Mar. Supt., Bur. of Reclamation, U.S. Dept. of Int.; *for mail*, Box 425, Coulee Dam, Wash.
Fordham, Nicholas Edmond ('30) (CHJORY), Retired; 537 N. Adams St., Glendale 6, Calif.
Foreman, Arthur Sherlock (J'37) (ACMEZS), Spec. Lecturer, Dept. of Mech. Engrg., Univ. of Toronto, Toronto 5; *for mail*, 217 St. George St., Toronto, Ont., Can.
Foresman, Robert All ('19) (BCDJK), Retired; 846—13th Ave., Prospect Park, Pa.
Forfar, Donald Mirick (J'13) (FSACOH), Mech. Engr., Power Piping, Grinnell Co., Inc., 240—7th Ave. S., Minneapolis 15; *for mail*, 4817 Emerson Ave. S., Minneapolis 9, Minn.
Forhecz, Joseph Anthony (J'45) (NBJMCO), Engr., & Supvr., Forhez Tool Co., 10-20 Orient Way, Rutherford; *for mail*, 434 Lafayette Ave., Passaic, N.J.
Forman, Armstrong Haslup (J'32) (MCLOD), 61 Burkshire Rd., Towson 4, Md.
Forman, Walter Wilbur ('29; '35) (SFECH), Asst. Gen. Supt. of Operas., Conn. Light & Power Co., 250 Freight St., Waterbury; *for mail*, South St., Middlebury, Conn.
Formanek, Frank J. (J'43) (EFSZ), Serv. & Erection Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 1627 Bogart Ave., Bronx 61, N.Y.
Fornes, Gaston G. (J'36) (BGM), Capt., U.S. Military Academy; *for mail*, 220 Lee Rd., West Point, N.Y.
Forrest, H. Dean ('37; '46) (SFEKH), Power Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Forrest, James ('17; '35) (COWJFA), Dept. Head, Contract Estimating, Babcock & Wilcox Co., Stirling Ave.; *for mail*, 664 St. Clair Ave., Barberton, Ohio.
Forrest, James Duncan (A'45) (BC), Safety Dir., Burrard Vancouver Dry Dock Co., Ltd., Ft. McLean Dr.; *for mail*, 1535 W. 13th Ave., Vancouver, B.C., Can.
Forrester, Joseph George (J'45) (COMD), Cons. Indus. Engr., Consltd. Mgmt. Consultants, 521—5th Ave., New York 17; *for mail*, 2266 Andrews Ave., New York 53, N.Y.
Forsberg, Roy Halvar (J'44) (AMB), Student, College of Engrg., New York Univ., New York; *for mail*, 119 Main St., Irving-on-Hudson, N.Y.
Forsman, Elmer John (J'29) (SFEKR), Dist. Serv. Engr., Babcock & Wilcox Co., 1203 Candler Bldg., Atlanta 3, Ga.
Forssell, Alfred G. (J'22) (CHM), V.P., Gen. Mgr., Morris Mch. Works; *for mail*, 18 Sunset Terrace, Baldwinsville, N.Y.
Forstall, Alfred Edmond ('99) (OEKY), 5 Champlain Terrace, Montclair, N.J.
Forstall, Walton, Jr. (J'43) (BSKC), 169 Commonwealth Ave., West Concord, Mass.
Forster, Hans Walter (J'45) (BKMSN), Design Engr., Enzinger Union Corp., Hardpan Rd., Angola; *for mail*, Paxon Rd., Eden, N.Y.
Forsyth, Stuart L. (J'37) (CMLDSA), Mgr., Elec. Htg. Dept., Westinghouse Elec. Corp., 5915 Green St., Emeryville 8, Calif.
Forsythe, Alfred Vernon ('42) (SJBF), Asst. Ch. Oper. Engr., Kansas City Power & Light Co., 115 Grand Ave., Kansas City 6, Mo.
Forsythe, Paul Ernest ('40; '42) (CHDBJL), Spec. Asst. to Mgr. of Sales & Engrg., West. Gear Works, 417—9th Ave. S., Seattle 4, Wash.
Fort, Tomlinson ('43) (CFS), Mgr., Cent. Sta. Sales, Westinghouse Elec. Corp., East Pittsburg; *for mail*, 2345 Hollywood Dr., Pittsburgh 21, Pa.
Fortmann, Edward H. ('14; '22) (CEHJB), Gen. Mgr., Natl. Transit Co., 206 Seneca St., Oil City, Pa.
Forzley, Victor George (J'44) (WSOBC), Stress Analyst, Charles T. Main, Inc., 201 Devonshire St., Boston; *for mail*, 88 Locust Ave., Worcester 4, Mass.
Fosdick, William Powell ('16) (O), Partner, Fosdick & Hilmer, Cons. Engrs., 1703 Union Trust Bldg., Cincinnati 2, Ohio.
Fosheim, Ivan Victor (J'43) (BCE), Lt. (j.g.), U.S.N.R., Am. Handling Sch., Office 29, Dock Area, Camp Peary, Va.
Foss, Eugene N., 2nd (J'37) (CJK), Lt., U.S.N.R.; *for mail*, c/o B. F. Sturtevant Co., Hyde Park 36, Mass.
Foss, Feodore F. ('21) (CDFL), Dir., Research & Metal., Wheeling Steel Corp., Wheeling Steel Corp. Bldg.; *for mail*, Emerson Rd., Woodsdale, Wheeling, W. Va.
Fossier, Mike Walter (J'45) (AH), Student, Calif. Inst. of Tech., 1201 E. Calif. St., Pasadena 4, Calif.
Foster, Albert Cecil ('26; '45) (FCDKLM), Asst. Ch. Engr., Stand. Brands, Inc., 595 Madison Ave., New York 22, N.Y.
Foster, Arthur Rowe (J'45) (SNKZBF), 39 Rockledge Rd., Newton Highlands 61, Mass.
Foster, Charles ('17) (OSKDF), Cons. Engr., 316 Medical Arts Bldg., Duluth 2, Minn.
Foster, Charles Alfred B. (J'36) (EGS), Assoc. Prof., Va. Poly. Inst.; *for mail*, Box 65, Blacksburg, Va.
Foster, Dean Edward ('44), Supt., Matls. Dept., Bethlehem Supply Co., 805 E. Archer St., Tulsa; *for mail*, 1435 S. Carson Ave., Tulsa 5, Okla.
Foster, Earle Willard ('44; '45) (LMOSZ), Plant Engr., Canadian Top & Body Corp., Ltd., Louise St., Tilbury, Ont., Can.
Foster, Ernest H. ('85; '94), Dir., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Foster, Esty ('25; '26; '35), Pres., Natl. Lock Washer Co., 40 Hermon St., Newark 5, N.J.
Foster, Francis E. ('44) (JSMC), Metal. Engr., Grinnell Co., Inc., 260 W. Exchange St., Providence, R.I.
Foster, Gerald Pentland (J'40) (MC), Graduate Student, Sch. of Business, Stanford Univ., Stanford University, Calif.; *for mail*, 815 College Ave., Palo Alto, Calif.
Foster, J. S. ('91; '03) (BDHJNR), V.P., Lidgerwood Mfg. Co., 775 Lidgerwood Ave., Elizabeth, N.J.
Foster, Jack William (J'45) (MCEHSN), 22 North Ave., Webster, N.Y.
Foster, Leonard Colley (J'38) (CAB), Prod. Engr., Montreal Works, Defence Industries Ltd., 9400 St. Lawrence Blvd., Montreal, Que.; *for mail*, 30 Hazelton Ave., Toronto, Ont., Can.
Foster, Newton Smith (J'42) (BJYNLT), Engrg. Dept., Congoleum-Nairn, Inc., 195 Belgrove Dr., Kearny; *for mail*, 15 E. Pierrepont Ave., Rutherford, N.J.
Foster, Pell William, Jr. ('43), V.P., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Foster, Richard Bruce (J'39) (ASCN), 271 Armstrong Ave., Jersey City 5, N.J.
Foster, Robert George (J'42) (BCS), Supvg. Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.; *for mail*, c/o Gen. Elec. S.A., Caixa Postal 109, Rio de Janeiro, Brazil, S.A.
Foster, S. Lattimore, Jr. (J'35), 2852 Emogene St., Mobile 18, Ala.
Foster, Theodore George (J'41) (AEK), Capt., U.S.A., Hdq., Air Tech. Serv. Command, TSIAN-A, Area "B", Wright Field, Dayton, Ohio.
Foster, Tillman Richard ('44), Indus. Sales Engr., Carrier Corp., 408 Chrysler Bldg., New York 17, N.Y.
Foster, Walter Hurlé ('40) (BDFHN), Mech. Engr., Ch. Draftsman, Stand. Steel Works Div., Baldwin Loco. Works, Burnham; *for mail*, 435 Valley St., Lewistown, Mifflin Co., Pa.
Foulds, Charles VonHoffmann ('19; '30) (HSB ONE), Cons. Engr., Empresa de Energia Electrica, Apartado Nacional 106, Medellin, Colombia, S.A.
Foulds, Henry William ('43), Exec. V.P., Permutit Co., 330 W. 42nd St., New York 18; *for mail*, 5 Tory Lane, Scarsdale, N.Y.
Fournier, John Andrew (J'42) (ABC), Devel. Engr., Johnson & Johnson, 4949 W. 65th St., Chicago; *for mail*, 814 S. Vine St., Hinsdale, Ill.
Fournier, Thomas F. ('15; '35) (ABCE), Pres., Dir., Sunset Oil Co., 208 W. 8th St., Los Angeles 14, Calif.
Foust, John D., Jr. (J'39), Asst. Engr., Newman Mch. Co., Inc., 507 Jackson St.; *for mail*, 215 S. Mendenhall, Greensboro, N.C.
Fowden, Wm. ('13), Plant Mgr., S.D. Cement Plant; *for mail*, Box 146, Rapid City, S.D.
Fowler, Anson Jansen (J'42) (SJAMFC), 606 Washington Ave., Palmyra, N.J.
Fowler, Edward L. (J'30), Tech. Serv. on Mill Products, Internatl. Nickel Co., Inc., 67 Wall St., New York; *for mail*, 3555—88th St., Jackson Heights, L.I., N.Y.
Fowler, Francis Raynor (J'36) (BNA), Devel. Engr., Aero. & Mar. Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Fowler, Franklin Harper, Jr. (J'38) (BAJCZ), Structures Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail*, 239 N. Oraton Pkwy., East Orange, N.J.
Fowler, Grant Lunney ('39) (CDOM), Works Engr., Haynes Stellite Co., Lindsey, Kokomo, Ind.
Fowler, Henry Charles, Jr. ('30; '45) (CMGAB), Asst. Plant Mgr., Gray Mfg. Co., 16 Arbor St., Hartford, Conn.
Fowler, Jackson Estes (J'43) (SBKFAR), Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 41 Steinmetz Homes, Schenectady 4, N.Y.
Fowler, Kenneth W. (J'41) (JMC), Pvt., Inf. Replacement Trng. Camp, Co. D, 30th Bn., 1st Regt., Ft. McClellan, Ala.; *for mail*, 8 Mayflower Circle, Worcester 6, Mass.
Fowles, George Milton (J'39) (SC), Asst. Engr., Works Engrg. Dept., Prest-O-Lite Co., Inc., Box 1478, Indianapolis 6, Ind.
Fox, Alfred W. (J'37) (SFEH), Power Engr., Barrett Div., Allied Chem. & Dye Corp., 40 Rector St., New York 6; *for mail*, 412 Congress Ave., East Welleston, L.I., N.Y.
Fox, Benjamin ('23) (AKHBC), Supt., Engrg. & Design, Shipbldg. Div., Bethlehem Steel Co., Quincy 69; *for mail*, 5 Buckingham Rd., Wollaston 70, Mass.
Fox, Chas. Hust ('00), Pres., Ahrens-Fox Fire Eng. Co., 800 Evans St.; *for mail*, 2966 Erie Ave., Cincinnati, Ohio.
Fox, Charles Smith (J'37) (LSDZ), Salesman, Aluminum Co. of Am., 3615 Olive St., St. Louis 8; *for mail*, 17 Oakleigh Lane, Clayton 5, Mo.
Fox, Earle B., Jr. (J'39) (HKS), Field Mech. Engr., Constr. Dept., Austin Co., 19 Rector St., New York, N.Y.; *for mail*, 210 W. 5th St., Plainfield, N.J.
Fox, Frank W. ('19; '35), Chmn., Fuel Economy Comm., Tech. Serv. Div., Stand. Oil Co. of N.J., Elizabeth; *for mail*, 9 Fair Hill Rd., Westfield, N.J.
Fox, Franklin Hunt ('34; '43) (SOC), Plant Betterment Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.; *for mail*, Ashland Farm, Hatboro, Pa.
Fox, J. George (J'42) (FSHE), Mech. Engr., Plate Co., 2215—1st Ave. S.; *for mail*, 701—15th St. W., Birmingham, Ala.
Fox, John H. ('04) (CJL), Tech. Adviser to Pres., Pittsburgh Plate Glass Co., Grant Bldg.; *for mail*, University Club, Pittsburgh, Pa.
Fox, Oscar Martin ('44) (NBFJKC), Ch. Engr., Indus. Equip. Dept., Blaw-Knox Co., Box 1198, Pittsburgh 30, Pa.
Fox, Robert Burnap (J'44) (BEF), Lt., Engrg. Officer, U.S.S. *Rainier*, F.P.O., San Francisco; *for mail*, 1616 San Antonio Ave., Alameda, Calif.
Fox, Rudolph H. ('13; '19; '35), V.P., Vulcan Iron Works Co., 1423 Stout St., Denver, Colo.
Fox, William James (J'41) (SZEC), Draftsman, Bethlehem-Sparrows Point Shipyard, Inc., Sparrows Point 19; *for mail*, 7303 Dunlawn Court, Dundalk 22, Md.

Foy, Thomas Douglas (A'45), Supvr. of Cost Reduction, Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1908 Baker Ave., Schenectady, N.Y.

Foyn, Tor (J'43) (SKZD), Mech. Engr., Norsk Hydro, Solligaten 7, Oslo, Norway.

Fraas, Arthur P. (J'44) (AEKBNO), Asst. Prof. Automotive Engrg., Case Sch. of Applied Sci., Univ. Circle, Cleveland 6; *for mail,* 1139 Clifford Rd., Cleveland Heights 21, Ohio.

Fradenburgh, John ('45) (NTMJD), Design Engr., U.S. Automatic Box Mchy. Co., Inc., 18 Arboretum Rd., Roslindale, Boston 31, Mass.

Fraher, Brendan Joseph (J'43) (MCJRLO), Industry Supvr., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34, Pa.

Fraley, Earl John, Jr. (J'41) (GCNM), Lt., S(E), U.S.N.R., Naval Shipyard, Mare Island; *for mail,* 148 Tallac St., Vallejo, Calif.

Fram, Morris (J'37) (OMCDHY), Ch. Engr., Owner, Fram Engrg. Co., 1358 Carmona Ave., Los Angeles 35, Calif.

Frame, John S., Jr. (J'46) (EFS), Am. Engrg. Co., Philadelphia 25, Pa.

Frame, John Warren, III (J'43) (CMA), Exper. Flight Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 101 Winding Lane, Maple Park, East Hartford, Conn.

Frame, Roy Allen (J'46) (FLY), Box 1663, Santa Fe, New Mex.

Frame, William Melvin ('24; '35) (CJM), *Junior Award,* '27; Works Mgr., Spang-Chalfant Div., Natl. Supply Co., Ambridge, Pa.

France, Albert Finley (J'41) (CM), Assembly Dept. Mgr., Control Instrument Co., 67—35th St., Brooklyn; *for mail,* 525 E. 89th St., New York 28, N.Y.

France, Wm. Henry ('34) (CJM), Mgr., New Products Div., Am. Type Founders, Inc., 271 Madison Ave.; *for mail,* 277 Park Ave., New York, N.Y.

Francis, Edw. (J'39) (FS), Barrage Balloon Sch. Det., Camp Davis, N.C.; *for mail,* 131 N. 16th St., Bloomfield, N.J.

Francis, Joseph Snelson (J'45) (ONLJMD), Owner, Francis Co., 343 S. Dearborn St., Chicago 4, Ill.

Francis, T. M. ('25) (OLESFK), Cons. Engr., 334 Brown Marx Bldg., Birmingham 3, Ala.

Francis, Thomas Frederic ('44) (ACOMB), Engr. Test Pilot, Design Specialist, N. Am. Aviation, Inc., Inglewood; *for mail,* 3323 W. 152nd St., Gardena, Calif.

Francisco, Ferris LeRoy ('06; '12) (OYDLS), Partner, Francisco & Jacobus, 511—5th Ave., New York 17, N.Y.

Franck, Clarence Chrisman ('39; '41) (SBCE FA), *Student Award,* '28; Mgr., Land Turbine Engrg., Steam Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* 421 Cornell Ave., Swarthmore, Pa.

Franck, Roscoe Whitaker (J'40) (ARS), 1st Lt., Army Air Forces, Sqd. A, 463rd Base Unit, Geiger Field, Spokane, Wash.

Franck, Russell E. (J'37) (ZCAJY), Asst. Designing Engr., Instrument Transformer Div., West Lynn Works, Gen. Elec. Co., 40 Federal St., Lynn; *for mail,* Box 331, Lynnfield Center, Mass.

Francone, Edmund A. (J'41), Field Engr., Plant Engrg. Div., U.S. Rubber Co., 6600 E. Jefferson; *for mail,* 19143 Roscommon St., Detroit, Mich.

Frank, Austin Corman (J'36) (DHY), Engr., U.S. Engr. Office; *for mail,* 907 Forrester Ave., Albuquerque, New Mex.

Frank, Carl F. W. (A'30) (MCDO), Prod. Engr., Cincinnati Milling Mch. Co., South & Marburg Sts.; *for mail,* 3306 Claramont Ave., Cincinnati 9, Ohio.

Frank, David S. ('40) (FCLKZ), Supt., Pure Oil Co., Box 1798, Toledo 3, Ohio.

Frank, Edwin ('09; '25) (HSNEB), Apt. 2, 1943 N. Summit Ave., Milwaukee 2, Wis.

Frank, Graham M. ('41) (CMNOHJ), Mech. Engr., Jeffrey Mfg. Co., Columbus; *for mail,* 191 W. Dunedin Rd., Columbus 2, Ohio.

Frank, Max (J'40) (JDSKHZ), Design Engr., Colgate-Palmolive-Peet Co., 1 Newark Ave., Jersey City 2; *for mail,* 53 Hawthorne Terrace, Leonia, N.J.

Frank, Philip E. ('28; '34) (LESO), Constr. Engr., Sinclair Oil Corp., 630—5th Ave., New York 20, N.Y.

Frank, Robert M. ('34; '44) (OSDAFT), Cons. Engr., 1390 Nelson Ave., New York 52, N.Y.

Frank, Wallace Everett (J'42) (BKS), Kintnersville, Pa.

Franke, Karl John (J'39), Tool Designer, Atlantic Diesel Corp., Codwise Ave., New Brunswick; *for mail,* 711 Academy Terrace, Linden 7, N.J.

Frankel, Abraham (J'41), Technician, 4th Grade, 756th Engr. Parts Supply Co., A.P.O. 757, c/o Postmaster, New York; *for mail,* 811 New Lots Ave., Brooklyn 8, N.Y.

Frankel, George Joseph (J'44) (YLMBCD), Mech. Engr., Metaplast Corp., 205 W. 19th St., New York 11; *for mail,* 712 W. 176th St., New York 33, N.Y.

Frankel, Nathan Anderson, Jr. (J'45) (CMDJ), Indus. Engr., Conlon Corp., 1824 S. 52nd Ave., Cicero; *for mail,* 247 S. Kenilworth, Oak Park, Ill.

Franken, Thomas Lincoln (J'40) (MNCDJH), Mech. Engr. & Supt., Spec. Products Div., Lodge & Shipley, 800 Evans St., Cincinnati 1; for mail, 6238 Marie Ave., Cincinnati 24, Ohio.

Frankena, August (J'35) (LSDFCN), Plant Engr., Am. Sugar Refining Co., Key Highway, E., Baltimore 3, Md.

Frankenberg, Theodore T. (J'34) (FKS), Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.

Frankenfield, C. Walter (J'34) (EFAS), Supvr. Automotive Lab., Tide Water Associated Oil Co., E. 22nd St., Bayonne, N.J.

Frankenhoff, Anthony George (A'44) (CLW), Asst, Secy., Mgr., East. Div., Dicalite Co., 120 Wall St., New York 5, N.Y.

Frankenhoff, Charles Anthony ('19; '31) (CMO), 6 Hathaway Rd., Scarsdale, N.Y.

Frankenhoff, E. T. ('44), Dicalite Co., 756 S. Broadway, Los Angeles 14, Calif.

Frankhouser, Elmer V. ('45) (SFKHDZ), Supt. of Power, RCA Victor Div., Radio Corp. of Am., Front & Cooper Sts., Camden, N.J.; *for mail,* 174 Glencoe Rd., Upper Darby, Pa.

Frankland, George Edward (J'39) (MLY), Process Engr., Seeger-Sunbeam Corp.; *for mail,* 313 S. Rotherwood Ave., Evansville, Ind.

Franklin, Edward Jay ('13) (EFOS), 17036 Rancho St., Encino, Calif.

Franklin, George McCarty (J'46) (KLP), Partner & Engr., Franklin Mfg. Co.; *for mail,* 2026 N. 2nd St., Abilene, Tex.

Franklin, Paul Arnold ('25) (WCZGYA), Designer, John Hartell & Co., 101 Park Ave., New York 17; *for mail,* 1 Lincoln Pl., Port Washington, N.Y.

Franklin, Paul Edward (J'42) (BZNHYA), Cons. Engr., O. M. Franklin Serum Co., 109 N. Monroe St.; *for mail,* 400 Kentucky St., Amarillo, Tex.

Franks, Ernest Hughes (J'41) (ANJBFS), Mech. Engr., Engrg. Sec., Indus. Test Lab., Bldg. 121, Navy Dept., Naval Base Sta., Philadelphia 12; *for mail,* 830 Guenther Ave., Yeadon, Pa.

Franks, Frederick B. ('04), V.P., Gen. Mgr., Natl. Portland Cement Co., Brodhead; *for mail,* 906 Club Ave., Allentown, Pa.

Frankum, Jay Lester (J'41) (JKSFMC), Sr. Staff Engr., Fabricating Works, Aluminum Co. of Am.; *for mail,* Springbrook Rd., Alcoa, Tenn.

Fransema, John A. ('38) (DLJK), Indus. Engr., Day & Zimmermann, Inc., 15th & Chestnut Sts., Philadelphia 2; *for mail,* 225 E. Penn St., Philadelphia 44, Pa.

Fransson, Karl Elof (J'39) (ABHNSR), Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* Box 1687, Hartford, Conn.

Frantz, Vincent Alan (J'41) (JRC), Asst. to Mech. Engr., Fabricating Div., Bethlehem Steel Co.; *for mail,* 120 Lincoln St., Steelton, Pa.

Franz, Erwin E. ('30) (BCMYHA), Devel. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 3 Claremont Pl., Cranford, N.J.

Franz, Frederick Louis ('19; '28; F'46) (BNJ CMO), Cons. Mech. Engr., 53 Wallace St., New Haven 11; *for mail,* 2 Woodruff St., West Haven 16, Conn.

Franzen, Carl Johannes (J'41) (EAM), Samaritan Hosp., Troy; *for mail,* Richmondville, N.Y.

Franzen, Paul Raymond (J'45) (GNJLS), Design Engr., L. W. Claybourn, Inc., 231 S. La Salle St., Chicago 4; *for mail,* 4063 W. 58th Pl., Chicago 29, Ill.

Fraser, James Perry (J'44) (ASBC), Lt., U.S.N.R., Arresting Gear Officer, Air Dept., U.S.S. *Bozer,* CV-21, F.P.O., San Francisco, Calif.; *for mail,* 102 Prescott St., Clinton, Mass.

Fraser, Lee ('40) (CMYDJ), Managing Dir., Foreign Subsidiary, Internatl. Stand. Elec. Corp., 67 Broad St., New York 6, N.Y.

Fraser, Oliver B. J. ('33) (JFLKCB), Dir., Tech. Serv. on Mill Products, Internatl. Nickel Co., Inc., 67 Wall St., New York 5, N.Y.

Fraser, Theodore Harold (J'44) (MNC), Prod. Engr., Fairchild Camera & Instrument Corp., 88-06 Van Wyck Blvd.; *for mail,* 148-45—87th Rd., Jamaica, N.Y.

Fraser, William Charles Gordon (J'39) (MJC LD), Process Engr., Rogers Majestic Ltd., 622 Fleet St. W., Toronto; *for mail,* 33 Claxton Blvd., Toronto 10, Ont., Can.

Fratcher, George Edward (J'37) (HBJEN), Design Engr., Pressure Vessel Div., A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1; *for mail,* 6115A-W. Locust St., Milwaukee 10, Wis.

Frauenthal, Henry Lafayette, ('31; '34; '36) (HAB), Asst. Hydro. Investigator, Sanitation & Water Supply Div., Dept. of Pub. Works, Old Court House, Mineola, N.Y.

Frawley, Patrick Joseph ('43) (CFKOS), Asst. Supt., Steam Plants, No. States Power Co., 15 S. 5th St., Minneapolis 5; *for mail,* 1979 Palace Ave,. St. Paul 5, Minn.

Frayer, Leon Webster (J'41) (EBA), Tech. Analyst, Gen. Elec. Co., Thompson Rd., Syracuse; *for mail,* 14 Hawk St., Schenectady, N.Y.

Frazier, Quentin (J'42) (NBAMZW), Mech. Engr., Evans Lab., Sig. Corps, War Dept., Belmar; *for mail,* 564 Cedar Ave., West Long Branch, N.J.

Frear, Hugo Pinkney ('21) (ACDMOY), Cons. Naval Arch., Shipbldg. Div., Bethlehem Steel Co., 25 Broadway, New York; *for mail,* 5 Cambridge Lane, Manhasset, N.Y.

Freberg, Carl Roger (J'41) (BEA), Devel. Dept., Carrier Corp., S. Geddes, Syracuse; *for mail,* 811 Salt Springs Rd., Syracuse 3, N.Y.

Frech, Harry Edward, Jr. (J'40) (JABDL), Ensign, U.S.N.R., 7070 Lindell, University City 5, Mo.

Fredd, John V. (J'40) (MNAC), Design & Devel., Engrg. Dept., A. C. Spark Plug Div., Gen. Motors Corp., Flint 2; *for mail,* 5201 N. Dort Highway, Flint 7, Mich.

Frede, Charles F. ('18), Asst. V.P., Gen. Steel Castings Corp., Granite City, Ill.; *for mail,* 7931 Gannon Ave., University City 5, Mo.

Frederick, Frank J., Jr. (J'41), Jr. Engr., Gen. Regulator Corp., 165 Broadway, New York; *for mail,* 6th Ave., St. James, L.I. N.Y.

Frederick, Herbert S. ('43) (SEKFLO), Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 61 Charlesbank Rd., Newton 58, Mass.

Frederick, Robert William (J'45) (BEM), 619 Locust St., Turtle Creek, Pa.

Fredericks, Harold S. (J'37), Box 126, West Nyack, N.Y.

Fredin, Roy William (J'45) (AGMNO), Cons. Engr., Sierra Mfg. Co., 610 S. Arroya Pkwy., Pasadena 2; *for mail,* 1945 E. Foothill Blvd., Altadena, Calif.

Free, Albert V. ('43) (BGARHS), Head Teacher, Saunders Trades School Annex, Bd. of Ed., Yonkers; *for mail,* 94 Buckingham Rd., Yonkers 2, N.Y.

Freed, Dean Winslow (J'43) (MCD), Indus. Engr., Sylvania Elec. Products, Inc.; *for mail,* 137 W. 5th St., Emporium, Pa.

Freedenfeld, Martin David (J'46), 2958 Brighton 1st St., Brooklyn, N.Y.

Freedley, Paul ('20), 413 Warwick Rd., Haddonfield, N.J.

Freeland, Emile Charropin ('19; '26; '35) (BC FLT), Cons. Indus. Engr., W. R. Grace & Co., 7 Hanover Sq., New York 5, N.Y.

Freeland, Wesley William (J'35) (CMJNW), Supvr., Mech. Maint. & Design, "Dominion" Ammunition Div., Canadian Industries, Ltd.; *for mail,* Brownsburg, Que., Can.

Freeman, Albert W. (J'41) (EFC), Dorset, Ohio.

Freeman, Benj. W. ('15; '25), (ABM), Pres., Louis G. Freeman Co., 1819 Freeman Ave.; *for mail,* 2613 Handasyde, Cincinnati, Ohio.

Freeman, Clarke ('15; '22; '35) (CML), *Manager,* '38-'41, *Vice-President,* '41-'43; Box 1485, Providence, R.I.

Freeman, Evert Wendell ('30; '35) (DO), Plant Engr., Brown & Sharpe Mfg. Co., Promenade St., Providence; *for mail,* 42 Freeman Pkwy., Providence 6, R.I.

Freeman, Frederick Charles ('11; '16) (EFJRS), Pres., Providence Gas Co., 100 Weybosset St., Providence 1, R.I.

Freeman, Henry Livingston ('17) (NCAGHB), Designer, Engrg. Dept., So. Natural Gas Co., Watts Bldg., Birmingham; *for mail,* 1325 S. 19th St., Birmingham 5, Ala.

Freeman, Herbert S. ('15; '22), Mfg. Agt., 49 Darwin St., Rochester, N.Y.

Freeman, Hovey Thomas ('19; '26; '35) (CZO), Pres., Treas., Mfrs. Mutual Fire Ins. Co., 815 Grosvenor Bldg., Providence 1, R.I.

Freeman, Howard Gilbert (J'41) (HYJBNC), Dir., Research & Devel., Rockwood Sprinkler Co., 38 Harlow St., Worcester 5; *for mail,* 8 Claremont St., Worcester 3, Mass.

Freeman, Lewis D. ('13; '25), Examiner, R.R. Div., Reconstruction Finance Corp., Washington 25, D.C.; *for mail,* Apt. 227, Arlington Village Apts., Arlington, Va.

Freeman, Mathew L., Jr. (J'41), Box 245, State College, Miss.

Freeman, Myron Frederick ('25; '45) (SFK), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 24 Newbury Pk., Needham 92, Mass.

Freeman, Perry John ('08; '14; '19) (FOS), Head Matls. Engr., Tenn. Valley Authority, 200 Union Bldg.; *for mail,* 1415 Kenesaw Ave., Knoxville, Tenn.

Freeman, Robert Gibarde ('42) (CMN), Tool Engr., Gen. Motors Inst., 3rd Ave. & Chevrolet, Flint; *for mail,* 2065 Seymour Rd., Swartz Creek, Mich.

Freeman, Victor Robert (J'41) (ESOFKL), Mech. Engr., Air & Refrigeration Corp., 475—5th Ave., New York; *for mail,* 55 Payson Ave., New York 34, N.Y.

Freeman, Walter B. (J'40) (SFD), Mech. Engr., Duke Power Co., Riverbend Sta., Mt. Holly; *for mail,* Route 4, Charlotte 2, N.C.

Freese, Coridon Elmer (J'35) (BKJS), Draftsman, Ferro Engrg. Co., 1208 Hanna Bldg., Cleveland; *for mail,* 209 E. 14th St., Elyria, Ohio.

Frei, Jack Ernest (J'43) (TMN), Partner, Prod. Mgr., French Powder Puff Co., 90 Prince St., New York 12; *for mail,* 180 Cabrini Blvd., New York 33, N.Y.

Freiberg, James M. (J'37) (BCG), Secy., Treas., Activisible Corp., 229 E. 6th St.; *for mail,* 774 Greenwood Ave., Cincinnati, Ohio.

Freiberg, John Lawrence (J'42) (AMW), Lt., U.S.N.R., Box 202-B, Shawnee Run Rd., Cincinnati, Ohio.

Freiday, Jay Anthony ('20) (SFKODH), Mech. Engr., Ebasco Services, Inc., 2 Rector St., New York 6, N.Y.; *for mail,* 35 Cleveland Terrace, East Orange, N.J.

Fremon, Edward Bernard ('34; '45) (BMJCES), Tech. Asst. Mgr., Indus. Oil Sales Dept., Socony-Vacuum Oil Co., Inc., 4140 Lindell Blvd., St. Louis 8, Mo.

French, Dudley Kimball ('28) (EFKS), Cons. Chem. Engr., Box 185, Winnetka, Ill.

French, Edward Vinton ('05), Retired; 20 School St., Andover, Mass.
French, George Ernest ('37) (CMWSDE), Asst. Mgr., U.S. Plywood Corp., 1319 W. Nickerson, Seattle; *for mail,* 2809—39th Ave. W., Seattle 99, Wash.
French, John C. (J'38), 223 G St., Copeland Park, Newport News, Va.
French, Paul Richard (J'43) (OKCL), Metal Inspr., Humble Oil & Refining Co.; *for mail,* 128 Utah St., Baytown, Tex.
Frentzel, G. Martin (J'44) (JMNLDB), Plant Engr., Charter Oak Stove & Range Co., Antelope & Conduit Sts.; *for mail,* 4521-A Red Bud Ave., St. Louis 15, Mo.
Freres, Robert Nick (J'42) (NYB), Tool Designer, Sterling Tool Co., Merrick Ave.; *for mail,* 3817 Wright Ave., Racine, Wis.
Freston, Robert Brettingham ('46) (JS), Sr. Engr., Commonwealth Edison Co., Rm. 600, 72 W. Adams St., Chicago 90; *for mail,* 936 Washington Blvd., Oak Park, Ill.
Freund, Clement Joseph ('23; '25; '33) (CME), Dean, College of Engrg., Univ. of Detroit, McNichols Rd. & Livernois; *for mail,* 18967 Pennington Dr., Detroit 21, Mich.
Freund, Herbert E. ('37; '45) (CDFOS), Plant Engr., Talon, Inc.; *for mail,* 204 Meadow St., Meadville, Pa.
Freund, Herman R. ('28), Works Mgr., Intertype Corp., 1440 Broadway, New York; *for mail,* 241 Sterling St., Brooklyn, N.Y.
Freundlich, Jules Lewis (J'43) (MLNBG), 360 Cabrini Blvd., New York 33, N.Y.
Frewin, Leroy (J'36) (SLNDBK), Power & Maint. Engr., Am. Smelting & Refining Co., Box 1111, El Paso, Tex.
Frey, Alfred T. ('27; '33; '35) (CMDLJO), Indus. Engr., McClure, Hadden & Ortman, Inc., 111 W. Washington Blvd., Chicago 2, Ill.
Frey, Charles August ('44) (BKS), M.M., Philadelphia Elec. Co., 2301 Market St., Philadelphia, Pa.; *for mail,* 858 N. 32nd St., Camden, N.J.
Frey, George John ('22; '35) (NMLYKD), Head, Dept. of Mech. Drawing & Design, Ohio Mechanics Inst., 1104 Walnut St., Cincinnati 10; *for mail,* 19 Glenwood Ave., Cincinnati 17, Ohio.
Frey, Gordon Walter (J'45) (EFHRSZ), Bailey Meter Co. Ltd., 980 St. Antoine St., Montreal 3, Que.; *for mail,* 82 Willingdon Blvd., Toronto 9, Ont., Can.
Frey, Ralph E. (J'27) (EFS), Indus. Sales Mgr., Kan. Gas & Elec. Co., Wichita, Kan.
Freytag, John Robert (J'46), Engrg. Trainee, U.S.N.; 1412 Lehigh St., Easton, Pa.
Frick, Clifford H. ('20; '35) (CFS), Plant Betterment Engr., Pa. Power & Light Co., 901 Hamilton St., Allentown, Pa.
Frick, Malcolm Stanley (J'43) (MAS), Capt., Hdq., Army Air Forces, India Burma Theater (AMX), A.P.O. 671, c/o Postmaster, New York, N.Y.; *for mail,* Rockwell City, Iowa.
Frick, Robert Jerome (J'46) (CJL), Serv. Engr., Linde Air Products Co., 311 Ross St., Pittsburgh, Pa.
Frick, Sidney W. (J'42) (ACM), 275 N. Latch's Lane, Merion, Pa.
Fridstein, Robert Bergman (J'39) (O), Partner, Fridstein Engrg. Co., 228 N. La Salle St., Chicago 1, Ill.
Fried, Anthony ('44) (CJL), Pres., Indus. Constr. Co., Inc., 55 W. 42nd St., New York 18; *for mail,* 25 E. 86th St., New York, N.Y.
Fried, Jerome A. ('14; '25) (AZNBY), Gen. Mgr., Ithaca Scien. Instr. Co., Box 555, Ithaca, N.Y.
Fried, Robert ('41) (LMSBHK), Plant Engr., H. Kohnstamm & Co., Inc., 537 Columbia St., Brooklyn 31; *for mail,* 3 Washington Pl., Port Washington, N.Y.
Friedeberg, Solon E. ('23; '34) (SOFDT), V.P., Treas., Consultant, Franklin Engrg. Corp., 45 W. 45th St., New York 19, N.Y.
Frieder, Marcus Aurelius (J'44) (ZABSK), Pvt., Army Air Forces, Sqd. H, Scott Field, Ill.; *for mail,* 470 Ocean Pkwy., Brooklyn 18, N.Y.
Friedman, Ferdinand J. ('13; '21) (OFES), Partner, Mech. Engr., McDougall & Friedman, 1235 McGill College Ave., Montreal, Que., Can.
Friedman, John H. ('22; '35), Supt., Natl. Mchy. Co., Tiffin, Ohio.
Friedman, Martin Hugh (J'41) (AB), Struc. Designer, Edo Aircraft Corp., College Point, L.I.; *for mail,* 1016—50th St., Brooklyn 19, N.Y.
Friedman, Milton (J'33) (JZMKS), 333 West End Ave., New York 23, N.Y.
Friedrich, Herbert Louis ('44) (LPS), Engr., Westinghouse Elec. Corp., 612 Petroleum Bldg., Houston 2; *for mail,* 2430 Glenn Haven Blvd., Houston 5, Tex.
Friedrich, William George ('46), Consultant, Engrg. Div., Foreign Economic Admin., Washington; *for mail,* Cosmos Club, H St. & Madison Pl., Washington 5, D.C.
Friend, Walter Frederic (J'13) (OSLE), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 6; *for mail,* 470 W. 24th St., New York 11, N.Y.
Fries, Donald Eugene (J'46), Ensign, U.S.N.; 1400 Lake Shore Dr., Chicago 10, Ill.
Fries, Geo. S. ('27; '35), Engr., Gilbert Associates, Inc., 412 Washington St., Reading; *for mail,* Jacksonwald, Pa.
Frigiola, Nicholas F. ('25; '37) (ABJ), Teacher, Bayonne Tech. High Sch., Bayonne; *for mail,* 101—64th St., West New York; N.J.
Frigon, Raymond Augustin (J'43) (BNJMCY), Regional Rep., Montreal Area, Research & Devel. Branch, Dept. of Reconstruction, Dominion Govt., 224 Youville Sq., Montreal; *for mail,* 125 Pagnuelo St., Outremont, Montreal 8, Que., Can.
Frisch, George Mason (J'41) (CMZJB), Tech. Asst., Methanol Dept., Carbide & Carbon Chemicals Corp., 437 MacCorkle Ave., South Charleston 3; *for mail,* Apt. 15, 1331 Virginia St., Charleston 1, W. Va.
Frisch, Martin ('22; '29; '35) (CBKSLF), Dir., Engrg. & Devel., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Fritsch, Robert Albert (J'43), Army Air Forces, 437th Fighter Sqd., A.P.O. 719-2, c/o Postmaster, San Francisco, Calif.; *for mail,* 2654 N. Palmer St., Milwaukee 12, Wis.
Fritsche, Carl B. ('42) (YALCW), V.P., Reichhold Chemicals, Inc., Box 927, Tuscaloosa, Ala.
Fritton, William John, II (J'44) (CLMJD), Works Engr., Van Der Horst Corp. of Am., 314 Penn Ave.; *for mail,* 312 Laurel Ave., Olean, N.Y.
Fritts, Stewart S. (J'33) (DFL), Lone Star Cement Corp., 342 Madison Ave., New York, N.Y.; *for mail,* 49 Filmore St., Phillipsburg, N.J.
Fritz, Charles ('42), 22—5th St., Fords, N.J.
Fritzinger, Walter Llewellyn (J'43) (ACD), Ensign, 26 Stanley St., Wilkes-Barre, Pa.
Froberg, Harold G. ('29; '39) (SFDKNO), Boiler Rm. Engr., Commonwealth Edison Co., 3501 S. Pulaski Rd., Chicago 32; *for mail,* 9631 S. Hamilton Ave., Chicago 43, Ill.
Frobes, C. David ('44) (CDFHLM), Prod. Mgr., Edgar Bros. Co., McIntyre, Ga.
Frocht, Max Mark ('38) (AB), Assoc. Prof. Mechanics, Carnegie Inst. of Tech., Schenley Park, Pittsburgh, Pa.
Froehlich, Kurt F. ('43) (EB), Designer, Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee; *for mail,* 182 W. Saveland Ave., Milwaukee 7, Wis.
Froehlig, George Edward (J'42) (CBMH), Ensign, U.S.N.R., Bachelor Officers' Quarters, Bldg. 2, Naval Shipyard, Washington 25, D.C.; *for mail,* 187 Summit Ave., Bogota, N.J.
Frohboese, Robert Harry (J'39) (CMZ), Foreman, Westinghouse Elec. Corp., 95 Orange St., Newark; *for mail,* 48 Hawthorne Rd., Short Hills, N.J.
Frohlin, Charles R. (J'38) (ES), Lt. (j.g.), U.S.N.R., Apt. 6-M, 37-05—79th St., Jackson Heights, N.Y.
Frohmuth, Robert Lee (J'36) (CBTYZH), Physicist, U.S.N., Bldg. 681, Naval Base, Philadelphia; *for mail,* 922 Argyle Rd., Drexel Hill, Pa.
Frohrieb, Louis C. ('13) (EKSFHY), Engr., Fed. Engrg. Co., 239—4th Ave., Pittsburgh 22; *for mail,* 1107 Peermont Ave., Pittsburgh 16, Pa.
Frolander, Frank Conrad ('27) (NBGZ), Ch. Designer, Research & Devel., Mergenthaler Linotype Co., Brooklyn, N.Y.; *for mail,* 156 Orchard St., Elizabeth 3, N.J.
Fromm, Charles William (J'42) (HNAZJB), 1461 Jefferson Ave., Brooklyn 27, N.Y.
Frost, Daniel Carter ('43) (CHM), R.D. 6, Lancaster, Pa.
Frost, Edward (J'42) (ABK), Test Engr., Wright Aero. Corp., Paterson; *for mail,* 237 Goffle Rd., Ridgewood, N.J.
Frost, George Henry (J'40) (ACMO), Mech. Engr., Associated Engrs., Inc., 230 E. Berry St., Ft. Wayne 2, Ind.
Frost, Raymond Benson ('45) (DKJY), 21 Wesley Ave., Bernardsville, N.J.
Frost, Robert Carlson (J'43) (NMDBCO), Design Layout, Perfex Corp., 500 W. Oklahoma St., Milwaukee; *for mail,* 4517 N. Larkin St., Milwaukee 11, Wis.
Frost, Vincent Morse ('14) (SFEBKJ), Asst. Engr., Elec. Engrg. Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 2; *for mail,* 51 Wilcox Ave., East Orange, N.J.
Frost, William G. (J'43) (BAM), Engr., Gen. Am. Trans. Corp., 209 W. Jackson Blvd., Chicago 6; *for mail,* 901 S. Grove Ave., Oak Park, Ill.
Frush, Donald W. (J'39), Prod. Mgr., Coca Cola Bottling Works, 933 Madison St., Gary, Ind.
Frusher, William Albert (J'43) (JAM), Ensign, U.S.N.R., c/o O. V. Russell, Fredonia, Kan.
Fry, Albert Henry ('31; '35) (SFK), Comdr., U.S.N.R., Naval Hospital, Philadelphia; *for mail,* 6142B N. 8th St., Philadelphia 20, Pa.
Fry, Alfred George (J'45) (BHM), Engr., Pa. Crusher Co., Liberty Trust Bldg., Broad & Arch Sts., Philadelphia 7; *for mail,* 6521 Edmund St., Philadelphia 35, Pa.
Fry, Horace Pugh ('20) (GB), Retired; 4806 Regent St., Philadelphia 43, Pa.
Fry, Lawford H. ('05; F'42) (RSKFE), *Worcester Reed Warner Medallist,* '38; Dir. of Research, Steam Loco. Research Inst., Inc., 60 E. 42nd St., New York 17, N.Y.
Fry, Winthrop Hastings ('45), V.P., Ch. Engr., E. W. Carpenter Mfg. Co., 1565 Railroad Ave., Bridgeport 5; *for mail,* 23 Harlem Ave., Bridgeport 2, Conn.
Frye, Calvin Barton (J'39) (BEFJKN), Assoc. Engr., Bur. of Ships, Washington 25; *for mail,* 1900 F St., N.W., Washington 6, D.C.
Frye, Charles Franklin ('37) (BMZ), Owner, Economy Valve Seat Co., 2807 Fletcher St., Chicago 18, Ill.
Frye, John Henry (J'40) (BANJW), Sr. Stress Analyst, Goodyear Aircraft Corp., Akron; *for mail,* 56 Marvin Ave., Akron 2, Ohio.
Fryer, Ross Lauder, Jr. (J'35), Asst. Eng. Design Engr., White Motor Co., 842 E. 79th St.; *for mail,* 3170 W. 159th St., Cleveland, Ohio.
Fryling, Glenn Rhodes (J'44) (OCRZBH), Cons. Engr., Susquehanna Engrg. Associates, 324 Market St., Lewisburg; *for mail,* 411 Market St., Sunbury, Pa.
Frysinger, Victor Gould (J'41) (AEFJ), Lt., Army Air Forces, Aircraft Maint. Engrg. Officer, Diamond L Motel, Box 864, Clovis, New Mex.; *for mail,* 341 Stein Court, New Braunfels, Tex.
Fuchs, Edward Alfred (J'40) (CRONLB), Supvg. Engr., Installations, Gen. Elec. Co., 1405 Locust St., Philadelphia 2, Pa.; *for mail,* Box 433, Schenectady, N.Y.
Fuchs, Henry Otten ('44) (BNR), Asst. Ch. Designer, Preco, Inc., 960 E. 61st St., Los Angeles 1, Calif.
Fuhro, Alfred Antoine ('45), Plant Mgr., Leslie Co., Lyndhurst; *for mail,* 223 North St., Jersey City, N.J.
Fujimoto, Kenji (J'43), Columbia Hotel, 3120 Broadway, New York, N.Y.
Fullam, Harland Oscar (J'40) (ACHJMN), Asst. Ch. Engr., L.C.N. Corp., 1103 Enoch Dr., Middletown; *for mail,* 6705 Hampton Dr., Silverton, Ohio.
Fuller, Andrew Lansing (J'46) (HMS), 1745 Chicago Blvd., Detroit 6, Mich.
Fuller, Charles Edward ('12) (BHR), Prof. Emeritus, Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39; *for mail,* 21 Great Plain Ave., Wellesley 81, Mass.
Fuller, Dudley Dean (J'43) (BNRAE), Assoc., Mech. Engrg. Dept., Columbia Univ.; *for mail,* 21 Claremont Ave., New York 27, N.Y.
Fuller, Earl Howard ('30), Maint. Engr., Kelsey-Hayes Wheel Corp., 3600 Military Ave., Detroit; *for mail,* 24 Maywood St., Pleasant Ridge, Mich.
Fuller, Floid Merrill ('07; '17; '19) (S), Distributing Engr., Pa. Power & Light Co., 901 Hamilton St., Allentown; *for mail,* 1627 W. Market St., Bethlehem, Pa.
Fuller, George F. ('20), Mch. Engr., Chmn. of Bd., Wyman-Gordon Co., 105 Madison St.; *for mail,* 14 Massachusetts Ave., Worcester 5, Mass.
Fuller, Robert B. ('31; '35) (TSKLOD), Supt., Sayles Biltmore Bleacheries, Inc., Biltmore; *for mail,* 12 Ridge Rd., Biltmore Sta., Asheville, N.C.
Fuller, Roy L. (J'39), 635 Mandalay Dr., San Antonio, Tex.
Fuller, Samuel Harry (J'40) (BZAO), Capt., Ord. Dept., U.S.A., Chief, Armament Sec., Devel. Div., Office of Chief of Ord., Center Line, Mich.; *for mail,* 1111 Richmond St., Pittsburgh 18, Pa.
Fuller, Walter D. (A'13) (CG), Pres., Curtis Publ. Co., Independence Sq., Philadelphia, Pa.
Fullerton, Herbert Palmer ('33; '45) (BNMKS), Assoc. Prof. Engrg. Drawing & Mch. Design, Dept. of Engrg., Univ. of Va., Thornton Hall, Charlottesville, Va.
Fulling, Roger William ('42) (CMDSON), Spec. Asst. to Gen. Dir., Indus. & Devel. Engrg. Divs., Engrg. Dept., E. I. du Pont de Nemours & Co., Wilmington 99, Del.
Fullman, C. Arthur ('34; '35), Apt. 701, 1167 Bush St., San Francisco 9, Calif.
Fullmer, Forrest Lee (J'43) (AJBN), 1669 Englewood Ave., Akron 5, Ohio.
Fullmer, Irvin Henry ('19; '25; '34) (JZBN), Sr. Physicist, Natl. Bur. of Stands., Washington 25, D.C.; *for mail,* 9411 Louis Ave., Silver Spring, Md.
Fulton, Clare Kendall (J'43), 57 Academy St., Bath, Me.
Fulton, Garland ('42) (MCKBNJ), V.P., Cramp Shipbldg. Co., Philadelphia 25, Pa.
Fulton, George Richard ('41) (CSEF), V.P., Div. Mgr., Gulf States Utilities Co., 314 Broad St., Lake Charles, La.
Fulton, Henry Roedel ('40; '45), 3065 Dwight Ave., Dormont, Pittsburgh, Pa.
Fulton, Marvin Oliver (J'44) (CMLWZ), Design Engr., Stand. Oil Co. (Ind.), 119th & Front Sts., Whiting, Ind.; *for mail,* Box 14, Oak Forest, Ill.
Fulton, Raymond Allinson ('44), Asst. Efficiency Engr., Pacific Gas & Elec. Co., 430—23rd St., San Francisco 7, Calif.
Fulweiler, John Edwin ('08; '13) (OLNJSH), Ch. Mech. Engr., Fraser-Brace Engrg. Co., 10 E. 40th St., New York; *for mail,* 54 Sagamore Rd., Bronxville, N.Y.
Fulweiler, W. H. ('42) (EFKWYS), Chem. Engr., 1600 Arch St., Philadelphia 3, Pa.
Funk, Herbert Becker (J'42) (AJS), 545 Nacogdoches St., Jacksonville, Tex.
Funk, Hubert Joseph (J'43) (JSR), Designer, Linde Air Products Co., 646 Frelinghuysen Ave., Newark 5; *for mail,* 488 Jersey Ave., Fairview 1, (Cliffside Park P.O.), N.J.
Funk, Nevin Elwell ('18; F'39) (SKHFDC), V.P., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Funk, William Francis ('92; '05) (JMS), Pres., Mgr., La Crosse Trailer & Equip. Co., 418 Gould St., La Crosse, Wis.

Funke, Richard Harrison, Jr. (J'43) (SR), Asst. to Ch. Engr., Doughnut Corp. of Am., Ellicott City; *for mail*, 9 Mallow Hill Rd., Baltimore 29, Md.
Furchgott, Arthur C., Jr. (J'33) (ACR), Apt. 5, 2225 S.W. 9th St., Miami, Fla.
Furcron, Wycliffe Sharpe (J'43), Ensign, U.S.N.R., LSM-506 Detail, Naval Receiving Sta., Houston, Tex.
Furedy, Alfred Stephen (J'42) (ACM), 641 Willis Ave., Williston Park, N.Y.
Furman, George Roland (J'37) (ZGBKAF), Mech. Engr., Research, Texas Co., Glenham; *for mail*, 67 Cross St., Beacon, N.Y.
Furnas, Clifford Cook ('43), Research Lab. Dir., Accounts Payable Dept., Curtiss-Wright Corp., Buffalo, N.Y.
Furutani, George John (J'43) (NJYMEA), Mech. Engr., Mch. Design, Shulton, Inc., 1500 Hudson, Hoboken, N.J.; *for mail*, Apt. 5, 304 W. 109th St., New York 25, N.Y.
Fusfeld, Robert David (J'42) (ABM), 2026 R St., N.W., Washington, D.C.
Fusner, George Roselle (J'42) (SEARFK), Turbine Generator Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 211 Jackson Ave., Schenectady 4, N.Y.
Futral, John Watson, Jr. (J'43) (HEFWA), Mech. Engr., Southeast. Pipe Line Co., 718 Forsyth Bldg., Atlanta 3, Ga.
Fykse, Lewis Dean (J'41), Prod. Control Engr., Stand. Tool Co., 6918 Central Ave., Cleveland 3; *for mail*, Bishop Rd., Wickliffe, Ohio.
Fyles, Cleveland Saunders (J'45) (WYNMCB), Plant Engr., Fyles & Rice Co., Inc.; *for mail*, Bethel, Vt.

G

Gabla, Naval Fakirji (J'36) (ESFCJM), Asst. Engr., Powerhouse, Karachi Elec. Supply Corp., Karachi, India.
Gabor, Harry W. ('32) (CDMN), Prin. Safety Supvr., N.Y. State Ins. Fund, 625 Madison Ave., New York 22; *for mail*, 17 Lawrence Ave., Tuckahoe 7, N.Y.
Gabriel, Adam F. (J'42) (JM), Prod. Engr., Acme Indus. Co., 200 N. Laflin St.; *for mail*, 3336 N. New England Ave., Chicago, Ill.
Gabriel, Edwin Z. (J'36) (CLS), Pvt., 1st Class, 918th Engr. Air Force, Hdq. Co., Army Air Force Tactical Air Comd., Orlando, Fla.
Gabriel, Wm. A. ('91), Retired; 570 E. Chicago St., Elgin, Ill.
Gabrielson, Gunnar ('39) (WACMDY), V.P., Gen. Mgr., Internatl. Aviation Corp., 1900 E. 24th St., Cleveland; *for mail*, 1012 Yellowstone Rd., Cleveland 21, Ohio.
Gadd, Charles W. (J'41) (BJZMN), Research Engr., Research Div., Gen. Motors Corp., 485 W. Milwaukee St.; *for mail*, 3339 Pingree St., Detroit 6, Mich.
Gaderlund, Harry A. (J'41) (ACM), 4911 N. Hoyne Ave., Chicago 25, Ill.
Gadzuk, John ('45) (KLO), Engr., Orr & Sembower, Inc.; *for mail*, 1522 Centre Ave., Reading, Pa.
Gaebler, George F. (J'40) (ACBHY), 32 Left Wing Dr., Middle River 20, Md.
Gaehr, David ('02; '09) (DFS), Instr., Charge Lake Carriers' Assn. Engrs.' Sch., 414 Superior Ave., N.W.; Mech. Engr. (Gen. Engrg. Practice), Rockefeller Bldg., Cleveland, Ohio.
Gaffert, Gustaf Adolph ('23; '30; '35) (SFOEK), Partner, Mech. Engr., Sargent & Lundy, 140 S. Dearborn St., Chicago 3; *for mail*, 1224 Cherry St., Winnetka, Ill.
Gage, Robert Thompson (J'45), Staff Mem., Radiation Lab., Mass. Inst. of Tech., Cambridge; *for mail*, 240 Lincoln St., Lexington, Mass.
Gage, Victor R. ('36) (ABEFHS), Prof. Mech. Engrg., College of Engrg., Cornell Univ.; *for mail*, 527 Highland Rd., Ithaca, N.Y.
Gagg, R. F. ('23; '34) (ACE), *Vice-President*, '43-'46; Asst. to Gen. Mgr., Wright Aero. Corp., Paterson 3, N.J.
Gagliardi, Frank Anthony (J'44), 6342 Highland Ave., Merchantville, N.J.
Gagne, Antoine Francis, Jr. (J'41) (NJHMDC), Gp. Head, Mch. Devel. Sec., Engrg. Dept., E. I. du Pont de Nemours & Co., Pompton Lakes, N.J.
Gagnebin, Albert Paul ('42) (JARSDL), Metallurgist, Internatl. Nickel Co., Oak St., Bayonne, N.J.
Gagnon, Alfred Joseph (J'45) (COMNDJ), Staff Engr., Cons. Mgmt. Engr., Booz, Allen & Hamilton, 285 Madison Ave., New York 17; *for mail*, 31 Cameron Ave., Hempstead, L.I., N.Y.
Gahagan, William Corthel ('45), V.P., Gahagan Constr. Corp., 90 Broad St., New York 4; *for mail*, 114 E. 52nd St., New York, N.Y.
Gahan, John J. (J'37) (BHJNSC), Research Engr., Taylor Forge & Pipe Works, 14th St. & Cicero Ave., Cicero; *for mail*, 1529 E. 85th St., Chicago 19, Ill.
Gahnkin, Valentine G. ('22; '28), Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail*, R.D. 1, Wappingers Falls, N.Y.
Gail, Stewart E. (J'41) (NMJACY), Engr., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, 38 Milton St., East Lynn, Mass.
Gaillard, John ('28) (COM), Mech. Engr., American Standards Association, 70 E. 45th St., New York 17; *for mail*, 400 W. 118th St., New York 27, N.Y.
Gailus, Walter Joseph (J'43) (BYZC), Research Engr., Mass. Inst. of Tech.; *for mail*, Rm. 7-007, Mass. Inst. of Tech., Cambridge 39, Mass.
Gaines, William Pendleton (J'45) (BJNO), 121 Hawthorne Rd., Homewood 9, Ala.
Gaisman, Henry J. ('38), Chmn., Bd. of Dirs., Gillette Safety Razor Co., Boston, Mass.; *for mail*, Hartsdale, N.Y.
Gaither, Robert H. ('18) (HKMCEF), Mech. Engr., Navy Matl. Catalog Office, 111 E. 16th St., New York; *for mail*, 61 Kilburn Rd., Garden City, L.I., N.Y.
Gajarsky, John Edward (J'38) (TSDNLC), Project Engr., Forstmann Woolen Co., 2 Barbour Ave., Passaic; *for mail*, 325 Palisade Ave., Garfield, N.J.
Galazzi, Joseph Angelo (J'43) (BEHKNS), Asst. Dir., Engrg. Lab., York Corp., Roosevelt Ave., York, Pa.
Galbraith, Frank William (J'44) (MCJDN), Lt. (j.g.), U.S.N.R., Asst. Ord. Planning Officer, Puget Sound Naval Shipyard, Bremerton, Wash.; *for mail*, 1394 W. Iowa Ave., St. Paul 4, Minn.
Galbreath, Paul J. (J'36) (NHTYGA), Mech. Engr., Tenn. Eastman Corp.; *for mail*, 1424 Fairidge Dr., Kingsport, Tenn.
Galdieri, Biag (J'44) (ZMNJAB), Prod. Engr., C.J. Tagliabue Div., Portable Products Corp., 540 Park Ave., Brooklyn 5; *for mail*, 345—86th St., Brooklyn 9, N.Y.
Gale, Albert Garland ('45) (CJNSME), Gen. Mgr., New England Trawler Equip. Co., 301 Eastern Ave., Chelsea 50; *for mail*, Depot Rd., Boxford, Mass.
Gale, Frank Carl ('42) (BEFHLS), Salesman, Brooks Equip. Co., 636 Portrero St.; *for mail*, Apt. 1, 540 Stockton St., San Francisco, Calif.
Gale, Horace Bigelow ('86) (TWA), Retired; 10 Highland St., Natick, Mass.
Gale, Philroy Clifton ('24) (HO), Asst. Admin., Chief of Operas., Southwest Power Admin., U.S. Dept. of Interior, P.O. Drawer 1619, Tulsa 1, Okla.
Gale, Warren D. ('14; A'26), Waterville, Que., Can.
Gale, William Keen ('44) (LOSCKN), Project Engr., Otto Constr. Co., 225 W. 34th St., New York 1; *for mail*, 86-10—34th Ave., Jackson Heights, N.Y.
Gall, Alexander Hutchison ('41) (MJC), Mfg. Engr., Dictagraph Sales Corp., 95-25—149th St., Jamaica, L.I.; *for mail*, 5 N. Tyson Ave., Floral Park, L.I., N.Y.
Gall, George Francis, Jr. (J'42) (CHKMSW), Asst. Hull Ship Supt., N.Y. Naval Shipyard, Brooklyn 1; *for mail*, 333 E. 43rd St., New York 17, N.Y.
Gall, William Rolfe (J'42) (SZEKMA), Engr. in Charge of Design, Clinton Engr. Works, Tenn. Eastman Corp., Box 590, Knoxville; *for mail*, 315 E. Forest Rd., Oak Ridge, Tenn.
Gallagher, Edmund Gordon (J'34) (DYEFKA), Sales Engr., Dunlop Tire & Rubber Goods Co. Ltd.; *for mail*, 87 Tremoy Rd., Noranda, Que., Can.
Gallagher, Edw. I. ('40) (FKS), Plant Supt., Richmond Sta., Philadelphia Elec. Co., Delaware Ave. & Lewis St.; *for mail*, 1010 W. Wagner Ave., Philadelphia, Pa.
Gallagher, Eugene Leo (J'43) (LKD), Ensign, E-V (S), U.S.N.; 6416 Regent St., Oakland 9, Calif.
Gallagher, Harry Gorsuch, Jr. (J'40) (ACH), Jr. Engr., Design & Devel., Plant Equip., Bigelow-Sanford Carpet Co., Inc., Main St., Thompsonville; *for mail*, River Blvd., Suffield, Conn.
Gallagher, John J. ('29), Owner, North Mt. Brick Co., 21 E. 40th St., New York, N.Y.
Gallagher, John S. ('21; '28) (FK), 1037 Burnside Ave., Los Angeles, Calif.
Gallagher, Paul (J'44) (ANKH), Sr. Engr., Tenn. Eastman Corp.; *for mail*, 1621 Linville St., Kingsport, Tenn.
Gallaher, Edward Beach ('19) (A), Pres., Clover Mfg. Co., 327 Main St., Norwalk, Conn.
Gallalee, John Morin ('20) (SCFEMN), Dir., Tech. Staff, Ala. Bldg. Comm., State Capitol, Montgomery; *for mail*, University, Ala.
Gallatin, Robert Ellis (J'44) (HSKEA), Devel. Engr., Bowser, Inc., 1302 E. Creighton St., Ft. Wayne 2; *for mail*, 1814½ Bayer Ave., Ft. Wayne 3, Ind.
Gallaway, Howard Murray (J'35) (SREFAH), Designer, Joshua Hendy Iron Works, Sunnyvale; *for mail*, 106 Beverly Dr., San Carlos, Calif.
Gallenkamp, Edward William ('21) (OMLKSC), Dir., Student Activities, York Corp., Roosevelt Ave.; *for mail*, 425 Rathton Rd., York, Pa.
Gallese, Julio J. (J'43) (AJKLMW), Casilla 317, Lima, Peru, S.A.
Galligan, Edward Joseph (J'43) (ACMH), Exper. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail*, 38 Bliss St., Hartford 6, Conn.
Galligan, John Edmund (J'41) (AKM), 3640 N. Paulina St., Chicago 13, Ill.
Gallini, Emil ('37) (CDM), Supvr. of Engrg., Chrysler Corp., 341 Massachusetts Ave., Detroit; *for mail*, 18515 Prairie Ave., Detroit 21, Mich.
Gallivan, John Daniel, III (J'43) (EKS), 1st Lt., Corps of Engrs., Box 64, Balboa, C.Z.
Gallmeyer, William Christian (J'38) (JBC), Asst. Ch. Engr., Gallmeyer & Livingston Co., 336 Straight Ave., Grand Rapids 4; *for mail*, 2731 Elmwood Dr., S.E., Grand Rapids 6, Mich.
Gallow, John Joseph (J'45), 40-08—171st St., Flushing, L.I., N.Y.
Galloway, Alexander Kennedy ('42; '44) (ACR), Gen. Supt., M.P. & Equip., Baltimore & Ohio R.R., B. & O. Cent. Bldg., Baltimore 1, Md.
Galloway, Chas. D. ('13) (BDL), Factory Engr., Elec. Storage Battery Co., 1900 Allegheny Ave., Philadelphia, Pa.
Galloway, Frank M. ('21; '27), M.P. Rep., Frigorifico Armour de la Plata, Reconquista 314, Buenos Aires, Argentina, S.A.
Galloway, Jas. W. ('19; '26; '35) (DJM), Plant Engr., B. Greening Wire Co., Ltd., Hamilton, Ont., Can.
Galloway, Lee (A'23) (C), R.F.D. 1, Lake Wales, Fla.
Galloway, Robert M. (J'37) (JL), Lt., U.S.N.R.; *for mail*, 4319 Shirley St., Omaha, Neb.
Gallup, Rockwell Loring ('18) (GNHBS), Mch. Designer, E. H. Wachs Co., 1525 Dayton St., Chicago 22; *for mail*, 2939 Leland Ave., Chicago 25, Ill.
Galopin, Frank Ernest (J'42) (ACL), Elec. Serv. Engr., Westinghouse Elec. Corp., 10 High St., Boston; *for mail*, 165 Burley St., Danvers, Mass.
Galt, John Gillespie (J'42) (CLJM), Plant Mgr., Keystone Trappe Rock Co., Glen Moore, Chester Co.; *for mail*, R.F.D. 1, West Chester, Pa.
Galusha, Albert Leet ('18) (FE), Ch. Engr., Wellman-Galusha Gas Equip., Wellman Engrg. Co., 420 Lexington Ave., New York 17, N.Y.; *for mail*, 19 Manor Rd., Verona, N.J.
Galuzevski, Rostislav Alexander (J'44) (JMBN), Asst. Research Engr., Univ. of Mich.; *for mail*, 535 Walnut St., Ann Arbor, Mich.
Galvanek, Edward Joseph (J'38) (KSBJ), Serv. Engr., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.; *for mail*, 67 West Ave., Port Reading, N.J.
Galvin, John E. ('21) (CJRMEL), Pres., Ohio Steel Fdy. Co., W. 4th St., Lima, Ohio.
Galvin, Ralph B. (J'40) (KHMCSF), Lt., U.S.N.R., Bur. of Ships, Navy Dept., Washington; *for mail*, Apt. 226, 3100 Connecticut Ave., Washington 8, D.C.
Galvis, Ricardo (J'45) (EOS), Calle 11 Sur, 9-46, Barrio San Fernando, Cali, Colombia, S.A.
Gamache, Jean Paul (J'42) (GYBC), Adv., Engrg. Specifications, Plessisville Fdy.; *for mail*, 147 St. Calixte, Plessisville, Que., Can.
Gamarekian, S. Edward (J'45) (ABK), University Club, 17 Front St., Schenectady, N.Y.
Gambert, George (J'33) (HKS), Power Methods & Stands. Engr., E. I. du Pont de Nemours & Co., Drawer B, Sta. B; *for mail*, 821 Elm St., Buffalo, N.Y.
Gamberton, James H. (J'38) (CM), Mech. Engr., Air Reduction Sales Co., 125 Monitor St., Jersey City; *for mail*, 1822 Lamberts Mill Rd., Westfield, N.J.
Gamble, Walter W. ('19), Mgr., Mech. & Power Opera., 3 N.J. Plants, Stand. Oil Co. of N.J., Bayonne; *for mail*, 42 Aberdeen Rd., Elizabeth, N.J.
Gamboa, Francisco Rodolfo ('38) (CEHRS), Tech. Rep., Baldwin Loco. Works, Casilla 86-D, Crescente Errázuriz 2195, Santiago de Chile, S.A.
Gammell, John (A'45) (SEHFAJ), Field Engr., Allis-Chalmers Mfg. Co., 887 Broad St. Sta. Bldg., Philadelphia 3, Pa.
Gamon, Thomas Hampton (J'42), Lt. (j.g.), U.S.N.R., LCJ (L) (3) 2, c/o Naval Sta., Astoria, Ore.
Gandelot, Howard Kenneth ('43) (BENMJY), Engr., Gen. Motors Corp., Gen. Motors Bldg., Detroit 2; *for mail*, 832 Hampton Rd., Grosse Pointe Woods 30, Mich.
Gang, Oliver F. (J'20), V.P., Wm. Powell Co., 50 Church St., New York 7; *for mail*, 7 Harcourt Rd., Scarsdale, N.Y.
Gangewere, Ernest Paul ('22; '28; '35) (EJMR SW), Supt., M.P. & Rolling Equip., 6th & Perry Sts.; *for mail*, 1229 N. 13th St., Reading, Pa.
Gangwere, Elliott D. ('41) (DN), Dir. of Equip., Westinghouse Elec. Corp., Pittsburgh; *for mail*, 136 Elmore Rd., Pittsburgh 21, Pa.
Gannett, Malcolm Frederick (A'42) (BGDHLN), Pat. Atty., 2325 Park Ave., Santa Clara, Calif.
Gannon, Jesse ('20; '35) (ACJMN), Mfg. Engr., Harvey Mch. Co., Inc., 6200 Avalon Blvd., Los Angeles 3; *for mail*, 4301 Jackson Ave., Culver City, Calif.
Gano, Howell McLain ('25) (CFKMOY), Cons. Engr., 115 Wisteria Dr., Dayton 9, Ohio.
Gans, James Wilson (J'44), Technician, 4th Grade, 400th Engrs. Maint. Co., A.P.O. 547, c/o Postmaster, San Francisco, Calif.; *for mail*, 228 W. Glentay Rd., Lansdowne, Pa.
Ganz, John Maria (J'43) (H), Engr., Charge Exper. Lab., Carver Pump Co.; *for mail*, 714 Iowa Ave., Muscatine, Iowa.
Garand, John C. (Non-Member), *Holley Medallist*, '41; Ch. Engr., Ord., Springfield Armory, U.S. War Dept., Springfield, Mass.
Garavaglia, Anthony Fortunato (J'44) (NBGZ LJ), Lab. Operational Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail*, 3515 Chesapeake Ave., Hampton, Va.
Garber, Milton Michael (J'44) (ABC), 105 E. Fairview Blvd., Inglewood, Calif.
Garbin, Albert John (J'43) (YTBCKJ), Tech. Mgr., V-Belt Dept., B. F. Goodrich Co., 500 S. Main St., Akron; *for mail*, 418 Portage Trail East, Cuyahoga Falls, Ohio.

Garcia, Bert H., Jr. (J'41) (CKDMAO), Instr. Mech. Engrg., Pa. State College; *for mail,* 425 S. Allen St., State College, Pa.
Garcia de la Torre, Luis ('45) (CDEHKS), Sales & Installation, Heavy Mchy., Miguel Morales Inc., San Juan; *for mail,* P.O. Box 4956, San Juan 24, P.R.
Gard, Earl William ('39) (LEFYCA), 2780 Via Campesina, Palos Verdes Estates, Calif.
Garden, Joseph MacKenzie (J'41) (PLK), 50 James Ave., Cranford, N.J.
Garden, Nelson B. ('26; '33), Box 399, Berkeley, Calif.
Gardiner, Cecil Merritt ('32; '42) (SKBZ), Turbine Engrg. Div., Gen. Elec. Co., Schenectady; *for mail,* 27 Catherine St., Schenectady 7, N.Y.
Gardner, Elston A. (J'37) (ABJNHC), Test Engr., Northrop Aircraft, Inc., Hawthorne; *for mail,* 1427 Greenfield Ave., Los Angeles 25, Calif.
Gardner, Eugene Knowles (J'41) (BCM), Indus. Engr., Sylvania Elec. Products, Inc.; *for mail,* 52 County Rd., Ipswich, Mass.
Gardner, John Allaire (J'45) (MBJEFS), 19 Rose Ave., Floral Park, L.I., N.Y.
Gardner, Karl Albert ('45) (KBH), Engr., Griscom-Russell Co., 285 Madison Ave., New York 17; *for mail,* 15 Elliott Pl., Freeport, L.I., N.Y.
Gardner, Kirtland Cutter ('41) (BCHJMO), Pres., Gen. Mgr., United Engrg. & Fdy. Co., 2500 1st Natl. Bank Bldg., Pittsburgh 22, Pa.
Gardner, Lester Durand (A'22) (AECFY), Chmn. of Council, Inst. of Aero. Sciences, 2 E. 64th St., New York 21, N.Y.
Gardner, Lester Hoke ('41) (OKSF), Owner, Firm of Lester H. Gardner, Cons. Engr., 318-21 O'Sullivan Bldg., Baltimore 2, Md.
Gardner, Lewis Robinson (J'29), Mar. Engr., Norfolk Naval Shipyard, Portsmouth; *for mail,* 2821 Marlboro Ave., Norfolk 4, Va.
Gardner, Marvin K. (J'39) (KC), Branch Mgr., ILG Elec. Vent. Co., 1215 Tex. Bank Bldg., Dallas 1, Tex.
Gardner, Robert Irving (J'36) (ONM), Cons. Engr., Firm of Robt. I. Gardner, 533 N. Cherokee Ave., Los Angeles 4, Calif.
Gardner, Thos. H. ('39) (BGR), c/o Ch. Engr., Cent. of Ga. Ry., Savannah, Ga.
Gardner, Wallace Webster (J'36) (NMBHJY), Design Engr., Perfex Metal Products Co., 25 Lafayette St., Brooklyn, N.Y.; *for mail,* 27 Oak Knoll Rd., Glen Rock, N.J.
Gardner, William Norment (J'43) (ABC), Aero. Engr., Langley Memorial Aero. Lab., Natl. Advis. Com. for Aero., Langley Field; *for mail,* Apt. 3, 204 Regent St., Hampton, Va.
Garey, Geo. W. ('22; '35), Ch. Engr., Compania Salitrera Anglo-Chilena, Casilla 808, Antofagasta, Chile, S.A.
Garey, Lloyd Leslie ('25; '38) (EFKS), Asst. Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Garey, Robert Bryant (J'41) (EBJ), Draftsman, Joshua Hendy Iron Works, Sunnyvale; *for mail,* 151 N. 10th St., San Jose 12, Calif.
Garfield, A. S. ('07), 45 Boulevard Beausejour, Paris, France.
Garger, John Henry (J'36) (JLYCW), Asst. Sales Mgr., Induction Htg. Corp., 389 Lafayette St., New York 3; *for mail,* 2923 Otis Ave., New York 61, N.Y.
Gariffo, Carl Joseph (A'45) (BJMNWY), Owner, Firm of C. J. Gariffo, 1607—1st Ave., New York 28, N.Y.
Garity, William Edward (J'45) (SHA), Test Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia 42; *for mail,* 5 E. Plumstead Ave., Lansdowne, Pa.
Garland, Claude M. ('06; '12) (EFS), Pres., C. M. Garland & Co., 38 S. Dearborn St., Chicago, Ill.
Garland, Clyne Frederick ('38) (BLMNZ), Assoc. Prof. Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.
Garland, James ('21), Gen. Supt., Dominion Engrg. Works Ltd., Box 220, Montreal, Que., Can.
Garman, Warren DeWitt ('37) (BNAKES), Assoc. Prof. Mech. Engrg., Bucknell Univ.; *for mail,* College Park, Lewisburg, Pa.
Garnar, L. Howard ('37; '42) (HLEYWS), Application Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 10 Birdseye Glen, Vernona, N.J.
Garneau, Leo A. (J'46) (AHJ), 156 Lake St., Brighton 35, Mass.
Garnier, Alphonse ('29), Mem. Bd. of Dirs., Cie. de Fives-Lille, 7 rue Montalivet, Paris; *for mail,* Villa Dianina, Villefranche-sur-Mer, Alpes-Maritimes, France.
Garnsey, Hamilton, Jr. ('29; '35) (CDH), V.P., Goulds Pumps Inc., Seneca Falls, N.Y.
Garrabrant, F. LeRoy, Jr. (J'43) (SCZ), De Laval Steam Turbine Co., Trenton 2; *for mail,* 834 Berkeley Ave., Trenton 8, N.J.
Garrahan, Thomas F. (J'46) (CDM), 4819 Western Ave., N.W., Washington 16, D.C.
Garrard, Oscar Julian, Jr. (J'41), 5768 Pine Tree Dr., Miami Beach, Fla.
Garratt, Ernest Albert ('07) (ORE), Dir., Jacobs, Barringer & Garratt, Ltd., 106 Fenchurch St., London; *for mail,* Crow's Nest, Pear Tree Lane, Shorne, Kent, England.
Garretson, Henry Chapin, Jr. (J'40) (CMJ), Lt. Col., Field Arty., Hdq., U.S. Forces, China Theater, A.P.O. 879, c/o Postmaster, New York, N.Y.
Garrett, Edward J. (J'45) (MNHCLJ), Asst. Ch. Engr., Reeves-Ely Labs., 215 E. 91st St., New York 28; *for mail,* 100 Ave. P, Brooklyn 4, N.Y.
Garrett, Elmer E., Jr. (J'29) (CHJNZ), Asst. Ch. Engr., Ford Instrument Co., 32-36—47th Ave., Long Island City 1; *for mail,* 284 Green Ave., Freeport, N.Y.
Garrett, Eric Hanover ('27) (CAMNZ), Mgr., Kendrick & Davis Co., Water St.; *for mail,* 20 Shaw St., Lebanon, N.H.
Garrett, John A. ('18), Sales Engr., Fairbanks, Morse & Co., 2401 Santa Fe Ave., Los Angeles; *for mail,* 1422 Wayne Ave., South Pasadena, Calif.
Garrett, Olen (J'40) (AE), Draftsman, Douglas Aircraft Co., Inc., El Segundo; *for mail,* 3451 Cloverdale, Los Angeles, Calif.
Garrett, Seymour S. ('16), 115 Oak Hill Rd., Ithaca, N.Y.
Garrety, John Francis (J'39) (CDMAJL), Asst. Oper. Mgr., Empire State Inc., Rm. 503, 350—5th Ave., New York 1, N.Y.
Garrish, George (J'42), Estimating Engr., Mech. Handling Systems Inc., 4600 Nancy, Detroit 12; *for mail,* 8804 Mt. Elliott, Detroit 11, Mich.
Garrison, Van Buskirk (J'39) (MCDJH), Engr., Air Reduction Sales Co., 181 Pacific Ave., Jersey City; *for mail,* Box 25, Haworth, N.J.
Garrison, Wyckoff L. ('16; '24), Asst. Mgr., Loco. Dept., Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.; *for mail,* 612 Embree Crescent, Westfield, N.J.
Garson, Thorvald Nagelstad ('18) (DJ), Chmn. of Bd., Treas., Bergen Point Iron Works, Bayonne, N.J.; *for mail,* 45 Bayview Pl., Ward Hill, Staten Island 4, N.Y.
Garst, Robert Ernest (J'46), 2140 Eastview Ave., Louisville, Ky.
Garstang, Donald Buie (J'40) (CDHAB), 957 Gramercy Dr., Los Angeles 6, Calif.
Gartmann, Hans ('30; '35) (HSBA), Engr., Pump & Compressor Dept., De Laval Steam Turbine Co., Nottingham Way, Trenton 2, N.J.
Gartz, William Joseph (J'36) (HBAMOS), Mech. Engr., Yeomans Bros., Co., 1433 N. Dayton St., Chicago 22; *for mail,* 15 N. Keeler Ave., Chicago 24, Ill.
Garvey, Robert Patrick (J'40) (CMK), Process Engr., Servel, Inc., Evansville 20; *for mail,* 1216 Washington Ave., Evansville 14, Ind.
Garwood, John Leslie (J'40) (CAY), Tire Constr. Engr., B. F. Goodrich Co., 5400 E. Olympic St.; *for mail,* 6034 Easton St., Los Angeles 22, Calif.
Gary, Francis Phinizy ('21; '35) (D), Partner, Sales Engr., Lawrence V. Fraley & Son, 6169 Westminster Pl., St. Louis 12; *for mail,* 339 N. Taylor Ave., St. Louis 8, Mo.
Gary, Hartwell Henry ('17) (CMJNB), Chmn. of Bd., Gary Steel Products Corp., 1225 W. 25th St., Norfolk; *for mail,* 1909 Claremont Ave., Norfolk 7, Va.
Gaston, Charles Murray ('45), Drafting Supvr., Bell Tel. Labs., Inc., 180 Varick St., New York 14, N.Y.
Gaston, Ernest Charles ('39), Engr., Commonwealth & So. Corp., 600 N. 18th St., Birmingham 2, Ala.
Gaston, John Hosking (J'44), Seaman, 1st Class, Co. 1562, Naval Trng. Center, Great Lakes, Ill.
Gaston, William Isaac (A'30) (GHMN), 245 N. 6th St., Newark 7, N.J.
Gasvoda, Robert F. (J'43) (EZSFMN), Research Engr., Stand. Oil Co. of Ind., Whiting, Ind.
Gatcombe, Ernest Kenneth (J'43) (NBKAH), Asst. to Ch. Engr., Jackson & Moreland Engrs., 31 St. James Ave., Boston; *for mail,* 10 Dana St., Cambridge, Mass.
Gately, William A. ('12; '19; '22) (CGTL), Exec. Dir., Hospital Bur. of Stands. & Supplies, Inc., 247 Park Ave., New York 17; *for mail,* 1034 Park Ave., New York 28, N.Y.
Gates, Arthur Oliver ('44) (HSDKNL), Treas., Mech. Engr., Utility Engrs., Inc., 630 Dooly Bldg., Salt Lake City 1, Utah.
Gates, Edwin Leroy (J'25) (BCN), Secy., Engrs.' Stands. Sec., Allis-Chalmers Mfg. Co., 1126 S. 70th St., Milwaukee 1, Wis.
Gates, Kenneth Graham (J'43) (JMBHC), Plant Engr., Charge of Maint., Link-Belt Ltd., 791 Eastern Ave.; *for mail,* 1 Butternut St., Toronto, Ont., Can.
Gates, Leroy Grant ('20) (EHSN), 2008 Brundate Lane, Bakersfield, Calif.
Gates, Robert Laverne (J'44), 481 E. 147th St., Cleveland 10, Ohio.
Gates, Robert McFarland ('18; F'36) (CFS), *Manager,* '28-'31, *Vice-President,* '31-'33, *President,* '44; Pres., Air Preheater Corp., 60 E. 42nd St., New York 17, N.Y.
Gates, Samuel J. ('16; '25) (OS), Partner, Gates, Moe & Weiss, 611 N. Broadway, Milwaukee 2, Wis.
Gates, Wallace Gordon (J'39) (CMD), Head, Methods & Stands. Dept., La Plant-Choate Mfg. Co., Inc.; *for mail,* 3126 C Ave., N.E., Cedar Rapids, Iowa.
Gatewood, Arthur Randolph ('40) (BEKGSH), Prin. Engrg. Surveyor, Am. Bur. of Shipping, 47 Beaver St., New York 4, N.Y.
Gathman, David W. (J'39) (CDJ), 310 Union Ave., Williamsport 12, Pa.
Gatje, Frederick C. ('20; '23; '35) (SFELKO), Sales Engr., Simler & Sengstaken, 271 Madison Ave., New York 16; *for mail,* 824 E. 22nd St., Brooklyn 10, N.Y.
Gattley, William Edwin (J'45) (CDEJNZ), Supt., Weston Motor & Elec. Co., 179 Main St. N.; *for mail,* 20 Sykes Ave., Weston, Ont., Can.
Gaucher, Jules (J'42) (MCDN), Layout Engr., Simmons Ltd., 4700 St. Ambroise St., Montreal; *for mail,* 1873 Sherbrooke St. E., Montreal 24, Que., Can.
Gaugh, William James (J'44) (ABZKNO), Asst., Dept. of Aero. Engrg., Bldg. 33, Mass. Inst. of Tech.; *for mail,* Rm. 420, Graduate House, M.I.T. Dormitory, Cambridge 39, Mass.
Gault, Charles E. (J'39), U.S. Army; *for mail,* Route 3, 6200 Blue Ridge Cutoff, Kansas City, Mo.
Gaum, Carl Gilbert ('27) (BCGNJL), Prof., Univ. Ext. Div., Rutgers Univ., New Brunswick, N.J.
Gausmann, Robert Warner ('28; '40) (FKSZ), Research Dir., Indianapolis Power & Light Co., 17 Meridian St., Indianapolis 6; *for mail,* 6062 Central Ave., Indianapolis 5, Ind.
Gauss, Henry F. ('26) (SFEKJB), Head, Dept. of Mech. Engrg., Univ. of Idaho; *for mail,* 721 E. 1st St., Moscow, Idaho.
Gavelek, William Arnold (J'44) (OZSFEK), Cons. Engr., 120½ S. Grove; *for mail,* 335 Hamilton, Elgin, Ill.
Gavit, Walter Palmer ('28) (SFEHKY), Power Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Gawain, Theodore H. (J'42) (BHK), Cons. Engrg. Dept., Baldwin Loco. Works, Eddystone, Pa.
Gay, Frazer Walker ('29) (EHKS), Asst. Research Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark, N.J.
Gaya, William Leon (J'41) (AEBOFM), Engr., United Fruit Co., Pier 3, North River, New York 6; *for mail,* 100 Stuyvesant Pl., St. George, S.I. 1, N.Y.
Gayer, George F. ('29; '41) (BSEHJA), Ch. Engr., Joshua Hendy Iron Works, Sunnyvale, Calif.
Gayer, John D. (J'37), Ch. Mech. Engr., Brown, Brockmeyer Co., 1000 Smithville Rd.; *for mail,* Apt. 2, 261 Niagara Ave., Dayton, Ohio.
Gayle, Gordon Donald (J'40) (CKDE), Lt. Col., U.S.M.C., Instr., Comd. & Staff Sch., Quantico, Va.; *for mail,* 243 Dillon St., Houston 12, Tex.
Gaylord, Laurence T. ('24) (H), V.P., Dir., Atlantic, Gulf & Pacific Co., 15 Park Row, New York, N.Y.
Gaylord, Lewis Eugene (J'42) (CMZADJ), Pvt., Army Ground Forces Replacement Depot 2, Ft. Ord, Calif.
Gaylord, William Waterman ('14; '19; '26) (SFEDML), Firm of William W. Gaylord, Cons. Engr., 131 Underhill Rd., Hamden 14, Conn.
Gayman, Bert A. ('14) (D), Retired; 555 Occidental Ave., San Mateo, Calif.
Gayman, W. Merlin (J'41) (BHW), Design Coordinator, Research Lab., Curtiss-Wright Corp., Buffalo 5, N.Y.; *for mail,* 712 Magnolia Ave., Pasadena 6, Calif.
Gaynor, Herbert Fred (J'44), c/o Meyer Gaynor, 477 Sheffield Ave., Brooklyn 7, N.Y.
Gaynor, Tom A. (J'40) (JD), Sales Mgr., Thos. A. Gaynor Co., 2900—11th Ave., S.W., Seattle 4, Wash.
Gayton, Loran DeLancey ('24) (SHF), Asst. City Engr., City of Chicago, Rm. 402, City Hall, Chicago 2, Ill.
Gazin, William Rynard ('43) (SFE), Mech. Engr., Middle West Serv. Co., 20 N. Wacker Dr., Chicago 6, Ill.
Gazzam, Jos. P. ('02), 4944 Lindell Blvd., St. Louis, Mo.
Gearon, Gerald ('27), Supvg. Mech. Engr., Boiler & Refrigeration Inspc. Dept., City of Chicago, 119 N. La Salle St., Chicago 2, Ill.
Geary, Donald Ervin (J'43) (AEYFJI), Mech. Engr. & Engrg. Draftsman, Inglewood Branch, N. Am. Aviation, Inc.; *for mail,* 5355 W. 120th St., Inglewood, Calif.
Gebben, Garold Edward (J'40) (LJM), Jr. Engr., Keller Brass Co., 955 Godfrey, S.E., Grand Rapids; *for mail,* 119 Barrett St., Grandville, Mich.
Gebert, Russell Charles, Jr. (J'43) (TCDMYZ), 315 S. Sterling Rd., Elkins Park 17, Pa.
Gebhard, Leslie Nathaniel (A'31) (T), Treas., Wool Buyer, Noyes-Gebhard Co.; *for mail,* 37 Harrison St., Taunton, Mass.
Gebhardt, Charles Wendel ('43) (BJKNOY), Devel. Engr., Twin Coach Co., W. Main St.; *for mail,* 330 E. Grant, Kent, Ohio.
Gebhardt, Geo. F. ('04) (FKS), Prof. Emeritus Mech. Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago, Ill.; *for mail,* 469 N.E. 69th St., Miami, Fla.
Gebhart, Arthur Fletcher, Jr. (J'44) (ABCRS), Graduate Student, Old Dormitory, Calif. Inst. of Tech., Pasadena 4, Calif.
Gebhart, Henry ('19) (RCABES), 611 Broad Blvd., Dayton 9, Ohio.
Geddes, Leslie H. ('34) (BCM), V.P., Greenlee Bros. & Co., Rockford, Ill.
Geenens, Leo ('24) (SBOJY), Elec. Engr., N.Y. City Tunnel Authority, 10 State St., New York 4, N.Y.

Geer, Roger Loran ('45) (JLM), Asst. Prof. & Acting Head of Matls. Processing Dept., Rand Hall, College of Engrg., Cornell Univ., Ithaca, N.Y.
Geers, Joseph F. ('29) (CHJ), Pres., Index Mchy. Corp., 49 Central Ave., Cincinnati, Ohio.
Geertz, Allan Ormsbee ('42) (FR), Fuel Engr., Pa. R.R., 30th St. Sta., Philadelphia 2, Pa.
Gefvert, Charles John (23) (NOS), Retired; 390 Riverside Dr., New York 25, N.Y.
Gegan, Ambrose Joseph (J'40) (SEFKZ), Devel. Engr., Combustion Engrg. Co., Inc., Chattanooga, Tenn.; *for mail,* 953 Highland View, N.E., Atlanta, Ga.
Gegan, John Brandon (J'42) (ASC), Helper, Gen. Constr. Dept., Pacific Gas & Elec. Co., 245 Market, San Francisco; *for mail,* 529 Talbot Ave., Albany, Calif.
Gehlhar, Norman Wallace (J'41) (YLM), U. S. Rubber Co., Eau Claire, Wis.
Gehres, Hewitt A. ('41) (EFKR), V.P., Dir. Engrg., Cooper-Bessemer Corp., Mt. Vernon, Ohio.
Gehrig, Eugene Joseph ('46) (L), Mech. Engr. (Milling), Quaker Oats Co., Bd. of Trade Bldg., 141 W. Jackson Blvd., Chicago; *for mail,* 2060 Ridge Ave., Evanston, Ill.
Geier, Frederick V. (A'28) (CN), Pres., Cincinnati Milling Mch. Co., Oakley, Cincinnati 9, Ohio.
Geiger, Erwin John (J'43) (ANBSHC), Aviation Machinist's Mate, 3rd Class, Naval Air Corps, FAETU-PAC, Det. 1, Naval Auxiliary Air Sta., Ream Field, San Ysidro, Calif.; *for mail,* 6041 S. Sacramento Ave., Chicago 29, Ill.
Geiger, John W. ('27; '33; '35) (ESKFAR), Asst. Prof. Mech. Engrg., Purdue Univ., West Lafayette, Ind.
Geiger, Joseph D. (J'40) (CDMLN), Pres., Geiger Iron Works, Inc., Scotts Ave. & Pilgrim St., Stockton, Calif.
Geiger, Walter Case ('34; '44) (FCL), Asst. Ch. Engr., Armour Leather Co., 500 Arch St., Williamsport, Pa.
Geisel, Charles, Jr. (J'41) (MCDZY), 146 Fairmount Ave., Chatham, N.J.
Geisinger, Joseph Morlyn ('36; '45) (SBKH), Jr. Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 640 Parkview Dr., Detroit 14, Mich.
Geissbuhler, John Otto ('35; '45) (BCJM), Devel. Engr., Gen. Elec. Co., 1133 E. 152nd St., Cleveland 10; *for mail,* 1314 Plainfield Rd., South Euclid 21, Ohio.
Geissler, William E. ('37), Ch. Draftsman, Petree & Dorr Engrs., Inc., 120 Wall St., New York; *for mail,* 18 Pocantico Rd., Ossining, N.Y.
Geist, James William (J'45) (LJFCDM), Engr., A. D. Sutherland, Mfrs. Rep., 515 Oliver Bldg., Pittsburgh 19; *for mail,* 2937 Glenmore Ave., Pittsburgh 16, Pa.
Geitmann, Russell J. (J'29) (BDS), Dist. Engr., Link-Belt Co., 548 Rockefeller Bldg., Cleveland 13, Ohio.
Gellert, Nathan Henry ('22) (CFKE), Pres., Seattle Gas Co., 1511—4th Ave., Seattle 2, Wash.
Gellert, Theodore Stanley (J'36) (SKFMCB), Estimator, Air Preheater Corp.; *for mail,* 117 Madison St., Wellsville, N.Y.
Gellman, Allen B. ('44), Pres., Ill. Watch Case Co., 853 Dundee, Elgin; *for mail,* 3540 Thorndale Ave., Chicago, Ill.
Gengenbach, Robert Edwin (J'42) (CMJSLH), Staff Asst. to Supt., Bldg. 38, New Departure Div., Gen. Motors Corp.; *for mail,* 44 Page Ave., Bristol, Conn.
Gennetti, William Thomas (J'45) (BJS), 337 Barnes St., Pittsburgh 21, Pa.
Genthon, Francis Xavier (J'43) (YJLTW), Devel. Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 258 Union Ave., Clifton, N.J.
Gentry, Charles Rogers (J'43) (SBJFH), Design Engr., Reynolds Metals Co., 5th & Cary Sts., Richmond 19; *for mail,* 2219—1st Ave., Richmond 22, Va.
Genung, Burton Edward (J'40) (AM), 983 Clayton, San Francisco, Calif.
George, Abraham (J'40) (MAJBN), 1st Lt., Army Air Forces, Engrg. Officer; 188 Daniel Low Terrace, Staten Island 1, N.Y.
George, Everett Dwaine (J'38) (NYHB), Sr. Mch. Designer, Goodyear Tire & Rubber Co., 1144 E. Market St., Akron; *for mail,* 2446 Chestnut Blvd., Cuyahoga Falls, Ohio.
George, Francis Xavier (J'34) (SKEBHN), Mar. Surveyor, Pillsbury & Martignoni, 220 Bush St., San Francisco 4; *for mail,* 2641 School St., Oakland 2, Calif.
George, Harry Frederick (J'44) (MNSKGB), Engrg. Draftsman, Combustion Engrg. Co. Inc.; *for mail,* 1202 Hanover, Chattanooga, Tenn.
George, Harvey Franklyn (J'45) (ABNKHJ), Stress Analyst, Lark Div., Fairchild Airplane & Eng. Corp., 184-10 Jamaica Ave., Jamaica; *for mail,* 191-54—112th Rd., St. Albans 12, N.Y.
George, Leonard Beattie ('25; '35) (ES), C. & R. Div., Mar. Dept., Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4; *for mail,* 14-A Castleton Pk., Staten Island 1, N.Y.
George, Vincent C. ('31) (BKSEF), Physics Instr., Los Angeles City College, 855 N. Vermont; *for mail,* 866 N. Edgemont, Los Angeles 27, Calif.
Georger, Francis Augustus, Jr. (J'45), Plant Engr., Precision Metalsmiths Inc., 6511 Cedar Ave., Cleveland, Ohio.
Georger, Roland Sylvester ('45) (NBCHDG), Partner, Reimann & Georger, Ellicott Sq., Buffalo 3; *for mail,* 119 Mt. Vernon Rd., Snyder 21, N.Y.
Georgian, John C. (J'40) (BESN), Engr., Charge of Torsional Vibration Analysis, Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7; *for mail,* Box 656, R.F.D. 5, Waukesha, Wis.
Gerber, Samuel R. ('41) (CJM), Pres., Gen. Mgr., Kent Metal Mfg. Co., Inc., 490 Johnson Ave., Brooklyn 6, N.Y.
Gerdes, Henry Albert (J'42) (SEKFO), Lt., U.S.N.R., Ship Supt., Naval Shipyard, Philadelphia 12, Pa.; *for mail,* 531 Fairway Rd., Ridgewood, N.J.
Gerdes, Wayne R. (J'41) (RMS), Gen. Elec. Co., 409 S. 17th St., Omaha 2, Neb.
Gerding, John E. (J'41) (CDM), 2057 Cardiff Rd., Schenectady 7, N.Y.
Gerhard, Sidney (J'45) (AZCM), Cpl., Devel. Engr., Supersonic Wind Tunnels Lab., Aberdeen Proving Ground, Md.; *for mail,* 1405—42nd St., Brooklyn 19, N.Y.
Gerla, Mrs. Miriam Kleeger (J'44) (HKNBAY), Devel. Engr., Shirgun Corp., 136 W. 52nd St., New York 19; *for mail,* 936—55th St., Brooklyn 19, N.Y.
Gerla, Morton (J'37) (NBJHOD), Project Engr., Ord. Design Supvr., Shirgun Corp., 136 W. 52nd St. New York 19; *for mail,* 936—55th St., Brooklyn 19, N.Y.
German, Abraham Johannes ('21) (SFKHCE), Ch. Engr. of Power Plants, Scovill Mfg. Co., Mill St., Waterbury 91; *for mail,* 22 Adelaide, Waterbury 61, Conn.
German, John George (J'44) (RSNBCD), Asst. to M.M., Hillyard Depot, Great No. Ry. Co.; *for mail,* E. 2727 Queen Ave., Spokane 14, Wash.
Germer, Fred W. (J'43) (MNBJC), Pres., Gen. Mgr., Ch. Engr., Advanced Aircraft, Inc., 33-22 Northern Blvd.; *for mail,* 43-41—45th St., Long Island City 1, N.Y.
Germond, Earl Goerke (J'40) (KCDJSN), Plant Engr., Natl. Carbide Corp., Ivanhoe, Va.
Germond, Robert Work (J'45) (BCJ), Engr., Charge of Research & Sales, R. P. Kincheloe Co., 1411 Fannin, Houston; *for mail,* Apt. 19, 1321 S. Main St., Houston 2, Tex.
Gero, George (J'42) (AHK), Pvt., Btry. E, 34th Bn., 8th Regt., Field Arty. Replacement Trng. Center, Ft. Sill, Okla.
Gerritzen, Howard Kenneth (J'43) (ACFHMS), Capt., Army Air Forces, Project Engr., Propeller Lab., Wright Field, Dayton; *for mail,* 1628 San Rae Dr., Dayton 9, Ohio.
Gershberg, Joseph ('32) (FHS), Div. Engr., Mech. Testing, Brooklyn Edison Co., 1 Hudson Ave., Brooklyn, N.Y.
Gershenow, Harold Joseph (J'44), Mass. Inst. of Tech.; *for mail,* Apt. 1, Drayton Hall, 48 Boylston St., Cambridge, Mass.
Gersoni, Louis J. ('41) (SFLMOD), Ch. Engr., Hygrade Food Products Corp., 30 Church St., New York, N.Y.
Gerstmyer, Robert George (J'43) (CMODYZ), Graduate Student, Wharton Graduate Sch., Univ. of Pa., Philadelphia, Pa.; *for mail,* 1510 Shadyside Rd., Baltimore 18, Md.
Gerstung, Harry Otto (J'43) (SAC), Steam Field Engr., Westinghouse Elec. Corp., 20 N. Wacker Dr., Chicago 6; *for mail,* 1819 W. Morse Ave., Chicago 26, Ill.
Gerstung, Harry Sinclair (J'41) (ESF), Lt. (j.g.), U.S.N., Naval Engrg. Exper. Sta., Annapolis, Md.
Gerzetich, Michael Augustine ('45) (FKS), Boiler Rm., Commonwealth Edison Co., Box 149, Pekin, Ill.
Gesell, Wm. H. ('18; '21; '35) (C), V.P., Lehn & Fink Products Corp., Bloomfield, N.J.
Gess, Louis ('28; '36) (HZB), Brown Instrument Co., Wayne & Roberts Ave., Philadelphia 44, Pa.
Gessling, George Fisher (J'39) (KL), Lt. (j.g.), U.S.N.R., Repair & Maint. Officer, Stand. Landing Craft Unit 62, F.P.O., San Francisco; *for mail,* 55 Kenyon Ave., Berkeley, Calif.
Gessner, Egon Fritz (J'41) (JMS), Ensign, U.S.N.R., 1627 Massachusetts Ave., Cambridge, Mass.
Gessow, Irving David (J'45) (YBKJTA), Matls. Engr., Consultant, Code 631, Bur. of Ships, Navy Dept., Washington, D.C.
Gethen, George S. ('43) (SFK), Sr. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.; *for mail,* 1106 Belmont Ave., Collingswood, N.J.
Getsug, Bertram ('34; '45) (JSE), Mech. Engr., Naval Air Trng. Bases, Pub. Works Dept., Bldg. 45; *for mail,* 1906 E. Bobe St., Pensacola, Fla.
Gettinger, Leonard A. (J'37) (FLS), Mech. Engr., Bailey Meter Co., 8527 Rosalie Ave., Brentwood, St. Louis, Mo.
Getz, Horace Eshelman, Jr. (J'45) (KHNZL), Jr. Mech. Engr., Field Research Dept., Magnolia Petroleum Co., Box 900, Dallas 1; *for mail,* 226 S. Edgefield, Dallas 11, Tex.
Getzen, John Edwin ('25; '28; '35) (CADMJS), Assoc. with Harry G. Mouat, 744 Martin Bldg., Birmingham 3, Ala.
Getzoff, Edward M. ('19; '25; '30) (AZNBYJ), Project Engr., Red Bank Div., Bendix Aviation Corp., Pearl St.; *for mail,* Conover Lane, R.F.D., Red Bank, N.J.
Geuss, Joseph Anthony (J'43) (NBHCLJ), Mch. Designer, Clearing Mch. Corp., 6499 W. 65th St., Chicago 38; *for mail,* 3203 Clinton Ave., Berwyn, Ill.
Gevirtz, Stanley Aaron (J'46), New York Univ., University Heights, New York; *for mail,* 388 Drew St., Brooklyn 8, N.Y.
Gevrenz, Theodore M. (J'39) (CELMRS), Ch. Engr., Indus. Oil Sales, Socony-Vacuum Oil Co., Inc., 230 Park Ave., New York 17, N.Y.
Geyer, Henry Elmer ('25; '32; '35) (OLHBCK), Constr., Maint. Engr., Pipe Line Dept., Stand. Oil Co. of Calif., Box 311, Pittsburg; *for mail,* 6449 Harwood Ave., Oakland 11, Calif.
Giacchino, George Edward (J'44) (BKS), 2512 Hillcrest Rd., Drexel Park, Pa.
Giampiccolo, James S. (J'41) (M), Aero. Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* 3308 Wallace Ave., New York, N.Y.
Gianelloni, Vivian Joseph ('17; '19; '24) (CBSRKD), V.P., Punta Alegre Sugar Sales Co., Apartado 1938, Havana, Cuba.
Giauque, Robert E. (J'36) (KS), Pres., Roberts-Konrad, Inc., 90 West St., New York 6, N.Y.
Gibb, John Collins ('44) (ERSFHK), Staff Engr., Lub. Dept., Tech. Div., Socony-Vacuum Oil Co., Inc., Rm. 1350, 26 Broadway, New York 4, N.Y.
Gibbons, Edward J. (J'42) (KLM), Camramatic Corp., New York; *for mail,* 1153 Grand Concourse, Bronx, N.Y.
Gibbons, James William ('30) (BCJ), Retired; 118-36—191st St., St. Albans 12, N.Y.
Gibbons, Joseph Butler (J'45) (MDCN), Test Engr., Gen. Elec. Co., Schenectady; *for mail,* 285 Lincoln Pl., Brooklyn, N.Y.
Gibbons, Michael Joseph ('17) (CDKO), Owner, M. J. Gibbons Supply Co., 601-31 E. Monument Ave., Dayton; *for mail,* 116 W. Thruston, Dayton 9, Ohio.
Gibbons, Philip Lake (J'41) (MNJSCG), Mfg. Planning Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 24 Springfield Ave., Cranford, N.J.
Gibbs, Francis Orland (J'34) (CM), Gibbs Mch. Co., 1023 W. Lee St.; *for mail,* 403 Aberdeen Terrace, Greensboro, N.C.
Gibbs, Lloyd Tolbert ('45) (BEHJS), Owner, Mgr., Lloyd T. Gibbs Co., Petroleum Bldg., Tulsa 3, Okla.
Gibbs, Maxwell ('44) (OCMBLN), Cons. Engr., 1927 W. 6th St., Los Angeles 5, Calif.
Gibbs, Robert Glenn (J'38) (CSA), Field Engr., Gen. Elec. Co., 235 Montgomery St., San Francisco 6, Calif.
Gibbs, Russell Everett ('31; '35) (SEFKAJ), Head, Dept. of Mech. Engrg., Mont. State College, Bozeman, Mont.
Gibillini, Emilio Paul (J'45) (CHM), Jr. Design Engr., Pelton Water Wheel Co., 2929—19th St.; *for mail,* 1471 Francisco St., San Francisco, Calif.
Gibney, James Joseph, Jr. (J'45) (SHBNR), Engr., Turbine Control Sec., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 907 Mason St., Schenectady 8, N.Y.
Gibson, Alfred E. ('32) (CDF), Pres., Wellman Engrg. Co., 7000 Central Ave., Cleveland 4, Ohio.
Gibson, Fred'k M. ('13) (FJKLS), Plant Engr., Am. Sugar Refining Co., 49 S. 2nd St., Brooklyn 11, N.Y.
Gibson, Frederick Montague, Jr. (J'40) (JMC), Ensign, U.S.N.R., M.E.I.U. 1, F.P.O., San Francisco, Calif.
Gibson, Geo. ('31; '40) (KLP), C. H. Leach Co., 117 Liberty St., New York, N.Y.; *for mail,* 110 Montclair Ave., Upper Montclair, N.J.
Gibson, Hamilton L. (J'31) (CS), Detroit Edison Co., 6603 W. Jefferson; *for mail,* 776 Beard Ave., Detroit 9, Mich.
Gibson, Harold D. (J'24) (BESONK), Mech. Engr., Equip. Devel., Stand. Oil Devel. Co., Box 37, Elizabeth, N.J.; *for mail,* 86 Crosshill St., Staten Island 1, N.Y.
Gibson, James (J'42) (WNCMHJ), Sr. Designer, Laighpark Works, T. White & Sons Ltd.; *for mail,* 39 Endrick Dr., Paisley, Scotland.
Gibson, James Edwin ('96; '01; '10) (CHO), Mgr., Engr., Commrs. of Pub. Works, 14 George St., Charleston A, S.C.
Gibson, John Stewart ('37) (SRJDN), Sales Engr., Drake Non Clinkering Furnace Block Co., 310 Renshaw Bldg., Pittsburgh 22, Pa.
Gibson, Norman R. ('27) (FHS), V.P., Buffalo Niagara Elec. Corp., 600 Elec. Bldg., Buffalo 3, N.Y.
Gibson, Robert MacDowell ('22; '35) (BELNRY), Engr., Koppers Co., Koppers Bldg., Pittsburgh 19, Pa.
Gibson, William A. ('44), Cons. Engr., 29-33 Wilson St., Redfern, Sydney, Australia.
Gibson, William Robert ('30) (SHKB), Owner, Mgr., Northwest Filter Co., 122 Elliott Ave. W., Seattle 99, Wash.
Gichner, Jacob Herman ('25; '32; '35) (CFJNRK), Comdr., U.S.N.R., Bur. of Ord., Navy Dept., Navy Bldg., Washington; *for mail,* 2101 New Hampshire Ave., N.W., Washington 9, D.C.
Giegengack, Augustus E. ('34) (GYC), Pub. Printer, Govt. Ptg. Office, N. Capitol & H Sts., Washington 25, D.C.
Giering, Percival L. ('45) (CO), Sr. Engr., Stevenson, Jordan & Harrison, 19 W. 44th St., New York 18; *for mail,* 60 Lakeville (Bates) Rd., Great Neck, N.Y.
Gies, Robert Paul (J'43), 8079 Whittaker Ave., Detroit 9, Mich.

Giesecke, Frederick Ernest ('36) (OGHBKZ), Cons. Engr., 276 E. Coll, New Braunfels, Tex.
Giesler, Jean V. ('23), Ch. Engr., Fulton Sylphon Co.; *for mail,* Box 104, Knoxville, Tenn.
Giffin, Daniel Anthony (J'43) (JOCBM), Student Test Engr., Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.; *for mail,* 22 W. 18th Ave., Eugene, Ore.
Giffin, Leverett William (J'39) (ABM), Engr., Boeing Aircraft Co., Seattle, Wash.; *for mail,* 2040 S. Columbine, Denver 10, Colo.
Gifford, Albert James ('03;'31) (AHC), Treas., Leland-Gifford Co., Box 989, Worcester 1, Mass.
Gifford, Charles R. ('26) (DFKLSY), Owner, Firm of C. R. Gifford, 7310 Woodward Ave., Detroit 2, Mich.
Gifford, Robt. L. ('02), Pres., Ill. Engrg. Co., cor. 21st St. & Racine Ave., Chicago, Ill.; *for mail,* 1231 S. El Molino Ave., Pasadena, Calif.
Giguere, George Hector ('44) (SCOKLM), Ch. Engr., Smith, Hinchman & Grylls, 800 Marquette Bldg., Detroit; *for mail,* 2253 Burns, Detroit 14, Mich.
Gil, Ramón I. ('26; '33; '35) (SHKBAR), Assoc. Prof. Mech. Engrg., College of Agric. & Mech. Arts, Supt. Bldg. & Grounds, Univ. of P.R.; *for mail,* 114 Hostos, Mayaguez, P.R.
Gilbert, Carl L. (J'27) (CMS), Lt. Comdr., U.S.N.R., Navy 728, Box A, F.P.O., New York, N.Y.
Gilbert, Earl Clark, Jr. (J'45), 4726 Bloomington Ave. S., Minneapolis, Minn.
Gilbert, Edward (J'43) (BHKEMN), Jr. Engr., Westinghouse Elec. Corp., Lester P.O.; *for mail,* 1844 Nolan St., Philadelphia 38, Pa.
Gilbert, Ernest M. ('16) (SFKHO), Chmn. of Bd., Ch. Engr., Gilbert Associates, Inc., 412 Washington St., Reading 7, Pa.
Gilbert, Harold Maier (J'40) (MCA), Design Engr. on Modifications, Douglas Aircraft Co., Inc., Santa Monica; *for mail,* 855½ N. Alta Vista Blvd., Los Angeles 46, Calif.
Gilbert, James R., Jr. (J'39) (FKMCZS), 9005 Muskegon Ave., Chicago 17, Ill.
Gilbert, Kermit C. (J'31) (CSZ), 1028—43rd St., Belview Heights, Birmingham, Ala.
Gilbert, Nathan (J'41) (BNMHKS), Asst. Mech. Engr., Design Dept., N.Y. Naval Shipyard, Brooklyn; *for mail,* 15 Westminster Rd., Brooklyn 18, N.Y.
Gilbert, Norman Roland (J'38) (MCLBJZ), Mech. Engr., Univ. of Calif., Box 1663, Santa Fe, New Mex.
Gilbert, Russell Lowe (J'24) (BJKNSY), Mech. Engr., Taylor Elec. Mfg. Co., 625 Princess Ave.; *for mail,* 522 Piccadilly St., London, Ont., Can.
Gilbert, William Wayne (J'35) (MNLA), Assoc. Prof., Metal Processing Dept., Univ. of Mich., Ann Arbor, Mich.
Gilbertson, John Stewart, Jr. (J'44) (ASEB), Jr. Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 170 N. Beacon St., Hartford 5, Conn.
Gilbreth, Frederick M. (J'38) (CMD), Assoc. Engr., McKinsey & Co., 60 E. 42nd St., New York 17, N.Y.
Gilbreth, Mrs. Lillian Moller ('26; F'45) (CMO), Pres., Gilbreth, Inc., 30 The Crescent, Montclair, N.J.
Gilbreth, William Moller (J'36) (EF), Indus. Lub. Sales Engr., Gulf Oil Corp., 17 Battery Pl., New York; *for mail,* 12 Delmar Pl., Delmar, N.Y.
Gildea, David Jerome (J'45), Asst. Prof., Aero. Engrg. Dept., Pa. State College; *for mail,* 116 W. Hillcrest Ave., State College, Pa.
Giles, Charlie M. (J'43) (KBJM), Ch. Engr., Sweden Freezer Mfg. Co., 1140 W. 53rd St., Seattle 7; *for mail,* 2617—29th Ave. W., Seattle 99, Wash.
Giles, James Edward, Jr. (J'45) (AMCOJK), Asst. Project Engr., Jackson & Moreland, 31 St. James Ave., Boston 16; *for mail,* 44 Olive St., Newburyport, Mass.
Giles, Stuart (J'40) (MCE), Lt., U.S.N.R., Spec. Devices Div., Office of Research & Inventions, 610 H St., Washington 2, D.C.; *for mail,* 806 Greenwood Ave., Takoma 12, Md.
Gilg, Frank Xavier ('41) (FSWOE), Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Gill, Charles A. ('21) (RYZM), V.P., Opera. & Maint., Reading Terminal, 12th & Market Sts., Philadelphia 1; *for mail,* 1026 Bellevue-Stratford Hotel, Philadelphia 2, Pa.
Gill, Edw. H. ('37), Pres., Dixie Steel Mfg. Co., Box 752; *for mail,* 3615 Waynoka St., Memphis, Tenn.
Gill, Harry Verlaine ('44) (CEFKR), M.M., Albuquerque Div., Atchison, Topeka & Santa Fe Ry. Co., 80 E. Jackson Blvd., Chicago, Ill.; *for mail,* Box 128, Winslow, Ariz.
Gill, James Presley ('41) (JMCN), Exec. V.P., Vanadium Alloys Steel Co., Latrobe, Pa.
Gill, Joseph Ralph ('18) (FSTDMC), Consultant, Am. Enka Corp., Enka; *for mail,* R.F.D. 2, Asheville, N.C.
Gill, Neal Fassig (J'41) (SKFCZ), Supvg. Engr., Steam Dept., Cleveland Elec. Illum. Co., 75 Public Sq., Cleveland 13; *for mail,* 3120 Kensington Rd., Cleveland Heights 18, Ohio.
Gill, Thomas Tapscott ('44) (BH), 867 N. Lafayette Park Pl., Los Angeles 26, Calif.
Gill, Walter M., Jr. (J'40) (CEH), 1410 W. Clinch, Knoxville, Tenn.
Gille, Hugo Edward (J'37), Lt., W.T.S. Navy Sch., Univ. of Tenn., Martin, Tenn.
Giller, F. S. ('44), Fairseat, Hextable, Kent, England.
Gillespie, Charles Wesley (J'39) (ACGMOH), 129 Oak Lane, Cranford, N.J.
Gillespie, Fontaine M. ('22;'35) (CHS), Supt., Lages Div. & Power Plant, Rio de Janeiro Tramway, Light & Power, Ltd., Caixa do Correio 571, Rio de Janeiro, Brazil, S.A.
Gillespie, John J. ('44) (DCMBKJ), Plant Layout Engr., Head Gp. in Plant Planning, East Main Plant, Bridgeport Brass Co., Bridgeport 2; *for mail,* 1663 Main St., Bridgeport 4, Conn.
Gillespie, Robert George (J'36) (FSY), Automotive Engr. (Acting), Toronto Trans. Comm., 35 Yonge St.; *for mail,* 1 De Vere Gardens, Toronto, Ont., Can.
Gillett, Chas. E. ('13; '21), Ch. Inspr., Norton Co., 1 New Bond St.; *for mail,* 4 Bay State Rd., Worcester, Mass.
Gillett, John ('20) (DSHMLO), Partner, Bellman-Gillett & Richards, 518 Jefferson Ave., Toledo 4, Ohio.
Gillette, Edmond Stephen ('40) (ACDMOZ), Capt., U.S.N., 1605 Georgina Ave., Santa Monica, Calif.
Gilliam, Howard Henry ('24; '35) (ACM), Dir. of Expediting, Am. Type Founders, Inc., 200 Elmora Ave., Elizabeth B; *for mail,* 372—10th Ave., Paterson 4, N.J.
Gillies, John Ackland (J'41) (AEC), Installation Engr., Canadian Pratt & Whitney Aircraft Co., Ltd., Longueuil, Que., Can.
Gillim, William Gates (J'38) (SK), Asst. Supt., Customer Serv. Dept., N.Y. Steam Corp., 130 E. 15th St., New York 3; *for mail,* 101 Paulding Ave., Tarrytown, N.Y.
Gilling, Ethelbert Newman ('19) (BJM), Ch. Draftsman, Continental Roll & Steel Fdy. Co., 903 Grant Bldg., Pittsburgh 19; *for mail,* 959 Davis Ave., Pittsburgh 12, Pa.
Gillivan, Charles Z. ('25) (CFSKZ), Gen. Supt., Ch. Engr., Columbus & So. Ohio Elec. Co., 215 N. Front St., Columbus 15, Ohio.
Gillmor, Reginald Everett ('40) (CZYFA), V.P., Pub. Relations, Sperry Corp., 30 Rockefeller Plaza, New York 20, N.Y.
Gillon, Verser C. ('40) (ACMSY), Prod. Control Supt., Consltd. Vultee Aircraft Corp., Ft. Worth; *for mail,* 4309 Donnelly, Ft. Worth 7, Tex.
Gillroy, Bernard J. ('29), Supt. of Bldgs., Borough of Manhattan, Dept. of Housing & Bldgs., Municipal Bldg., New York, N.Y.
Gillum, Robert Greene (J'44) (KEHCAB), Engr., Graduate Trng. Program, Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.; *for mail,* 144 Madison Blvd., Terre Haute, Ind.
Gilman, Franklin W. ('13; '15; '20) (DKLS), Ch. Engr., Loft Candy Corp., 38-38—9th St., Long Island City 1, N.Y.
Gilman, Marvin A. (J'42) (BNJHZ), Test Engr., Matl. Lab., N.Y. Naval Shipyard, Brooklyn; *for mail,* 34 W. 65th St., New York 23, N.Y.
Gilman, Ray Edward (J'45) (SF), Power Engr., S. D. Warren Co., Cumberland Mills; *for mail,* Box 385, Gorham, Me.
Gilmer, G. Walker, III (J'41) (ABJ), *Student Award,* '41; Supvr., Maint. Prod., Am. Airlines, Inc., La Guardia Field, New York, N.Y.
Gilmor, Robert E. (J'46), 1315 Adams St., Denver 6, Colo.
Gilmore, Charles Gordon (J'45), Staff Engr., Am. Viscose Corp.; *for mail,* Route 1, Meadville, Pa.
Gilmore, Gerald Donald (J'46), Plastics Research Dept., Dow Chem. Co., Midland; *for mail,* c/o Mrs. Lona Gilmore, Kent City, Mich.
Gilmore, Gordon Malcom ('31) (BCJ), Sales Engr., Los Angeles Steel Casting Co., 2444 S. Alameda, Los Angeles 11, Calif.
Gilmore, John Winfield ('27) (SKJBFE), Research Engr., Boiler Div., Engrg. Dept., Fidelity & Casualty Co., 80 Maiden Lane, New York 7; *for mail,* 215 W. 91st St., New York 24, N.Y.
Gilpatrick, Arlo E. (J'40) (ACMONL), Sr. Engr., Aircraft Engrg. Dept., Am. Overseas Airlines, Inc., La Guardia Field, New York, N.Y.
Gilroy, John Annan ('34; '45) (SFECDK), Supt. Steam Prod., Wyman Gordon Co., Harvey; *for mail,* 9225 S. Hoyne Ave., Chicago 20, Ill.
Gilson, Harry W. ('09; '17; '35) (SFDB), Pres., Sales Engr., Vincent-Gilson Engrg. Co., 30 Church St., New York 7, N.Y.; *for mail,* 124 Sheridan Terrace, Ridgewood, N.J.
Ginsparg, Joshua Ovsel (J'43) (NMLJ), Mech. Design Engr., RCA Victor Div., Radio Corp. of Am., Camden, N.J.
Ginter, John Emil (J'46), Engr., Charge of Prod., Miss. Plant, LeTourneau Co. of Ga., Vicksburg, Miss.
Gioia, Joseph Charles (J'44) (MCDNBW), Mfg. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny, N.J.; *for mail,* 50 Westminster Rd., Brooklyn 18, N.Y.
Giordano, Joseph (J'46), Baldwin Loco. Works, Eddystone, Pa.
Girard, Arnold Thomas (J'44) (WDSLNC), Engrg. Instr., Univ. of Toronto; *for mail,* Apt. 6, 223 Woodbine Ave., Toronto, Ont., Can.
Girardi, Giovino (J'43) (SABJ), Asst. Mch. Shop Planning Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 468 S. 17th St., Newark 3, N.J.
Girouard, Robert Louis (J'44) (NAHJMC), Mech. Engr., Charge of Planning, Natl. Advis. Com. for Aero., Langley Field; *for mail,* 36 Henry St., South Hampton, Va.
Girton, William Zender (J'46), Ensign, U.S.N.R., YF-881, F.P.O., Key West, Fla.
Girvin, Harvey F. ('27) (BCJ), Prof. Mech. Engrg., Purdue Univ., West Lafayette, Ind.
Gish, James Albert, Jr. ('14; '35) (CM), Supt., Sandt's Eddy Plant, Lehigh Portland Cement Co., Box 149, Easton, Pa.
Gisler, Max ('43), Chemotek Mfg. Co., 4400 Clinton Dr., Houston 10, Tex.
Gismond, Howard E. ('45) (SC), Mech. Engr., Dayton Power & Light Co., 25 N. Main, Dayton 1, Ohio.
Gisonno, George L. (J'42) (BSAE), Asst. Ch. Designer, L. J. Wing Mfg. Co., 154 W. 14th St., New York; *for mail,* 864 Atlantic Ave., Brooklyn, N.Y.
Githens, Thomas Francis ('15; '17; '24) (NMJBZA), Mech. Engr., Charge of Engrg., Cleveland Twist Drill Co., 1242 E. 49th St., Cleveland 14, Ohio.
Givan, Albert ('13) (HEC), Cons. Engr., Sacramento Mun. Utility Dist., 800 Fruit Bldg., Sacramento 14; *for mail,* 2175—35th St., Sacramento 17, Calif.
Givens, James Irvin (J'42) (AERJKS), Lt., Army Air Forces, Engrg. Officer, 478th Bomb. Sqd., Army Air Base, Lake Charles, La.; *for mail,* 207 Clark St., McComb, Miss.
Gjesdahl, Donald Jacob (J'44) (BCH), Prod. Engr., Fisher Flouring Mills Co., Harbor Island; *for mail,* 4205 Francis Ave., Seattle, Wash.
Gladden, Charles Stephen ('01; '25) (SKFA), Hotel St. George, Clark St. at Henry, Brooklyn, N.Y.
Gladeck, Frederick Charles ('16; '24) (NOJMC), Pres., Mch. & Tool Designing Co., 1011 Chestnut St., Philadelphia 7, Pa.
Glancy, Walter Perry (J'40) (CMYA), Mfg. Engr., Lockheed Aircraft Corp., 2555 N. Hollywood Way, Burbank; *for mail,* 1360 Kent Pl., Glendale 5, Calif.
Glas, Anatol ('37) (SLJNBF), Project Engr., Perth Amboy Plant, Am. Smelting & Refining Co., Barber, N.J.
Glasby, John Baizley ('36) (ESF), Power, Test Engr., Atlantic Refining Co., 260 S. Broad St., Philadelphia 1; *for mail,* 2601 Parkway, Philadelphia 30, Pa.
Glaser, C. E. (J'13), 412 Eaton Ave., Hamilton, Ohio.
Glaser, Sidney (J'38) (ABGE), Prod., Graphic Arts, Metro-Goldwyn-Mayer Pictures, 1540 Broadway, New York 19; *for mail,* 501 Wyona St., Brooklyn, N.Y.
Glasgow, Arthur Graham ('92), Chmn., Humphreys & Glasgow, Ltd., Humglas House, Carlisle Pl., London, S.W. 1, England.
Glasgow, Carr Lane ('04; '13) (WLEFS), Pres., Calcasieu Paper Co., Elizabeth, La.
Glasgow, Clarence Ogden (J'34) (JMCNY), Ch. Engr., Natl. Tank Co., Box 1710, Tulsa 1, Okla.
Glasgow, John G. ('18), Asst. Mgr., Mfg. Div., Gulf Refining Co., Gulf Bldg., Box 1214, Pittsburgh, Pa.
Glasgow, William Harry (J'42) (ENAMYC), Draftsman, Engrg. Dept., Chef Boy-Ar-Dee Quality Foods, Inc., Milton, Pa.
Glasius, Einar ('28; '35), Mech. Engr., Great Lakes Steel Corp., Ecorse, Detroit; *for mail,* 9498 Grosse Ile Pkwy., Grosse Ile, Mich.
Glass, Joseph E. (J'39) (CLD), Supt. (of Safety), Jeffersonville Boat & Mch. Co., Jeffersonville, Ind.; *for mail,* 3003 Spencer Ave., Louisville, Ky.
Glassco, James Brinton (J'40) (ABJ), Stress Analyst "A", Douglas Aircraft Co., Inc., Santa Monica; *for mail,* 6114 S. La Brea Ave., Los Angeles 43, Calif.
Glassco, Robert Bond (J'40) (ABJ), Lt. (j.g.), U.S.N.R., Res. Rep., Bur. of Aero., c/o N. Am. Aviation, Inc., Inglewood; *for mail,* 6114 S. La Brea Ave., Los Angeles 43, Calif.
Glasser, Sidney Philip (J'45) (SA), Graduate Student, Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 605 Hampton Ave., Pittsburgh 21, Pa.
Glassey, Philip P. ('45) (CEHJKM), Asst. to Pres., Gen. Admin., Easy Washing Mch. Corp., Solar & Spencer Sts., Syracuse 1, N.Y.
Glassie, Donelson Caffery (A'45) (KMO), Manag-ing Dir., Don Caffery Glassie Co., 208 Munsey Bldg., Washington 4, D.C.
Glauch, Edmund S. ('20; '30) (JR), Mech. Engr., Lubricants Dept., Joseph Dixon Crucible Co., Wayne & Monmouth Sts., Jersey City 3; *for mail,* 49 Diamond Bridge Ave., Hawthorne, N.J.
Glauser, James Ernest (J'42), Aero. Div., Pacific Scientific Co., Inc., 1430 Grande Vista Ave., Los Angeles 23, Calif.
Glazebrook, Robinson Cox ('32; '40) (HSZLNO), Ch. Engr., Hyd. Div., Fairbanks, Morse & Co., Lawton Ave.; *for mail,* 90 E. Ridge Rd., Turtle Ridge, Beloit, Wis.
Glazer, Edward (J'41) (AOM), Lt. (j.g.), U.S.N.R., Naval Air Facility, Columbus 3, Ohio.
Glazer, Marvin Ralph (J'43) (MJCNDO), Partner, Auto Grille Mfg. Co., 5020 Euclid Ave., Cleveland; *for mail,* 1587 Coventry Rd., East Cleveland, Ohio.
Gleason, Gilbert Howe ('06; '12), Retired; 10 Edgehill Rd., Winchester, Mass.

Gleason, James E. ('22), *A.S.M.E. Medallist,* '39; Pres., Gleason Works, 1000 University Ave., Rochester, N.Y.
Gleeson, John M. ('19; '26; '35) (C), Asst. Mgr., Westinghouse X-Ray Div., Westinghouse Elec. Corp., 21-16—43rd Ave., Long Island City; *for mail,* 2 Willets Court, Plandome, L.I., N.Y.
Gleeson, Mortimer Joseph (J'43) (SFB), Lt., U.S.C.G.R., 311 Custom House, New Orleans, La.
Gleeson, William Savage ('24; '30; '35) (CJM), Staff Engr., Am. Mch. & Fdy. Co., 511—5th Ave., New York 17, N.Y.
Glenn, Edward Robins ('41) (JM), Mfrs. Agent, 3701 N. Broad St., Philadelphia 40, Pa.
Glennie, George W. ('25; '31; '35) (SECAFH), Asst. Supt., Boiler & Mchy. Dept., Employers' Gp. Ins. Co., 110 Milk St., Boston 9; *for mail,* 21 Wolcott Ave., Andover, Mass.
Glennon, Miss Nan (J'45) (KHSEC), Engr., Designs Div., El Segundo Refinery, Stand. Oil Co. of Calif., El Segundo; *for mail,* 830 N. Orlando Ave., Los Angeles 46, Calif.
Glick, Ashton Eugene (J'33) (NJMOBD), Asst. Ch. Engr., Charge Engrg. Dept., Broderick & Bascom Rope Co., 4203 N. Union Blvd., St. Louis 15; *for mail,* 6559 Mardel Ave., St. Louis 9, Mo.
Glidewell, James Elvan ('44) (NMCBJK), Asst. Ch. Engr., Hall Scott Motor Car Co., 2850—7th St., Berkeley 2; *for mail,* 1855 San Antonio Ave., Berkeley 7, Calif.
Glimm, William Frederick ('17; '26; '35) (CDFHKS), Plant Supt., Barrett Div., Allied Chem. & Dye Corp., 86 Elizabeth Ave., Elizabeth B, N.J.
Glinske, Edward Joseph, Jr. ('46) (BCS), Head, Specialties Sales Dept., Cochrane Corp., 17th St., Philadelphia 32; *for mail,* H-1, Markham Apts., 3213-19 N. 17th St., Philadelphia 40, Pa.
Gliss, George E. (J'40) (YDNMC), Devel. Engr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore 24, Md.
Gloor, Wilbur Tell (J'40) (NZHBCJ), Research Engr., Motch & Merryweather Mchy. Co., 713 Penton Bldg., Cleveland 13; *for mail,* 1504 E. 256th St., Euclid 17, Ohio.
Gloss, Eric A. (J'27) (CLNYKT), Ch. Engr., Irvington Varnish & Insulator Co., 6 Argyle Terrace, Irvington, N.J.
Glossa, Aloysius J. (A'19) (C), Head, Constr. Maint. Sec., Boston Edison Co., 182 Tremont St.; *for mail,* Suite 14, Fenmore Apts., 1109 Boylston St., Boston, Mass.
Glover, James Bolan ('28; '38) (CJM), Maj., Coast Arty. (AA); *for mail,* Box 85, Marietta, Ga.
Gluckman, Isidore B. (J'40) (BCE), USAFI Sec., I. & E. Serv. Hdq. USFET, A.P.O. 757, c/o Postmaster, New York, N.Y.
Glueck, Frank J. ('30) (MCDLJH), Mech. Engr., Ballinger Co., 105 S. 12th St., Philadelphia; *for mail,* 1504 E. Cliveden St., Philadelphia 19, Pa.
Glunz, Wm. H. (J'38), 190 Claremont Ave., New York, N.Y.
Gmitro, Arthur F. (J'41) (COMD), Lt. (j.g.), U.S.N.R., Res. Inspr. of Naval Matl., U.S.N., Estes St., Ipswich, Mass.; *for mail,* 510 E. 83rd St., New York 28, N.Y.
Gnade, Edward Richard ('95; '05) (EA), Retired; 128 Wyllis St., Oil City, Pa.
Goben, Stanley James (J'44), Cost Accountant, Am. Potash & Chem. Corp.; *for mail,* Box 15, Trona, Calif.
Goble, George H. (J'42) (AJC), Mech. Design Engr., Radio Corp. of Am., Camden; *for mail,* 220—9th Ave., Haddon Heights, N.J.
Godbey, John Walter (J'42) (KLYSNG), Mech. Engr., Designer, Votator Div., Girdler Corp., 224 E. Broadway, Louisville 2; *for mail,* 1146 Lincoln Ave., Louisville 8, Ky.
Goddard, Archibald N. ('13) (JKM), Pres., Goddard & Goddard Co., 12280 Burt Rd.; *for mail,* 630 Virginia Pk., Detroit 2, Mich.
Goddard, Warren Blake (J'40) (BJAKNC), Engr., Aircraft Gas Turbine Div., Gen. Elec. Co., 920 Western Ave., Lynn 3; *for mail,* 282 Puritan Rd., Swampscott, Mass.
Godeke, H. F. ('35) (CEK), Head, Mech. Engrg. Dept., Tex. Tech. College, Lubbock, Tex.
Godeke, Henry Lawrence (J'39) (EFKOS), Mech. Engr., Stanley Engrg. Co., Hershey Bldg., Muscatine, Iowa.
Godfrey, Ralph ('24; '30; '35) (BJM), Sr. Mech. Engr., Naval Shipyard, Philadelphia; *for mail,* 318 Walnut Pl., Havertown, Pa.
Godfrey, Walter ('45) (WCYMO), Pres., Gen. Mgr., River Edge Industries Corp., 5 River Edge Rd., River Edge, N.J.; *for mail,* 80-59—210th St., Queens Village 9, N.Y.
Godfrey, William G. ('34; '36) (SFKCMB), Asst. to Supt., C. R. Huntley Stas., Buffalo Niagara Elec. Corp., Elec. Bldg., Buffalo 3; *for mail,* 235 Hartwell Rd., Buffalo 16, N.Y.
Godley, Edgar Bartlett ('44) (CMZOA), Sr. Product Engr., Naval Ord. Plant, 21st & Arlington, Indianapolis; *for mail,* 5026 Central Ave., Indianapolis 5, Ind.
Godschall, Milton G. (J'27) (CDM), Mech. Engr., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 11 Piedmont St., East Lynn, Mass.
Goe, Jack H. (J'41) (EDAR), 710 Hickory St., Anaconda, Mont.
Goebel, Gordon W. ('22; '23) (CMS), c/o Sam I. Bousman, West. Mchy. Co., 760 Folsom St., San Francisco, Calif.
Goebel, Rudolph Conrad ('42) (RCMS), Gen. M.M., Minneapolis & St. Louis Ry. Co., Minneapolis 5; *for mail,* 1563 Fulham St., St. Paul 8, Minn.
Goedecke, Mark (J'35) (CMB), Philadelphia Rep., New Departure Div., Gen. Motors Corp., 1624 Hunting Pk. Ave., Philadelphia 40, Pa.
Goehring, Walter William (J'34) (CMNAB), Sales Engr., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia; *for mail,* 1831 Beverly Rd., Philadelphia 38, Pa.
Goelkel, Walter Leo William (J'44), c/o Calvert Distilling Co., Relay, Md.
Goeltz, Philip Henry (J'29) (NBJH), Stands. Engr., Gleason Works, 1000 University Ave., Rochester 3; *for mail,* 86 Meadowbrook Rd., Rochester 10, N.Y.
Goelz, Arnold H. ('14; '35) (CKO), Pres., Kroeschell Engrg. Co., 215 W. Ontario St., Chicago 10, Ill.
Goentner, William B. ('05; '14) (JLH), Mech. Engr., 221 W. Mt. Carmel Ave., Glenside, Pa.
Goepfert, Otto Frederick ('38) (CDMNT), Ch. Engr., Champagne Paper Corp., Endless Belt Corp., Pisgah Forest; *for mail,* Route 2, Brevard, N.C.
Goerg, Bernard ('25; '31) (FKS), Am. Radiator & Stand. Sanitary Corp., 40 W. 40th St., New York 18; *for mail,* Box 67, Bronxville 8, N.Y.
Goerg, Emil F. (A'44) (BCMNJD), Supt. Mfg. Dept., Air Reduction Co., Inc., 125 Monitor St., Jersey City 4; *for mail,* 400 S. Washington Ave., Dunellen, N.J.
Goerky, Charles M. (J'41) (AHM), Jr. Engr., Bendix Aviation Corp.; *for mail,* 1007 E. Washington Ave., South Bend, Ind.
Goerlitz, Joseph Julius (J'41) (AFE), Mech. Engr., Air Tech. Serv. Comd., Wright Field, Dayton; *for mail,* 1208 Wayne Ave., Dayton 10, Ohio.
Goetz, Frank L., Jr. (J'39), 176-36—88th Ave., Jamaica, N.Y.
Goetz, Harold E. ('20; '26; '35) (BES), Designing & Testing Engr., Skinner Eng. Co., 337 W. 12th St.; *for mail,* 3927 Myrtle St., Erie, Pa.
Goetz, John H. (J'37) (NCM), Pres., Goetz-Voss Corp., 6242 W. State, Milwaukee 13; *for mail,* 6930 Cedar St., Wauwatosa 13, Wis.
Goetze, Fred'k A. ('95; '00), Treas., Columbia Univ., 76 William St., New York, N.Y.
Goetzenberger, Ralph Leon ('16; '20; '29) (CZL), V.P., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.; *for mail,* 1704 Hillcrest Rd., Chestnut Hill, Philadelphia, Pa.
Goff, John Alonzo ('29; '35) (BEHK), Dean, Towne Scien. Sch., Univ. of Pa., Philadelphia 4, Pa.
Goga, George Frank (J'42) (DNHBZ), Research Engr., Columbus McKinnon Chain Corp., Fremont St., Tonawanda; *for mail,* 75 Enola Ave., Kenmore 17, N.Y.
Goglia, Gennaro Louis (J'45) (JK), Test Engr., Gen. Elec. Co., Broadway; *for mail,* 917 Garden St., Ft. Wayne, Ind.
Goglia, Mario Joseph (J'40) (SZKL), Application Engr., Republic Flow Meter Co., 2240 Diversey Blvd., Chicago 47; *for mail,* 3842 Irving Park Rd., Chicago 18, Ill.
Goheen, Richard Warner (J'41) (SONC), Asst. Test Engr., Burlington Generating Sta., Pub. Serv. Elec. & Gas Co. of N.J., Burlington; *for mail,* 52 Church St., Beverly, N.J.
Goin, Newbold Crittenden ('46) (ABC), Exec. Secy., American Gear Manufacturers Association, 301 Empire Bldg., Pittsburgh 22; *for mail,* 79 Summer Ave., Bellevue, Pittsburgh 2, Pa.
Gokey, Franklyn Charles (J'42) (OMC), 4862—28th St. S., Arlington, Va.
Goland, Martin (J'42) (BAHEWZ), *Spirit of Saint Louis Junior Award,* '44; Chmn., Engrg. Mechanics Research, Midwest Research Inst., 4049 Pennsylvania, Kansas City 2, Mo.
Gold, David (J'36) (BZJKAY), Bur. of Ord., Navy Dept., Washington, D.C.; *for mail,* 3835 N. Pennsylvania St., Indianapolis 5, Ind.
Gold, David Horace (J'45) (KOMZ), Test Engr., Devel. Dept., Carrier Corp., 300 S. Geddes St., Syracuse; *for mail,* 122 Clifton Pl., Syracuse 6, N.Y.
Goldberg, Bernard Wilcox (J'42) (ANMEZY), Mech. Draftsman, Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 25 Parkview Pl., Passaic, N.J.
Goldberg, Charles K. ('45) (HJO), Owner & Ch. Engr., Chicago Mech. Engrg. Co., Rm. 308, 111 W. Jackson Blvd., Chicago 4, Ill.
Goldberg, Herman ('35) (BCM), Owner, Snow Mfg. Co., 615 S. California Ave., Chicago 12, Ill.
Goldberg, Julian N. (J'42) (ABCH), 2057 White St., Alexandria, La.
Goldberg, Maxmilian Meier ('21) (BGL), Retired; Biltmore Hotel, Dayton 2, Ohio.
Goldberger, Earl Samuel (J'44) (MJC), 1403 Rust St., Eau Claire, Wis.
Goldberger, Robert Seymour (J'45) (EHSKCA), Pvt., Army Air Forces, 3704th Base Unit, Flight 389, Sqd. Z, Keesler Field, Miss.
Golden, Gene E. (J'38) (CHM), Pvt., H. & S. Co., 319th Engr. Bn., A.P.O. 94, Camp Phillips, Kan.
Golden, Robert F. ('30) (JDB), Asst. to Pres., Natl. Lock Washer Co., 40 Hermon St., Newark 5; *for mail,* 67 Fielding Court, South Orange, N.J.
Golden, Thomas Richard (J'43) (ESMN), 31 Ocean St., Lynn, Mass.
Golder, John D. ('46) (CMO), Ch. Indus. Engr., Ohio Injector Co.; *for mail,* 198 Broad St., Wadsworth, Ohio.
Goldey, John Stuart (J'44) (MCD), Indus. Engr., Eastman Kodak Co., Rochester 4; *for mail,* 45 Pleasant Ave., Point Pleasant, N.Y.
Goldgraber, Howard D. (J'43) (ACFBYK), Ensign, U.S.N.R., Office of Research & Invention, Navy Dept., Washington, D.C.; *for mail,* 24-75—38th St., Long Island City 3, N.Y.
Goldie, Alexander Rodgers ('02; '21) (CDEJNZ), Chmn. of Bd., Babcock Wilcox—Goldie-McCulloch Co., Grand Ave. S., Galt, Ont., Can.
Golding, Harold B. ('29; '35) (JMBL), V.P., Plant Engr., Internatl. Equip. Co., 352 Western Ave., Brighton 35; *for mail,* 48 Massachusetts Ave., Dedham, Mass.
Goldman, Jack (J'45) (SMO), Power Plant Engr., Gruenwald & Niemitz, 44-27 Purvis St., Long Island City; *for mail,* 650 Ocean Ave., Brooklyn 26, N.Y.
Goldman, Joshua Harold ('42), Pres., Fibre Products Labs., Inc., 51 Washington St., New Brunswick, N.J.
Goldman, Oscar G. ('19; '35) (H), Asst. Supt., San Francisco Water Dept., 425 Mason St., San Francisco, Calif.
Goldreyer, Alfred (J'39) (EHKS), Secy., Job Supt., Pipe & Engrg. Co., Inc., 531 Coster St., New York 59, N.Y.
Goldrosen, Louis (J'42) (MCJYA), Ensign, U.S.N.; *for mail,* 89 Richmond Ave., Worcester 2, Mass.
Goldsberry, Loyd Woodrow (J'41) (MSCD), Mech. Engr., Plant Engrg. Dept., Nash-Kelvinator Corp., Kenosha, Wis.
Goldsbury, John ('36) (ESFKBA), Mech. Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 74 Fairview Rd., South Lynnfield, Mass.
Goldsmith, Clarence ('11) (HSO), Asst. Ch. Engr., Natl. Bd. of Fire Underwriters, 222 W. Adams St., Chicago 6, Ill.
Goldsmith, Clarence Stoddard ('42) (FECBMN), Engr. of Distribution, Brooklyn Union Gas Co., 176 Remsen St., Brooklyn 2, N.Y.
Goldsmith, Lester Morris ('17; '21; '35) (ESFOLZ), *Melville Medallist,* '39; Ch. Engr., Atlantic Refining Co., 260 S. Broad St., Philadelphia 1, Pa.
Goldsmith, Philip H. (M'39) (WHKC), Sales & Devel. Engr., Pusey & Jones Corp., Front & Poplar Sts., Wilmington; *for mail,* 120 Alapocas Dr., Wilmington 284, Del.
Goldstein, David Lester (J'43) (HKESO), Pvt., 1st Class, Ord. Dept., Aberdeen Proving Ground, Md.; *for mail,* 3227 Balboa St., San Francisco 21, Calif.
Goldstein, Irving R. (J'39) (ACM), 164 Bergen Ave., Jersey City, N.J.
Goldsworth, Elmer C. (J'41) (SEARZ), Field Engr., Gen. Elec. Co., 570 Lexington Ave., New York 22; *for mail,* 77 Church Ave., Islip, L.I., N.Y.
Golem, George Charles William ('44) (SEFK), Mgr., Power Dept., Atlantic Wire Co., 1 Church St.; *for mail,* Box 373, Sybil Ave., Indian Neck, Branford, Conn.
Goller, George N. (J'44) (JEARZS), Supvg. Metallurgist, Rustless Iron & Steel Div., Am. Rolling Mill Co., 3400 E. Chase St., Baltimore 13, Md.
Goller, John Rudolph, Jr. (J'43) (LAH), Engr., A. M. Lockett & Co., Ltd., 308 Whitney Bank Bldg., New Orleans; *for mail,* 2465 N. Galvez St., New Orleans 17, La.
Gollin, Charles M. (J'33) (CKLM), Mech. Engr., Designer, Puget Sound Naval Shipyard; *for mail,* 1008 Parkside Dr., Bremerton, Wash.
Gollin, Sylvan (J'42) (CLS), Ensign, U.S.N.R., 441 Ocean Ave., Brooklyn, N.Y.
Gollmer, Carl Edward (J'35) (A), Supvr., Eng. Checking Gp., Consltd. Vultee Aircraft Corp., San Diego; *for mail,* 2316 Bancroft St., San Diego 2, Calif.
Gollmer, Hugo C. ('39) (CMS), Mech. Engr., Mech. Div., Bd. of Water Supply, City of N.Y., Rm. 1401, 120 Wall St., New York, N.Y.
Golner, David (J'43) (LZJSHA), Design & Devel. Engr., Hammel-Dahl Co., 243 Richmond St., Providence; *for mail,* 243 Camp St., Providence 6, R.I.
Goloff, Michael John ('44) (CDGHMN), Mech. Supt., Plant Engr., Consltd. Paper Corp.; *for mail,* 406—4th Ave., Port Alfred, Que., Can.
Golom, John Peter ('44) (CLNZDJ), Ch. Engr., Burroughs-Wellcome Co. (U.S.A.), Inc., 1 Scarsdale Rd., Tuckahoe, N.Y.; *for mail,* 60 New St., East Orange, N.J.
Golrick, Mark Augustine, Jr. ('23; '31) (FEKSMO), Gen. Coupling Co., 120 Broadway, New York; *for mail,* 6 Kingston Ave., Poughkeepsie, N.Y.
Golwelkar, Wasudeo Govind (J'45) (FJO), R.B.M. Poona Mills, Ltd., Poona, India.
Gomberg, William ('40; '43; '44) (CT), Ch. Indus. Engr., Internatl. Ladies Garment Workers Union, 1710 Broadway, New York, N.Y.
Gomes, Anthony Alfred (J'42) (AMK), 131—2nd St., Cambridge, Mass.
Gompf, Arthur M. ('34; '41) (OKS), Egli & Gompf, Inc., 425 St. Paul Pl., Baltimore 2; *for mail,* Box 337, Pikesville 8, Md.

Gonder, Warren W. (J'41) (CNHKLM), Engr., Design & Constr., Bur. of Reclamation, U.S. Dept. of the Interior, Rm. 332, Custom House, Denver, Colo.
Gongwer, Burr Franklin (A'42), V.P., Firemen's Mutual Ins. Co., Providence, R.I.; *for mail,* 10 Reimer Rd., Scarsdale, N.Y.
Gongwer, Calvin Andrew (J'42) (HABSZR), Project Engr., Research Dept., Aerojet Engrg. Corp.; *for mail,* 1126 Orange Ave., Azusa, Calif.
Gonzalez, Armando (J'44) (CHSW), Sales Engr., Otis, McAllister & Co., 310 Sansome St., San Francisco 4, Calif.; *for mail,* Diriamba, Nicaragua, C.A.
Gonzalez, Eduardo Damaso ('31; '38) (BEKLMS), Ch. Engr., Am. Sugar Refining Co., Central Jaronu, Camagüey, Cuba.
Gonzalez, Miguel Angel (J'41) (HKS), Mech. Engr., East. Sugar Associates; *for mail,* Central Santa Juana, Caguas, P.R.
Gooch, Fred P. (J'42), Mech. Calculations Gp., Westinghouse Elec. Corp., South Philadelphia; *for mail,* 609 Elm Ave., Swarthmore, Pa.
Good, Charles Winfred ('20; '25) (ABEKNW), Asst. Dir., Dept. of Engrg. Research, Prof. Mech. Engrg., Univ. of Mich., Ann Arbor, Mich.
Good, Harry Joseph (J'42) (ACE), Inspr., Assembly Dept., Plant 1, Wright Aero. Corp., Paterson; *for mail,* 427 Franklin St., Elizabeth, N.J.
Good, John F. (J'39), 343 Cleveland St., Akron, Ohio.
Good, Paul Eckert ('15), Retired; 407 Linden Ave., Riverton, N.J.
Good, Sam (J'42) (HMRONB), Bethlehem Steel Co., Sparrows Point; *for mail,* 2940 Sollers Point Rd., Baltimore 22, Md.
Good, William Eckert (J'41) (ABM), Lt., 3rd Military R.R. System, A.P.O. 404, c/o Postmaster, San Francisco, Calif.; *for mail,* 407 Linden Ave., Riverton, N.J.
Goodale, Francis ('33; '35) (BDFKLS), Asst. Engr., McCahan Sugar Refinery, Foot of Tasker St., Philadelphia 48, Pa.; *for mail,* 638 Benson St., Camden 3, N.J.
Goodall, Ray Gilbert (J'36) (ABC), Capt., U.S.A., c/o H. A. Goodall, 5416 Arbor Ave., Oakland 9, Calif.
Goode, Curtis B. ('20) (DHW), Mgr., Dominica Farm & Ind. Corp.; *for mail,* La Dominica, Turrialba, Costa Rica, C.A.
Goodell, Richard A. (J'42), 2nd Lt., Coast Arty., U.S.A., 308th Balloon Barrage Bn., Seattle, Wash.; *for mail,* Winchester, N.H.
Gooden, Maurice Price ('38; '46) (CLT), Asst. to Plant Engr., Am. Viscose Corp.; *for mail,* 20 Franklin Ave., Lewistown, Pa.
Goodfriend, Nathan (J'43) Ensign, U.S.N.R., 809 S. Bundy Dr., West Los Angeles 24, Calif.
Goodier, Jas. N. ('37) (ABH), Prof. Mechanics, Cornell Univ.; *for mail,* 109 Cayuga Heights Rd., Ithaca, N.Y.
Goodin, Albert Brownfield ('45) (BKS), Staff Technician, So. Calif. Gas Co., 509 Aliso St., Los Angeles; *for mail,* 901 S. Ardmore, Los Angeles 6, Calif.
Goodison, Louis Emory (J'45) (JKLS), Mech. Engr., J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6; *for mail,* Box 2421, 819 E. 83rd St., Kansas City 5, Mo.
Goodman, Edwin Francis (J'41) (CMT), Lt. (j.g.), PT-25, F.P.O., San Francisco, Calif.
Goodnow, John M. (J'23) (NWHMB), Pres., Hardware Products Co., Inc., 103 Richmond St., Boston 13; *for mail,* Cornet Stetson Rd., Greenbush, Mass.
Goodrich, Charles W. McK. ('19; '24; '29) (BCDEHM), Mech. Engr., Charge Equip. Devel. & Design, Carborundum Co.; *for mail,* 1517 Niagara Ave., Niagara Falls, N.Y.
Goodrich, Thomas MacLenathen ('25; '35) (EF), Sr. Gas Engr., Pub. Serv. Comm., State of N.Y., 233 Broadway, New York 7, N.Y.
Goodwill, Arthur L. (J'41) (EFS), Serv. Dept., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Goodwin, Benjamin Schoppaul (J'45) (BNZ), Maj., Ord. Dept., U.S.A., Chief, Arty. & Rocket Branch, Ord. Research & Devel. Center, Aberdeen Proving Ground, Md.
Goodwin, Curtis Leslie (J'37) (BWK), Mch. Design Engr., Merritt Engrg. & Sales Co., Inc., 120 S. Niagara St.; *for mail,* 23 Erie St., Lockport, N.Y.
Goodwin, Eugene Wiley ('31) (ERS), Mech. Engr., Pub. Bldgs. Admin., Washington 25, D.C.; *for mail,* 7024 Hampden Lane, Bethesda 14, Md.
Goodwin, George L. (J'37) (BEHJMN), Eng. Specialist, Worthington Pump & Mchy. Corp., 400 W. Madison St., Chicago, Ill.
Goodwin, Harold, Jr. ('24) (CFSEKO), Cons. Engr., Wyncote, Pa.
Goodwin, Louis (J'41) (YMLCA), Jr. Engr., E. I. du Pont de Nemours & Co., Arlington; *for mail,* 745 West Side Ave., Jersey City, N.J.
Goodwin, Richer Mathieu (J'32) (EHJLS), Capt., U.S.A.; 4806 Springfield Ave., Philadelphia, Pa.
Goodwin, Walter Hardenbroecke (J'44) (BJS), Graduate Student, Calif. Inst. of Tech., Pasadena 4; *for mail,* 330 Lola Ave., Pasadena 8, Calif.
Goodyear, Howard Russell (J'37) (RKS), Capt., Field Arty., 736th Field Arty. Bn., Ft. Ord, Calif.
Goodzey, John Robert (J'40) (CJL), Engr., Anaconda Wire & Cable Co., California St., Sycamore 1; *for mail,* 2649 Haddon Ave., Chicago, Ill.
Goral, Edwin Bernard (J'44) (AKMNEJ), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail,* 53-C Elizabeth Rd., Hampton, Va.
Gorbaty, Arthur Mitchell (J'44) (BDM), 506—4th St., Niagara Falls, N.Y.
Gorder, Cyrus Garfield ('44), Asst. Boiler Rm. Engr., Commonwealth Edison Co., 3501 S. Pulaski Rd., Chicago 23, Ill.
Gordon, Allan Plummer (J'42) (CLMOS), 1115 Minnesota Ave., San Jose, Calif.
Gordon, Arthur (J'43) (NYMS), Technician, 5th Grade, 2881st Engrs., Army Ground Forces, A.P.O. 88, c/o Postmaster, San Francisco, Calif.; *for mail,* 216—9th Ave., Belmar, N.J.
Gordon, Charles William ('21; '25; '35) (KLFSRB), Ch. Engr., Drying Dept., Combustion Engrg. Co., Inc., 1319 N. Branch St., Chicago 22; *for mail,* 535 Oak St., Glen Ellyn, Ill.
Gordon, David ('27; '43) (LOFCKJ), Exec. V.P., David Gordon & Co., Inc., 29 Broadway, New York 6, N.Y.
Gordon, Edward David (J'41) (YNMJC), Staff Engr., Mech. Rubber Goods, Firestone Indus. Products, Firestone Pkwy., Akron 1; *for mail,* 890 Stadelman Ave., Akron 2, Ohio.
Gordon, Elliott Morton ('43) (CMJFGK), Exec. V.P., Gorham Mfg. Co., 333 Adelaide Ave., Providence 7, R.I.
Gordon, Eugene ('41) (SCFEH), Supt. of Power, Potomac Edison Co., 55 E. Washington St.; *for mail,* 528 Summit Ave., Hagerstown, Md.
Gordon, Hayden S. (J'41) (KAO), Engr., Radiation Lab., Univ. of Calif.; *for mail,* 2545 A Benvenue Ave., Berkeley 4, Calif.
Gordon, Henry Robert (J'41) (ABJ), 1st Lt., Corps of Engrs., 813th Engr. Aviation Bn., A.P.O. 986, Seattle, Wash.
Gordon, Myron B. ('20; '25) (ACMSJE), V.P., Charge of Operas., Fairchild Eng. & Airplane Corp., 30 Rockefeller Plaza, New York 20, N.Y.
Gordon, Peter Benjamin ('43) (SKLBZ), Treas., Engr., Wolff & Munier, Inc., 222 E. 41st St., New York, N.Y.
Gordon, Robert J. ('18; '26) (FSDKLM), Plant Engr., Walter Baker & Co., Inc., 1197 Washington St., Dorchester 24, Mass.
Gordon, Sidney (J'40) (BNAM), Gp. Leader, Mech. Design, Edo Aircraft Corp., College Point, L.I.; *for mail,* 187 Hicks St. (5-B), Brooklyn 2, N.Y.
Gordon, Vincent Joseph, Jr. (J'42) (MCSA), Jr. Mfg. Engr., Steam Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 718 Morgan Ave., Drexel Hill, Pa.
Gordy, George Baxter, Jr. ('44) (M), Mech. Engr., Arabian Am. Oil Co., 200 Bush St., San Francisco; *for mail,* 83 Pleasant St., San Francisco 8, Calif.
Gore, John Calvin (J'40) (SKCFEM), Power Engr., Post Products Div., Gen. Foods Corp., Battle Creek; *for mail,* 427 High St., Marshall, Mich.
Gore, Linn A. (J'34) (BAE), Dunlap, Kan.
Gore, Robert Thomas (J'45) (JKBZLF), Raw Matls. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* Kings Highway, Middletown, N.J.
Gorham, John Milton ('42), V.P., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford 2, Conn.
Gorlin, Boris (J'43) (MAB), Ch. Engr., Prod. Mgr., Burkaw Elec. Co., 105 E. 29th St., New York 16, N.Y.
Gorman, Joseph H. (J'39) (FKM), Motor Machinist's Mate, 3rd Class, Instr., Naval Trng. Sch., Richmond, Va.
Gormel, Ernest M. (J'32) (MNCKFE), Mech. Engr., Delco Appliance Div., Gen. Motors Corp., 391 Lyell Ave., Rochester; *for mail,* 260 Lincoln Ave., Rochester 11, N.Y.
Gormeley, James Francis (J'44), Pvt., 1st Class, Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.; *for mail,* 570 Jersey Ave., Jersey City, N.J.
Gormley, John Joseph, Jr. (J'46) (KOS), Apt. 22, Barcroft Apts., 1303 S. Thomas St., Arlington, Va.
Gorrell, C. Woodrow (J'38), 146—9th Ave., South Charleston, W. Va.
Gorrie, Harvard H. (J'27) (BCJMZ), Design Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 12; *for mail,* 1155 Erieview Rd., Cleveland 21, Ohio.
Gorrie, John M. ('39) (CES), 356 Fuller Ave., S.E., Grand Rapids 6, Mich.
Gorton, Charles E. ('15; F'36), *Manager,* '25-'28, *Vice-President,* '28-'30; Chmn., Uniform Boiler Legislation, Am. Uniform Boiler Law Soc., 95 Liberty St., New York 6, N.Y.
Goryainoff, Alexis Nikolas (J'44) (BHK), Internatl. Gen. Elec. Co., Schenectady, N.Y.
Goryl, William Munder (J'46), 165 Hillside Ave., Hillside, N.J.
Goss, Robert J. (J'42) (LJMOSD), Mech. Engr., Maint., Am. Cyanamid Co., Warners St., Linden; *for mail,* 64 Pine Grove Terrace, Newark, N.J.
Goss, Stanley T. ('14), Pres., Goss & DeLeeuw Mch. Co., New Britain, Conn.
Gossman, Angelo Ludwig ('21; '25; '35), West Coast Mgr., Tile-Tex Co., 530 W. 6th St., Los Angeles 14; *for mail,* 1424 E. Mountain St., Glendale 7, Calif.
Gosting, Richard Ernest (J'42), Ensign, U.S.N.R., Kildare, Okla.
Gothberg, Edwin G. ('37; '40) (SEFCHK), Asst. Engr., Gas & Steam Dept., Pacific Gas & Elec. Co., 245 Market St., San Francisco 6; *for mail,* 2070—16th Ave., San Francisco 16, Calif.
Gotherman, Charles Warren ('42) (LDJKFO), Mech. Engr., N.J. Zinc Co., of Pa.; *for mail,* Residence Park, Palmerton, Pa.
Goto, Lawrence Harvey (J'45) (BNSMCH), Mech. Design Engr., Gen. Am. Trans. Corp., 209 W. Jackson Blvd., Chicago; *for mail,* 2913 N. Mulligan Ave., Chicago 34, Ill.
Gottesman, Arthur H. (J'44) (BZMCNA), Design Engr., Farrand Optical Co., Inc., 4401 Bronx Blvd., New York 66; *for mail,* 2200 Grand Ave., New York 53, N.Y.
Gottilla, Salvatore Charles (J'45) (BSYZCJ), 1779—46th St., Brooklyn 4, N.Y.
Gottlieb, Edward ('21; '27; '35), (CGO), Pat. Atty., 80 Wall St., New York 5, N.Y.
Gottsch, Donald Chris (J'42) (ASWOCF), Aviation Instr., Pilot, Bay Shore High Sch., 3rd Ave., Bay Shore; *for mail,* 74 W. Islip Rd., Babylon, L.I., N.Y.
Gotwals, Charles Scheetz ('20; '26) (CJM), Supt., Plant 1, SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34; *for mail,* 107 Cypress Ave., Jenkintown, Pa.
Gough, A. C. ('18) (GJM), Dir., Div. of Engrg., So. Branch, Univ. of Idaho, Pocatello, Idaho.
Gough, John Bernard ('24) (BHKDSL), Ch. Engr., Mead Corp., S. Paint St.; *for mail,* Box 93, Chillicothe, Ohio.
Gould, George Douglas (A'45) (GJCM), Engr., Chemco, Inc., New St., Glen Cove; *for mail,* Hickory Hill, Roslyn, L.I., N.Y.
Gould, Gerald B. ('31) (FSOE), Pres., Fuel Engrg. Co. of N.Y., Partner, Gould, Henderson & Brunjes, 215—4th Ave., New York 3, N.Y.
Gould, Jack U. ('46), Mech. Engr., Ferdinand Gutmann & Co., 3611—14th Ave., Brooklyn 18, N.Y.
Gould, Leo J. (J'28) (BNMFDH), Asst. Ch. Engr. of Constr., Bethlehem Steel Co.; *for mail,* 1803 Easton Ave., Bethlehem, Pa.
Gould, Norman J. ('00; '08), Pres., Goulds Pumps, Inc., Seneca Falls, N.Y.
Gould, Ralph L. ('44) (BCDJMN), Dir. Basic Research, Natl. Can Corp., 179 Hamilton Ave., Greenwich, Conn.; *for mail,* 164 Maple St., Springfield 5, Mass.
Gould, William Richard (J'42) (SNCH), Lt. (j.g.), U.S.N.R., Mchy. Inspc. Officer, c/o Supvr. of Shipbldg., U.S.N., San Pedro; *for mail,* 440 Coronado Ave., Long Beach 4, Calif.
Govan, John Hammond (J'36) (HLKJMD), Branch Mgr., Sales Engr., Can. Blower & Forge Co. Ltd., 1221 Bay St., Toronto 5, Ont., Can.
Gove, Norris Dickinson ('42), Engr., Westinghouse Elec. Corp., Lester; *for mail,* 847—11th Ave., Prospect Park, Pa.
Gow, Ralph Frederick ('26; '33; '35) (C), Works Mgr., Norton Co., 1 New Bond St., Worcester 6, Mass.
Gowdy, Allen Clair (J'43), 308 Standard Bldg., Atlanta 3, Ga.
Gowen, Raymond B. (J'40) (AJM), Lt., O-V(S), U.S.N.R., Naval Air Sta., Norfolk, Va.; *for mail,* 3619 Bowne St., Flushing, N.Y.
Gowling, Lawrence E. ('20; '35) (FKS), Mech. Engr., Compania Cubana de Electricidad, Box 1715, Monte 1, Havana, Cuba.
Graboski, Leo D. ('40) (EMR), Asst. Ch. Engr., Vulcan Iron Works, Box 570; *for mail,* 75 Graham Ave., Lee Park, Wilkes-Barre, Pa.
Grabowski, Edward John (J'44) (KLP), Chem. Engr., Hydrocarbon Research, Inc., 115 Broadway, New York; *for mail,* 2140 Chatterton Ave., New York 61, N.Y.
Grace, Charles Theron ('36; '45) (NBEJSO), Assoc. Prof. Mech. Engrg., Engrg. Hall, Univ. of New Mex., Albuquerque, New Mex.
Grace, John Francis ('15) (SKR), Condenser Engr., Worthington Pump & Mchy. Corp., 401 Worthington Ave., Harrison; *for mail,* 24 Alpine Pl., Arlington, N.J.
Grace, William A. ('27) (CEFKSY), Supt. of Maint., J. H. Williams & Co., 400 Vulcan St.; *for mail,* 251 Woodlawn Ave., Buffalo, N.Y.
Gracik, John Walter (J'38) (CAODHM), 1st Lt., Ord. Dept., Army Serv. Forces, Property Disposal Officer, Detroit Ord. Dist., 1832 Natl. Bank Bldg., Detroit 32, Mich.
Gradisar, Albin A. (J'39) (ABC), Devel. Engr., Spencer Lens Co., 19 Doat St.; *for mail,* 19 Oakmont Ave., Buffalo 15, N.Y.
Gradisar, Ivan A. (J'38) (ACJLM), Asst. Maint. Supvr., Columbia Chem. Co.; *for mail,* 40—14th St., Barberton, Ohio.
Grady, Charles B. ('14) (BDJKOS), Retired; Box 96, Skytop, Pa.
Grady, Edward Glennan ('41) (SWZMKC), Plant Engr., Masonite Corp., S. 4th Ave., Laurel, Miss.
Graef, Louis Fred ('18; '26; '35) (BDEH), Ch. Maint. Man, Rancho Los Amigos, Hondo, Calif.
Graesser, Carl H. ('38), V.P., Charge of Engrg. & Devel., Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Conn.
Graesser, Elmer C. (J'37) (CJMNY), Plant Supt., Natl. Vulcanized Fibre Co., Maryland Ave. & Beech St., Wilmington 99, Del.
Graf, Frederick John ('31; '41) (CMZ), Asst. V.P., & Ch. Engr., Mass. Bonding & Ins. Co., 85 Water St., Boston 9; *for mail,* Doublet Hill Rd., Weston 93, Mass.

Graf, John C. ('13; '18; '35) (HYWK), Sales Mgr., Hyd. & Spec. Mchy. Div., William Sellers & Co., Inc., 1600 Hamilton St., Philadelphia 30; *for mail,* 705 E. Longshore Ave., Philadelphia 11, Pa.
Graf, John Richard (J'37) (NMJBHC), Mgr. Procurement, Stand. Vendors Div., Lehigh Fdys., Inc., Easton; *for mail,* 123 Roseland Ave., Philadelphia 11, Pa.
Graf, Julius Eicher ('39) (JBCM), Asst. Ch. Engr., Jones & Laughlin Steel Corp., 3rd Ave. & Ross St., Pittsburgh 30; *for mail,* 609 Perrysville Rd., West View 2, Pa.
Graf, Samuel Herman ('12; '20) (BJZO), *Manager, '43-'45, Director-at-Large, '45-'46;* Dir., Engrg. Exper. Sta., Ore. State College, Corvallis, Ore.
Graf, William, Jr. (J'29) (KLSCWR), 96 Summit Ave., Upper Montclair, N.J.
Graff, Paul Taylor ('44), Engr., Groton, Mass.
Graff, Truls Wiel (J'43) (BJKSHA), Elliott Co., Jeannette, Pa.
Graffeo, Alphonse J. (J'40) (BKS), 25 College Ave., Medford 55, Mass.
Graham, David P. (J'24), Engrg., Sales Devel., Peabody Engrg. Corp., 580—5th Ave., New York; *for mail,* 536 Livingston St., Westbury, L.I., N.Y.
Graham, Frederick C. (J'41) (MJOW), 1st Lt., U.S.A., Hdq., 140th Bomb Wing, P.R.O., Germany; *for mail,* 26 Bucklin, North Attleboro, Mass.
Graham, Glenn Robert (J'41) (BCJ), 1594 Wood Rd., Cleveland Heights, Ohio.
Graham, John Joseph (J'40) (FKS), 1st Lt., Sferics Div., Army Air Forces; 20020 Lomond Blvd., Shaker Heights 22, Ohio.
Graham, John Joseph (J'42) (MC), Mech. Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1735 Wendell Ave., Schenectady 8, N.Y.
Graham, John Stewart (J'44) (CMSZY), Ensign, U.S.N., YMS-124, F.P.O., San Francisco, Calif.; *for mail,* 16 Auburn St., Woburn, Mass.
Graham, Miss Lois (J'45) (AFK), Carrier Corp., Syracuse; *for mail,* 1704 S. Salina St., Syracuse 5, N.Y.
Graham, Mack Edward (J'41) (CMJ), Ensign, U.S.N.R., 9213 Annetta Ave., South Gate, Calif.
Graham, Paul S. ('43) (DCJMY), Switch Board Engrg., Mech. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 1058 Blackridge Rd., Pittsburgh 21, Pa.
Graham, Robert Fisher (J'43) (ACM), Lt. (j.g.), Instrument Maint., U.S.N., Bur. of Ships, Code 945, Navy Dept., Washington, D.C.; *for mail,* 5616 Pimlico Rd., Baltimore 9, Md.
Graham, Warren Wright ('18; '35) (SEFK), Sr. Mech. Engr., Plumbing & Htg., Fed. Pub. Housing Authority, Elec. Bldg., Ft. Worth 2; *for mail,* 611 W. 4th St., Ft. Worth 3, Tex.
Graham, William M. (J'33) (CLDOS), 9137 S. Ada St., Chicago 20, Ill.
Granata, A. J. (J'35) (CKL), Mgr., Applied Engrg. Dept., Pure Carbonic, Inc., 60 E. 42nd St., New York, N.Y.
Granberg, Bengt Rudolf ('41) (CHJLMN), Ch. Devel. Engr., Sundstrand Mch. Tool Co., Rockford, Ill.
Grandahl, Roland L. (J'41) (CMNJHB), Partner, Tool Engr., Grandahl Tool & Mch. Co., 1200 Park St., Hartford 6; *for mail,* 252 Sisson Ave., Hartford, Conn.
Grandin, Frederick ('46) (FMT), Application Engr., Westinghouse Elec. Corp., 1180 Raymond Blvd., Newark; *for mail,* 9 Theodore St., Wharton, N.J.
Grandinetti, John Russell ('45) (HBKOGC), Mech. Engr., Bur. of Engrg., Dept. of Pub. Works, City of Philadelphia, 1103 City Hall Annex, Philadelphia 7, Pa.
Granell, Edward Charles (J'45) (BJLMOZ), 285—18th Ave., Newark 3, N.J.
Granet, Irving (J'44), U.S. Army; 375 Rockaway Pkwy., Brooklyn 12, N.Y.
Graney, Robert William (J'39) (ARO), 4761 Rapid Run Pike, Cincinnati 5, Ohio.
Granger, Charles H. (A'34) (HCYL), V.P., Gen. Mgr., Waterbury Tool Div., Vickers, Inc., Waterbury 91; *for mail,* 199 Euclid Ave., Waterbury Conn.
Granger, Graham ('37) (FCERDS), Asst. V.P., East. Gas & Fuel Associates, 250 Stuart St., Boston 16, Mass.
Granger, John Maxwell (J'42) (ACM), 1266 Haro St., Vancouver, B.C., Can.
Granger, Thomas Scott (J'41) (NMJYG), Mech. Engr., RCA Victor Co. Ltd., 1001 Lenoir St., Montreal, Que., Can.
Granniss, Edward Robert ('39) (MGCD), Dir., Indus. Div., Natl. Conservation Bur., 60 John St., New York 7, N.Y.
Gransee, Harold Clifton (J'40) (SFEKA), Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Grant, Arthur A. ('19) (CMP), 1745 Canal Bldg., New Orleans 12, La.
Grant, George Allen (J'43) (MCS), 920 Spruce St., Seattle, Wash.
Grant, H. C., Jr. (J'23), 165 Linwood Ave., Ridgewood, N.J.
Grant, Harold Robert (J'42) (BCM), Jr. Prod. Engr., Fluorescent Products Inc., 1240 S. Main St., Los Angeles; *for mail,* 751 S. Mariposa, Los Angeles 5, Calif.
Grant, Irving George ('31; '45) (SFHKED), Mech. Plant Betterment Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Grant, Ray Michael (J'45) (ELZ), Dist. Rep., Orr & Sembower, Inc., 228 N. La Salle St., Chicago 1; *for mail,* 11132 S. Edbrooke Ave., Chicago 28, Ill.
Grant, Taylor Banker ('23; '35) (CM), Specification Supvr., Bell Tel. Labs., Inc., 463 West St., New York 14; *for mail,* 99-41—65th Rd., Forest Hills, L.I., N.Y.
Grant, William Wulfing ('25; '35) (C), Dir., Marketing Research, Westinghouse Elec. Corp., 4th St.; *for mail,* 641 Glendale Blvd., Mansfield, Ohio.
Grapnel, Stefan Ludwik (J'43) (TZCN), Research Engr., Belding Heminway Co., Putnam, Conn.
Gras, Ranulf Worcester (J'43) (BAZNM), Staff Mem., Div. of Indus. Cooperation, Mass. Inst. of Tech., 77 Massachusetts Ave.; *for mail,* 48 Massachusetts Ave., Cambridge 39, Mass.
Graser, Theodore Nestor ('38) (CFSTZ), V.P., Treas., Cochrane Steam Specialty Co., 80 Federal St., Boston 10, Mass.
Grasse, Harold ('37; '43) (SOHE), Design Engr., Black & Veatch, Cons. Engrs., 4706 Broadway, Kansas City; *for mail,* 6444 Main St., Kansas City 5, Mo.
Grassi, Raymond Charles (J'41) (MJCNBD), Instr. in Mech. Engrg., 234 Mech. Bldg., Univ. of Calif., Berkeley 4, Calif.
Grassl, Robert N. (J'41) (ABNS), Design Engr., Westinghouse Elec. Corp., Lester Branch P.O.; *for mail,* 712—10th Ave., Prospect Park 13, Pa.
Gratch, Serge (J'44) (BKS), Instr., Dept. of Mech. Engrg., Univ. of Pa., Philadelphia 4, Pa.
Grauel, James Laverty ('43) (CLM), Ch. Engr., W. C. Hamilton & Sons, Box 48; *for mail,* Joshua Rd., Miquon, Pa.
Gravelle, Paul ('44) (FKS), Results Engr., Toledo Edison Co., Toledo; *for mail,* 2143 Central Grove Ave., Toledo 9, Ohio.
Gravenhorst-Brouwer, Otto C. (J'42) (H), Engr., Charge of Pump Tests, Pacific Gas & Elec. Co., 203 N. Sutter, Stockton, Calif.
Graves, Benjamin Parker ('23) (BHJMN), Dir. of Design, Brown & Sharpe Mfg. Co., Promenade St., Providence 1, R.I.
Graves, Colburn Rundio ('45) (CEFS), Supt. (Holder-Distribution), Brooklyn Union Gas Co., 176 Remsen St., Brooklyn 2; *for mail,* 134 Tullamore Rd., Garden City, L.I., N.Y.
Graves, Edwin Hawley (J'39) (CLWMDS), Asst. Gen. Mgr., Hummel-Ross Fibre Corp., Hopewell; *for mail,* 417 Libby Ave., Richmond, Va.
Graves, Frederick Emmons (J'43) (FLKZS), Devel. Engr., Surface Combustion Div., Webster Engrg. Co., 2375 Dorr St., Toledo 1, Ohio.
Graves, Lawrence Paul (J'45), 52 Circuit Park, Melrose 76, Mass.
Graves, Walter Joseph ('17) (BCDHOW), Retired; 16811 Ferguson Ave., Detroit 27, Mich.
Gravitz, Edward (J'44), U.S.N., 6103 Ellsworth St., Philadelphia, Pa.
Gray, Cecil ('43) (CRS), Mgr., Westinghouse Elec. Corp., 801 S. 5th St., Richmond 19, Va.
Gray, Charles J. (J'39) (RS), Draftsman, Atlantic Coast Line R.R. Co.; *for mail,* 1511 Rankin St., Wilmington, N.C.
Gray, David R. ('30; '36), Mech. Engr., Charge Maint., Diamond Match Co., Box 1483, Spokane, Wash.
Gray, Donald D. A. (J'42) (CHMNYG), Lt. Comdr., U.S.N.R., Office of Asst. to Indus. Mgr., U.S.N., Ferry Bldg., San Francisco, Calif.
Gray, Edward Greeley ('42) (YMBJ), Plastics Div., Gen. Elec. Co., 1 Plastics Ave.; *for mail,* 19 Brunswick St., Pittsfield, Mass.
Gray, Ellsworth S. ('37) (EFK), Prof. & Head, Dept. of Mech. Engrg., Sch. of Engrg., Univ. of Kan., 117 Marvin Hall, Lawrence, Kan.
Gray, Frank L., Jr. (J'43) (MYBADL), 807 W. Midland St., Bay City, Mich.
Gray, Frederick Ramsey ('22) (SE), Supvr., Westinghouse Elec. Internatl. Co., 40 Wall St., New York 5, N.Y.
Gray, George Francis ('42) (GNZB), V.P., Pratt & Gray Co., Inc., Box 127, Norwalk, Conn.
Gray, H. Liggett ('16; '25; '35) (AER), Asst. Mgr., Oakite Products, Inc., 22 Thames St., New York, N.Y.
Gray, Harland Alexander ('44) (ACJ), East. Dist. Engr., Cleveland Automatic Mch. Co., 15 Washington St., Newark, N.J.; *for mail,* Chestnut Hill, Norwalk, Conn.
Gray, Harry ('28) (BDH), Box 502, Morgan City, La.
Gray, Henry C. (J'17), Engrg. Dept., Falk Corp., Milwaukee; *for mail,* 4429 N. Cramer St., Milwaukee 11, Wis.
Gray, John Cameron (J'45), St. Lawrence Starch Co., Ltd., Port Credit, Ont., Can.
Gray, John Wilson ('95; '26) (CDAFMY), Retired; 226 Windermere Ave., Wayne, Pa.
Gray, Newenham Acton ('30) (BJL), Cons. Engr., 103 McIndoe St., Wausau, Wis.
Gray, Percy Scott (J'35) (CSZ), Mgr., H. S. Gray Co., 74 S. Queen St., Honolulu 2, T.H.
Gray, Robert Lee (J'41) (ECHSMD), Mar. Supt., Ashland Oil & Refining Co., 14th & Winchester, Ashland, Ky.
Gray, Thomas Collier ('39) (RMS), Comdr., U.S.N.R., Bur. of Ships, Navy Dept., Washington, D.C.
Gray, Wharton Kinsey ('28; '31; '35) (CDKLZ), Ch. Engr., Spreckels Sugar Co., 2 Pine St., San Francisco 11, Calif.
Gray, William Edward ('44), Box 264, High Point, N.C.
Gray, William Enoch ('36) (RBZSK), V.P., Engrg., Peerless Equip. Co., 332 S. Michigan Ave., Chicago 4, Ill.
Gray, William Thurston ('44) (BZKJH), Research Engr., Research & Devel. Dept., Corning Glass Works, Corning, N.Y.
Graybeal, Herbert Lee (J'40) (NBHJRW), Prin. Ord. Engr., Naval Shipyard, Mare Island; *for mail,* 58 B St., Vallejo, Calif.
Greacen, Walter, 3rd ('37) (SFKC), Mech. Engr., Charge Power Stas., N.Y. State Elec. & Gas Corp., 62 Henry St.; *for mail,* 37 Crestmont Rd., Binghamton, N.Y.
Greagan, John J. ('17; '21), Mgr., Allis-Chalmers Mfg. Co., 1303 1st Natl. Bldg., Birmingham 3, Ala.
Greanias, Gus George (J'43) (HAM), 242 E. Grand, Decatur 8, Ill.
Greaves, Donald Leonard, Jr. (J'41) (SFKOEB), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 7, N.Y.; *for mail,* 130 Prospect St., East Orange, N.J.
Greaves, Fred George, Sr. ('25; '35) (DKLWYZ), Sr. Partner, Ch. Engr., Fred G. Greaves Co., 502 Polson Bldg.; *for mail,* 2015—35th Ave. W., Seattle 4, Wash.
Grebe, John Josef ('30; '34) (AFLSZ), Dir., Physical Research Lab., Dow Chem. Co., Midland, Mich.
Greeley, Charles E. (J'37) (CEK), Mar. Engr., Merchant Mar.; Ada, Minn.
Green, Allen Percival ('44), Pres., A. P. Green Fire Brick Co.; *for mail,* 16 S. Jefferson Rd., Mexico, Mo.
Green, Andrew Shaw ('36; '46) (HBNEAJ), Supvr. of Design, Naval Arch., Office of Asst. to Indus. Mgr., U.S.N., Ferry Bldg., San Francisco 11; *for mail,* 612—37th St., Richmond, Calif.
Green, Arthur Brooks ('22) (CMWZ), Treas., Paprex Co., 284 Harris Ave., Needham 92, Mass.
Green, Ben H. ('21; '35) (CFS), Gen. Supt. Steam Plants, Cent. N.Y. Power Corp., Box 418; *for mail,* 228 E. 5th St., Oswego, N.Y.
Green, Boynton Morris ('21; '35) (NMJBL), *Student Award, '16;* Prof. Mech. Engrg., Stanford Univ.; *for mail,* 691 Salvatierra St., Stanford University, Calif.
Green, Charles E. (J'41) (CJS), Apt. 4J, New Hall Apts., 8309 Talbot St., Kew Gardens, L.I., N.Y.
Green, Charles Henry (J'45), Partner, Wm. F. Surgi Equip. Co., 1017 Magazine St., New Orleans 13, La.
Green, Clarence Raymond (J'37) (SFCZD), Jr. Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 325 Helen Ave., Trenton, Mich.
Green, Edward Orville (J'40) (AM), 3710 College Ave., Kansas City 3, Mo.
Green, Frank Benjamin Kitchener (J'43) (SLM), 528—5th St., Brooklyn 15, N.Y.
Green, Fred'k W. ('18) (BRS), Ch. Opera. Officer, St. Louis Southwest. Ry. Lines, 515 Cotton Belt Bldg., St. Louis, Mo.
Green, Heatley ('19), Pres., Automatic Products Co., 1145 W. Grand Blvd., Detroit, Mich.
Green, John S. ('37) (MNY), Works Mgr., Johnston & Jennings Co., 877 Addison Rd., Cleveland; *for mail,* 2600 S. Taylor Rd., Cleveland Heights, Ohio.
Green, Lorn Albert ('24; '28; '35) (CEHAS), Mgr., Ingersoll-Rand Co., 1460 E. 4th St., Los Angeles 33, Calif.
Green, Luther Stehley, Jr. ('44) (SJFBLZ), Pres., Mickle-Milnor Engrg. Co., 1348 Arch St., Philadelphia 7; *for mail,* 534 Ott Rd., Bala-Cynwyd, Pa.
Green, Milton (J'43) (AHBS), Aerodynamics Engr., Chance Vought Aircraft Div., United Aircraft Corp., Main St.; *for mail,* 482 Windsor Ave., Stratford, Conn.
Green, Robert Boyce (J'44) (JMY), 315 Spring St., Thomasville, N.C.
Green, Robert Joseph (J'35) (EACRYM), Sales Engr., Koppers Co., Inc., Box 626, Baltimore 3, Md.
Green, Thomas Allen ('41) (CJMNZB), Ch. Draftsman, Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail,* 2552 Kingston Rd., Cleveland Heights 18, Ohio.
Green, Truman Jos. (J'38) (CEH), City Engr., City of Spur, Spur, Tex.
Green, Walter Frederick ('44), Asst. Works Engr., Timken Roller Bearing Co., Canton 6; *for mail,* 1621 Trinity Pl., N.W., Canton 8, Ohio.
Green, Walter Lawrence ('26; '35) (CMBNA), Oper. Mgr., Todd Shipyards Corp., Foot of Dwight St., Brooklyn 31, N.Y.
Green, Walter Roberts (J'45) (ABMSOJ), Engrg. Draftsman, Douglas Aircraft Co., Inc., 3000 Ocean Pk. Blvd., Santa Monica; *for mail,* 1524 S. Bentley, West Los Angeles 25, Calif.
Green, William ('27; '35) (JMS), Head Master, Engrg. Trades Sch., Education Dept. of South Australia; *for mail,* 12 Shipster St., Torrensville, Adelaide, South Australia.
Green, William Andrew, Jr. (J'41) (FWSOKD), Field Engr., Hofft Co., Inc., 914 Capitol St., Richmond 19, Va.

Green, Wilson Payne ('32; '41) (EKAFL), Assoc. Prof. Mech. Engrg., Univ. of Md.; *for mail*, 6900 Dartmouth Ave., College Park, Md.

Greenawalt, Russell Faust (J'34) (SLHKBM), Mech. Engr., Asst. to Ch. Engr., Birds Eye-Snider, Inc., Gen. Foods Corp., 40 Franklin St., Rochester 4, N.Y.

Greenberg, J. H. (J'41) (JNKZBL), Metal. Engr., A. J. Boynton & Co., 58 E. Washington Blvd., Chicago 2; *for mail*, 5131 University, Chicago 15, Ill.

Greenberg, Morris ('40), Mech. Engr., Mid-West Heat Serv., 3336 Franklin Blvd., Chicago 24; *for mail*, 5113 Kimbark Ave., Chicago 15, Ill.

Greenberg, Sidney (J'34) (BCEHJS), Engr., Mech. (P4), U.S. Engr. Office, Corps of Engrs., War Dept., 120 Wall St., New York 5; *for mail*, 2033 Valentine Ave., New York 57, N.Y.

Greenberg, Yale Joel (J'45) (DMNJWY), Keystone Mfg. Co., 151 Hallet St., Dorchester, Mass.

Greene, Arthur M., Jr. ('95; '03; F'36; H'40) (OS), *Manager*, '13-'16, *Vice-President*, '16-'18; Dean Emeritus, Princeton Univ.; *for mail*, 19 Maple St., Princeton, N.J.

Greene, Ernest Woodruff ('12; '13; '35) (CHL), V.P., Hawaiian Sugar Planters' Association, 731 Investment Bldg., Washington 5, D.C.

Greene, Fred Herbert (J'41) (MCSY), Tenn. Eastman Corp.; *for mail*, 424 E. Tennessee Ave., Oak Ridge, Tenn.

Greene, Frederick William ('45) (HJS), Ch. Engr., Walsh Holyoke Steam Boiler Works, 110 Appleton St.; *for mail*, 11 Lexington Ave., Holyoke, Mass.

Greene, George Franklin ('17; '19; '35) (SKFJ), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 28 Booth St., Hempstead, N.Y.

Greene, Herbert A. (J'40) (CM), 1900 Quentin Rd., Brooklyn 29, N.Y.

Greene, Isaac C. ('82; '86); *address unknown.*

Greene, Lawrence S. (J'45), Pierre Hotel, 61st St. & 5th Ave., New York, N.Y.

Greene, Nathanael Bacon, Sr. (A'46) (DJR), Design Engr., Am. Steel & Wire Co., New Haven 7; *for mail*, Long Hill Rd., Guilford, Conn.

Greene, Richard Schoonmaker (J'46) (AEH), Sr. Test Engr., Wright Aero. Corp., Paterson; *for mail*, 44 St. Luke's Pl., Montclair, N.J.

Greene, Russell deC. ('35) (CLS), Ch. Engr., Am. Cyanamid Co., 30 Rockefeller Plaza, New York, N.Y.

Greene, Thomas William ('27) (BJS), Devel. Engr., Linde Air Products Co., 30 E. 42nd St., New York, N.Y.

Greenfield, Benjamin ('18) (BEF), Research Engr., Combustion Utilities Corp., Linden; *for mail*, 1062 Morriss Ave., Union, N.J.

Greenfield, Horace Dobson ('45) (MCDAJ), Engr., Plant & Process Layout, Gen. Ry. Sig. Co., 801 West Ave., Rochester 2; *for mail*, 34 Rand Pl., Pittsford, N.Y.

Greenhalgh, John ('21; '35) (CSMBLO), Mech. Engr., Holtzer-Cabot, Div. of First Indus. Corp., 125 Amory St., Boston; *for mail*, 28 Ware Rd., Auburndale, Mass.

Greenhalgh, John Eric (J'39) (CMJAO), Sr. Financial Analyst, United Nations Relief & Rehabilitation Admin., Washington, D.C.; *for mail*, 2807 N. Glebe Rd., Arlington, Va.

Greenhill, Harold ('29) (C), Pres., Wm. P. Greenhill & Son, Inc., 500 W. Division St., Chicago 10, Ill.

Greenleaf, Leland B. ('30; '37) (MCYW), Ch. Engr., C. G. Conn, Ltd., 1449 E. Beardsley Ave.; *for mail*, 1449 Greenleaf Blvd., Elkhart, Ind.

Greenman, Edwin G. ('07; '13) (HNBWSG), Devel. Engr., Norton Door Closer Co., 2900 N. Western Ave., Chicago 18; *for mail*, Cedar St., New Lenox, Ill.

Greenman, Hugh Merrill (J'40) (MCANHJ), Asst. Project Engr., Woodward Governor Co., 5001 N. 2nd St.; *for mail*, 5405 Hollis Ave., Rockford, Ill.

Greenslade, Grover Rawle ('39) (RBJKN), Dir. of Research, Flannery Bolt Co., Bridgeville; *for mail*, Box 100, R.D. 5, Pittsburgh 5, Pa.

Greenstreet, Mervelle Edward (J'43) (ABC), Aviation Cadet, Army Air Forces, Sqd. A, 42nd Unit, Box 45, Maxwell Field, Ala.; *for mail*, 1039 Maple Ave., Bartlesville, Okla.

Greenwald, D. Ulrich (J'38) (MCD), Asst. Prof. Mech. Engrg., Univ. of Del.; *for mail*, 160 Elkton Rd., Newark, Del.

Greenwald, Daniel Jack, Jr. (J'44) (CJMN), Project Mgr., Indus. Tool & Die Works, Inc., 2824 University Ave., S.E., Minneapolis 14; *for mail*, 4707 Dupont S., Minneapolis 9, Minn.

Greenwall, Walter L. ('11) (BES), Mech. Engr., Nordberg Mfg. Co., cor. Chase & Oklahoma Sts.; *for mail*, 3036 S. Superior St., Milwaukee, Wis.

Greenwood, Heman ('18; '25; '35) (HSC), V.P., Charge Internatl. Div., Carrier Corp., S. Geddes St., Syracuse 1; *for mail*, 125 Brewster Rd., Scarsdale, N.Y.

Greenwood, Lloyd Etchells (J'41) (LSZ), Mech. Supvr., Calco Chem. Div., Am. Cyanamid Co.; *for mail*, Gateville Rd., R.F.D. 1, Bound Brook, N.J.

Greenwood, Marvin H. (J'38) (AMC), Route 3, Main St., R.D., Houston, Tex.

Greenwood, Orville W. (J'41), Draftsman, Hyd. Design, Glenn L. Martin Co., Middle River; *for mail*, 9-B Glenwood Rd., Baltimore 21, Md.

Greer, Charles Henry ('35; '44) (CHSY), Plant Engr., Clarksville Div., B. F. Goodrich Co., Pettus St.; *for mail*, 203 Maplemere St., Clarksville, Tenn.

Greer, Everett Sidney ('21; '35) (CKFM), Supt., Plant 2, Hazel-Atlas Glass Co., Zanesville, Ohio.

Greeson, Edward Owen (J'41) (CKSW), Serv. Dept., Meyer-Blanke Co., 310 Russell Blvd., St. Louis 4, Mo.

Greffe, C. Dale (J'40) (NBMEG), Assoc. in Mech. Engrg., Univ. of Ill., Urbana, Ill.

Greger, Henrik ('04; '15), Cons. Engr., 11 Broadway, New York, N.Y.

Gregg, Donald (J'40) (ACD), Exec., Faber, Coe & Gregg, Inc., 206 W. 40th St., New York 18, N.Y.; *for mail*, 115 Undercliff Rd., Montclair, N.J.

Gregg, Frank Davis ('28; '34; '35) (JZKLBG), Dist. Mgr., E. A. Wilcox Co., 406 S. Main St., Los Angeles 13; *for mail*, 221 N. Rampart Blvd., Los Angeles 26, Calif.

Gregory, Richard Sears (J'42) (JMD), Mech. Engr. & Partner, Kemp Mchy. Co., 211 President St.; *for mail*, 204 Lambeth Rd., Baltimore 18, Md.

Gregory, William Bres ('22; '38) (EFHLS), Asst. Mgr., Dallas Office, A. M. Lockett & Co., Ltd., 401 Magnolia Bldg., Dallas 1, Tex.

Gregory, William Kendrick ('41) (AEMC), Maj., Army Air Forces, Unit Chief, Equip. Lab., Wright Field, Dayton, Ohio.

Greiner, Charles James ('32; '35) (LRWBCD), Div. Engr., Kimberly-Clark Corp., 128 N. Commercial St., Neenah; *for mail*, 813 Manitowoc St., Menasha, Wis.

Greiner, Frank ('25; '35), Branch Mgr., Landis Tool Co.; *for mail*, 166 Connecticut Ave., Detroit, Mich.

Greiner, Joseph Charles ('41; '45) (NMCBJ), Mch. Designer, E. W. Bliss Co., 1420 Hastings St., Toledo 7; *for mail*, 3655 Anderson Pkwy., Toledo 6, Ohio.

Greiner, Joseph Edwin (J'45) (ZMNA), Product Designer, Taylor Instrument Cos., 95 Ames St., Rochester; *for mail*, 101 Washington St. S., Rochester 8, N.Y.

Greist, Alva O. ('17) (SHFEOC), Advis. Engr., Reconstruction Finance Corp., 811 Vermont Ave.; *for mail*, 2100 Connecticut Ave., Washington, D.C.

Grenoble, David H. (J'39) (MS), Application Engr., Gen. Elec. Co., 700 E. Franklin St., Richmond 17; *for mail*, R.F.D. 9, 25 Cherokee Rd., Richmond, Va.

Grenzke, George R. (J'45), Ensign, c/o Wolverine Tool Co., 1480 Woodbridge St., E., Detroit 7, Mich.

Greten, Nicholas John, Jr. (J'43) (MJHSBC), Ensign, U.S.N., Bachelor Officers' Quarters 376, N.T.S., Newport, R.I.; *for mail*, 1624 W. 5th St., Brooklyn 23, N.Y.

Grether, Edward C. ('21), Equip. Engr., Am. Sugar Refining Co., 120 Wall St., New York 5, N.Y.

Greve, Frederick William (M'43) (HBO), Prof. Hyd. Engrg., Purdue Univ., West Lafayette, Ind.

Grey, Ralph Eugene, Jr. (J'45), Aero. Engr., Eng. Research Div., Natl. Advis. Com. for Aero., Cleveland Airport; *for mail*, 9414 Clifton Blvd., Cleveland, Ohio.

Greyson, Frederick Raymond ('19; '25) (DBOSMN), Ch. Mech. Engr., Kaiser Engrs., Inc., 1924 Broadway, Oakland 12, Calif.

Gribbel, John, II (J'40) (RJEC), Customer Contact Engr., Edw. G. Budd Mfg. Co., Red Lion Rd., Bustleton; *for mail*, Buck Rd., Huntington Valley, Pa.

Gridley, Allen Hubert ('20; '30) (HSKDEO), Ch. Engr., H. S. Ferguson & Co., 200—5th Ave., New York 10, N.Y.

Grier, Alex Monro (A'45) (CDEL), Gen. Pur. Agt., McColl-Frontenac Oil Co. Ltd., 420 Royal Bank Bldg., 360 St. James St., Montreal, Que., Can.

Grieshaber, Emil ('23; '35) (EBSCF), Ch. Engr., Nordberg Mfg. Co.; *for mail*, 3209 W. Layton Ave., Milwaukee 7, Wis.

Grieve, Albert ('15; '35), Prof. Mch. Design, Lima Sch. Min. Engrs., Municipal Elec. Engr., Lima City; Owner, Factoria y Garage; *for mail*, Apartado 615, Lima, Peru, S.A.

Griffin, Charles Lewis ('93), Retired; Box 238, Vaughan Arms Apts., Palm Springs, Calif.

Griffin, Fred Swigart ('28) (HSBA), Prof. Mech. Engrg., Chmn., Mech. Engrg. Dept., Univ. of Akron, 302 E. Buchtel Ave., Akron 4, Ohio.

Griffin, Gifford (J'39) (SCF), Asst. Boiler Rm. Engr., Kearny Generating Sta., Pub. Serv. Elec. & Gas Co., Kearny; *for mail*, 737 Irvington Ave., Maplewood, N.J.

Griffin, James Birney ('43) (HKN), Hyd. Engr., Byron Jackson Co., 2150 E. Slauson, Los Angeles 54, Calif.

Griffin, John Buckbee ('44), Acting Ch. Engr., U.S. Rubber Co., 74 Brown St., Waterbury; *for mail*, 510 Main St., Oakville, Conn.

Griffin, Joseph Francis ('44), Asst. Ch. Engr., Superheater Co., 151st St. & R.R. Ave., East Chicago; *for mail*, 43 Warren St., Hammond, Ind.

Griffin, Norman Edward (J'43) (CMZDJL), Coordinator, Fractional Hp. Motor Planning, Gen. Elec. Co., 1635 Broadway; *for mail*, 226 E. Washington Blvd., Ft. Wayne 2, Ind.

Griffin, Randall D. (J'38) (SRHK), Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee; *for mail*, 2146 N. 73rd St., Wauwatosa 13, Wis.

Griffis, William K. (J'39) (CJM), 418 Hardin Apts., Waco, Tex.

Griffith, Earl Leonard ('45) (EFKMOS), Asst. V.P., Constr. & Shops, Consltd. Edison Co. of New York, Inc., Rm. 1414, 4 Irving Pl., New York 3, N.Y.

Griffith, Forrest Lee, Jr. (J'41) (AGCMWS), Lt., Head of Exper. Projects Sec., Radio & Elec. Branch, Bur. of Aero., Navy Dept., 18th & Constitution Ave., Washington, D.C.; *for mail*, 7718 Liberty Rd., Baltimore 7, Md.

Griffith, Gilbert B. (J'35) (ABH), Assoc. Engr. (Mech.), U.S. Engr. Dept., 300 Broadway, Little Rock, Ark.

Griffith, Leigh M. ('15; '28) (AEJN), 115 Pocahontas Pl., Indian River Park, Hampton, Va.

Griffith, Spencer Ritner (J'40) (JBA), Engr., Product Design Dept., Hamilton Watch Co.; *for mail*, 248 W. Orange St., Lancaster, Pa.

Griffiths, Edward ('40) (BCDJNY), Dir., Hdq., Mfg. Engrg. Dept., Westinghouse Elec. Corp., Braddock Ave., East Pittsburgh, Pa.

Griffiths, William Thomas, Jr. (J'42) (LKBSM), Engr., Design, E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington; *for mail*, 808 W. 26th St., Wilmington 245, Del.

Grim, George B. (J'40) (ECN), Research Lab. Engr., Caterpillar Tractor Co., 600 W. Washington St., East Peoria; *for mail*, 211 North St., Washington, Ill.

Grimberg, John Charles (J'43) (CKM), Air Conditioning Engr., Griffith-Consumers Co., 1413 New York Ave., Washington; *for mail*, 2030—37th St., S.E., Washington 20, D.C.

Grimes, Charles C. (J'40) (A), Application Engr., Minneapolis-Honeywell Regulator Co., 2507 Commerce St., Dallas 1; *for mail*, 5615 Ridgedale, Dallas 6, Tex.

Grimes, Charles Edmund ('43) (JSD), Chief, Spiegel Dept., N.J. Zinc Co. (of Pa.); *for mail*, Residence Park, Palmerton, Pa.

Grimes, Patrick Lawrence (J'40) (FSKB), Opera. Supvr., Tidd Plant, Ohio Power Co.; *for mail*, Box 440, Brilliant, Ohio.

Grimes, Richard Vernon (J'43) (ANWBY), Ch. Engr., Hartzell Propeller Co.; *for mail*, 921 South St., Piqua, Ohio.

Grimison, Edwin Douglas ('27; '39) (FKLSEC), Exec. Asst., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.

Grimm, Adolph ('21; '25) (BGHMNO), Dir., Mech. Engrg., Design & Engrg. Div., Conlan Elec. Corp., 1042 Atlantic Ave., Brooklyn 16; *for mail*, 4 Mountain View Ave., Pearl River, N.Y.

Grimm, Eugene L. ('23) (R), 3759 Brayton Ave., Long Beach 7, Calif.

Grimmer, Ernest A. ('21; '31) (SFDBLK), Mech. Engr., Lathrop-Trotter Co., 1340 Union Trust Bldg., Cincinnati 2, Ohio; *for mail*, 907 Roosevelt Bldg., Indianapolis 4, Ind.

Grimmer, George C. (J'41) (BEP), Capt., Sig. Corps, Force Hdq., Signal Sec., A.P.O. 512, New York, N.Y.

Grimmett, Elmer J. ('26; '32; '35) (CLMS), Div. Mfg. Mgr., Gen. Foods Corp., 250 Park Ave., New York 17, N.Y.

Grimshaw, William Frederick (J'21) (EH), Serv. Engr., Ingersoll-Rand Co., 1637 Blake St., Denver 2, Colo.

Griner, John Edward (J'45) (CFR), Caterpillar Tractor Co., Peoria; *for mail*, 139 S. Eleanor Pl., Peoria 5, Ill.

Grinter, Linton E. ('44) (BCAN), V.P., Dean of Graduate Sch., Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.

Grisbaum, Leonard D. ('17; '23; '35), Charge Engrg., Research, Ry. Serv. & Supply Corp., 510 S. Harding, Indianapolis, Ind.

Griscom, Elmer W. (J'31) (CF), Tech. Dept. Foreman, Sinclair Refining Co., Marcus Hook; *for mail*, 208 E. 26th St., Chester, Pa.

Grise, Warren K. (J'42), 2nd Lt., Army Air Forces, Box 32, Army Air Field, Amarillo, Tex.

Grissy, William Andrew ('45) (CMDOLN), Indus. Engr., C.P.A., 41 Linnaean St., Cambridge, Mass.

Griswold, Edgar Allen (J'45) (BCHO), Owner, Edgar A. Griswold & Associates, 106 W. 3rd St., Los Angeles; *for mail*, 6641 Pollard St., Los Angeles 42, Calif.

Griswold, Frederick, Jr. (A'44), 233 Broadway, New York, N.Y.

Griswold, Nelson D. ('24; '38) (SELCMJ), Asst. Gen. Mgr., Tex. Div., Dow Chem. Co.; *for mail*, 1514 W. 4th St., Freeport, Tex.

Griswold, Robert Gray ('22) (SFD), Pres., Elec. Advisers, Inc., 70 Pine St., New York 5, N.Y.

Griswold, Thos., Jr. ('20) (KLW), Cons. Engr., Dow Chem. Co., Midland, Mich.

Grob, John J. ('19; '25) (EFSZ), Test. Coordinating Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.

Grobholz, Raymond Kenneth (J'44) (EHS), Sales Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail*, 564 Devon St., Arlington, N.J.

Grodner, Abraham ('27; '42) (LJNBKY), Ch. Engr., Alloy Fabricators Div., Robinson Foundation, Inc., Market St., Perth Amboy, N.J.

Groene, Willard LeBlond (A'45) (ABEHNR), Mgr., Cincinnati Office, Toulmin & Toulmin, 1109-10 Carew Tower, Cincinnati 2, Ohio.
Groff, Howard M. ('17; '35) (CMNJTY), Asst. to Pres., Conmar Products Corp., 140 Thomas St., Newark, N.J.; *for mail*, 219 Marietta St., Mt. Joy, Pa.
Groff, Joseph Coblentz (J'24) (BEHNS), Devel. Engr., Charge Power Pump Design, Aldrich Pump Co., Foot of Gordon St.; *for mail*, 111 N. 4th St., Allentown, Pa.
Groff, Wm. C. ('40) (EFP), Cons. Engr., 7252 Clayton Rd., Richmond Heights 17, St. Louis Co., Mo.
Grogan, Edward John, Jr. (J'43), 416 Thieriot Ave., Bronx, N.Y.
Gronbach, John H. ('27; '35) (SO), Dist. Mgr., Natl. Aluminate Corp., Chicago, Ill.; *for mail*, R.R. 3, Chagrin Falls, Ohio.
Gronemeyer, George Edward (J'32) (OSFEKY), Power Engr., E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington; *for mail*, 203 South Rd., Wilmington 274, Del.
Groom, Joseph Howard ('27; '35) (BCHJMN), Mech. Engr., Cincinnati Milling Mch. Co., Cincinnati; *for mail*, Box 153-J, R.R. 508, Cincinnati 27, Ohio.
Groothuis, Herman ('22) (S), Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5; *for mail*, 7108 Hilltop Rd., Upper Darby, Pa.
Groschoff, Ernst H. (J'33) (JMNWY), Ch. Draftsman, Andrea Radio Corp., 43-20—34th St., Long Island City 1; *for mail*, 56-15—141st St., Flushing, N.Y.
Groseclose, Frank Farrier ('42) (CDMO), Prof. & Head, Indus. Engrg., Ga. Sch. of Tech., 225 North Ave., N.W., Atlanta, Ga.
Grosjean, Charles Henry (J'46) (KLS), Dist. Mgr., Yarnall-Waring Co., 6432 Cass Ave., Detroit 2; *for mail*, 15087 Ward Ave., Detroit 27, Mich.
Gross, Albert (J'40) (EFKOS), Engr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York; *for mail*, 41-41—41st St., Long Island City 4, N.Y.
Gross, Claude M. ('27) (CDLW), 1727 E. 3rd St., Tucson, Ariz.
Gross, Frank Robert ('45) (BKO), Project Engr., Firestone Tire & Rubber Co., Akron; *for mail*, 565 Malvern Rd., Akron 8, Ohio.
Gross, Joseph Watson, Jr. (J'42) (CM), Engr., Indus. Statistics, Tube Div., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Gross, Lester (J'45) (KCB), Ch. Engr., Gen. Installation Co., 2234 Olive, St. Louis, Mo.
Gross, Michael F. ('24), Ch. Engr., Colonial Sugars Co.; *for mail*, Box 64, Gramercy, La.
Gross, Samuel ('23; '35) (LSCTY), Partner, Sheffler-Gross Co., 1000 Drexel Bldg., Philadelphia 6, Pa.
Grossenbacher, Ernest ('21; '35) (BMS), Ch. Engr., Fajardo Sugar Co., Fajardo, P.R.
Grossenbacher, Ernest, Jr. (J'40) (NMDRSZ), Mech. Design Engr., Pittsfield Apparatus Works, Gen. Elec. Co., 100 Woodlawn Ave.; *for mail*, 66 Taylor St., Pittsfield, Mass.
Grosser, Christian Ernest ('44) (HJNBSO), V.P., Ch. Engr., Stand. Mchy. Co., Providence 7, R.I.
Grosser, Wilfred R. (J'41) (BES), 616 Ramapo Rd., Teaneck, N.J.
Grosskopf, LaVerne R. (J'41), Lt., U.S.A.; 2370 Rockingham Rd., Davenport, Iowa.
Grossman, Franklin Arthur (J'37) (EFKMHB), Sales Engr., Servel, Inc., 122 S. Michigan Ave., Chicago 3, Ill.
Grossman, Martin Luther ('45), Celanese Corp. of Am., 180 Madison Ave., New York 16, N.Y.
Grossman, Nicholas (J'44) (BJCM), Staff Mem., Research Engr., Div. of Indus. Cooperation, Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Grossman, Paul Robert (J'43) (FSECL), Research & Devel. Div., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Grosswendt, Carl J. ('42) (CMO), Indus. Engr., Fed. Tel. & Radio Corp., 200 Mt. Pleasant Ave., Newark; *for mail*, 111 McKinley St., Nutley 10, N.J.
Grosswendt, Carl Theodore (J'42) (CMDW), William Mogey & Sons, Inc., North Plainfield; *for mail*, 527 Carleton Rd., Westfield, N.J.
Grosz, Melvin Harry (J'40) (CLY), Admin. Asst. to Mgr., Chem. Products Div., Stand. Oil Co. of N.J., North Baton Rouge, La.
Groth, Hans F. A. ('45) (NMCZ), Cons. Engr., 52 Summit Rd., Port Washington, N.Y.
Grothouse, Frank Thomas ('37) (BNMKSY), Engr., Automatic Blanket & Sunlamp Div. 22-E, Gen. Elec. Co., 1285 Boston Ave., Bridgeport 2, Conn.
Grove, Clifford Theodore (J'41), Tool Engr., Glenn L. Martin Co. (of Middle River, Md.); *for mail*, 1111 Park Ave., Omaha, Neb.
Grove, Paul Forest (J'42) (BKS), Engr., Serv., Westinghouse Elec. Corp., 411 N. 7th St., St. Louis 1, Mo.
Grove, Willard Virgil (J'42) (BLS), Design, Babcock & Wilcox Co., Barberton; *for mail*, R.D. 1, Clinton, Ohio.
Grove, Wm. G. ('18) (CBMOAD), Engr., Bridges & Structures, Conn. State Highway Dept., Hartford, Conn.
Grover, Ronald E. (J'42) (AJM), 12 State Rd., Binghamton, N.Y.
Grow, Harlow Bovier (J'43) (ACS), Pilot, Transcontinental & West. Air, Inc., 10 Richards Rd., Kansas City; *for mail*, Box 87, Linden, Mo.
Grow, Joseph Albert ('18; '35) (E), Design Engr., Bovaird & Seyfang Mfg. Co., 161 Main St.; *for mail*, 199 South Ave., Bradford, Pa.
Growdon, James Paul ('35) (HSKO), Ch. Hyd. Engr., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19, Pa.
Grubb, Wm. C. (J'46) (ACD), 1007 W. Main St., Dothan, Ala.
Grube, Charles Wallace (J'41) (LDB), Pvt., U.S.A.; 5409 E. 25th St., Kansas City, Mo.
Grube, Donald E. ('27; '39) (EFSKR), Engr., Owens-Ill. Glass Co.; *for mail*, 2330 Mound St., Alton, Ill.
Grube, Frederick John (J'44) (AJS), 62 Colonial Ave., Lynn, Mass.
Grube, Wallace McQuown (J'44) (AEH), 545 Prospect Ave., Oradell, N.J.
Grubelich, Matthew J. (J'39) (BCEFJN), Asst. Exec. Engr., Plainfield Plant, Mack Mfg. Corp., 935 S. 2nd St.; *for mail*, 300 W. 7th St., Plainfield, N.J.
Gruber, Gottlieb John ('24) (BHL), Design-Devel. & Test Engr., Morris Mch. Works; *for mail*, 18 W. Oneida St., Baldwinsville, N.Y.
Gruber, Jerome M. (J'41) (EJS), Capt., Hdq. 1103rd Engr. Combat Gp., A.P.O. 230, New York, N.Y.
Gruca, Walter (J'39) (SKJF), Asst. Engr., Stand. Forgings Corp., East Chicago; *for mail*, 918—174th St., Hammond, Ind.
Gruenberg, Emil ('45) (KLS), 3405 Putnam Pl., New York 67, N.Y.
Gruenberg, Otto C. ('27; '32) (RCESKB), Supt. M.P., N.Y., Susquehanna & West. R.R. Co., 160 Market St., Paterson 1, N.J.
Gruendler, Wm. P. ('27; '35) (BCDFJL), Treas., Gen. Mgr., Gruendler Crusher & Pulverizer Co., 2915 N. Market St., St. Louis 6; *for mail*, 27 The Orchards, Clayton 5, Mo.
Gruetjen, Frederick A. ('37) (CLBMRA), Products Engr., A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1, Wis.
Gruett, LeRoy Emil (J'45) (BHMNSE), Technician Charge Exper. Lab., Falk Corp., 3001 W. Canal St., Milwaukee 8; *for mail*, 4460 N. 30th St., Milwaukee 9, Wis.
Grugan, William Thornton (J'45), Lt. (j.g.), U.S.N.R., U.S.S. *Miller*, DD-535, F.P.O., San Francisco, Calif.
Grumblatt, Victor Joseph (J'43) (MJE), Refrigerator Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.; *for mail*, 820 W. 22nd St., Erie, Pa.
Grundberg, Curtis Axel (J'43) (NMJBCA), Designer of Serv. Tools, Sperry Gyroscope Co., Inc., 4th Floor, 81 Willoughby St., Brooklyn 2, N.Y.
Grundman, Richard William (J'38) (DLON), Mech. Engr., John Brackett Co., 1860 Broadway, New York 23, N.Y.; *for mail*, 14 Ryder Rd., Fairlawn, N.J.
Grundy, Alfred Stephen (J'44) (KHBALZ), Mech. Engr., Shell Devel. Co., 100 Bush St., San Francisco 6, Calif.
Gruner, Frederick Rusticus ('46) (BJL), Allis-Chalmers Mfg. Co., Milwaukee; *for mail*, 3160 N. 41st St., Milwaukee 10, Wis.
Grunert, Arthur Earl ('15; '21; '28) (CFKS), *Melville Medallist*, '31; Supt. Generating Stas., Commonwealth Edison Co., 1111 W. Cermak Rd., Chicago 8, Ill.
Grunewald, Robert Lockhart (J'44) (CDJ), Engr., Gen. Elec. Co., Broadway, Ft. Wayne; *for mail*, 627 W. Main St., New Haven, Ind.
Grupe, William F. ('13; '21; '35) (G), 4511 New York Ave., Union City, N.J.
Grutle, Reinert O. M. (J'35) (SEFM), Twin Valley, Minn.
Grutzner, Fritz Paul ('23) (RCESFA), Engr., Charge Diesel-Loco. Engrg., Fairbanks, Morse & Co.; *for mail*, 1133 Milwaukee Rd., Beloit, Wis.
Gruver, John Henry ('45) (BCLMN), Mech. Engr., Addressograph-Multigraph Corp., 1200 Babbitt Rd., Cleveland 17; *for mail*, 1849 Raylor Rd., Cleveland 12, Ohio.
Gryglas, Steven (J'38), 3121 N. 79th Ave., Elmwood Park 35, Ill.
Grygotis, Withold J. (J'37) (LMS), 2122 Ingalls Ave., Linden, N.J.
Gschwind, Jean F. ('42) (LKTYS), Devel. Engr., J. O. Ross Engrg. Corp., 350 Madison Ave., New York 17; *for mail*, 720 Grand St., Mamaroneck, N.Y.
Guastella, Salvador Felix (J'40) (BDHKNS), Auxiliary to Ch. Engr., Cia. Azucarera Atlantica del Golfo, Central Moron, Piña, Camagüey, Cuba.
Gubitose, Nicholas Francis (J'44) (ABDNRT), Asst. Engr., U.S. Textile Mch. Co., 411 Gilligan St., Scranton; *for mail*, 748 Monroe Ave., Scranton 10, Pa.
Guckelberger, George Weyler ('40) (CFK), Secy., Treas., Gen. Mgr., Stegmaier Brewing Co., 152 E. Market St., Wilkes-Barre; *for mail*, 29 Hedge Pl., Kingston, Pa.
Guden, Jack C. (J'34), 52 Roosevelt Ave., Lynbrook, L.I., N.Y.
Gudmens, H. Wm. ('27) (CAMLSY), Financial Engr., Thomas Emery's Sons, Inc., 4500 Carew Tower, Cincinnati 1; *for mail*, 3521 Zumstein Ave., Cincinnati 8, Ohio.
Gudmundsen, Austin ('24; '34) (ABE), 4018—97th St., Inglewood, Calif.
Gudmundson, Gunnar G., Jr. (J'39) (CGM), Tool Engr., Mich. Designing & Engrg. Co., 8445 W. Fort St., Detroit, Mich.; *for mail*, 160 Lincoln Ave. E., Roselle Park, N.J.
Guentert, Donald Charles (J'41), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 9014 Beech Ave., Cleveland 9, Ohio.
Guenther, Edgar Gardiner (J'37) (TL), Asst. Supt., Acetate Rayon Div., Tenn. Eastman Corp.; *for mail*, 1308 Linville St., Kingsport, Tenn.
Guenther, Michael J., Jr. (J'37) (HM), 24 Day St., Clifton, N.J.
Guenzel, Rudolph Andrew (J'43) (JAMC), Welding Specialist, Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4; *for mail*, 4602 Wayne Ave., Philadelphia 44, Pa.
Guerdan, George Alfred ('39) (NMBWOC), Asst. Prof. Mech. Engrg., Sch. of Tech., College of the City of N.Y., 138th St. & Convent Ave., New York, N.Y.; *for mail*, 755 Anderson Ave., Cliffside Park, N.J.
Guernsey, Charles O. ('30) (CENS), V.P., Marmon-Herrington Co., Inc., 1511 W. Washington St., Indianapolis 7, Ind.
Guerrero, Joaquin Eugene ('45) (KOS), 4801 Lemmon Ave., Dallas 9, Tex.
Guggenheim, S. Frederic (J'39) (BEKNYS), Engr., Madison Metal Works, 42 Park Ave., Madison; *for mail*, 283 S. Center St., Orange, N.J.
Guiberson, S. Allen, III ('41), Pres., Guiberson Corp., Box 1106, Dallas, Tex.
Guidon, Michael, III (J'42) (AHM), Student Engr., Hyatt Bearings Div., Gen. Motors Corp., Harrison; *for mail*, 121 Mt. Pleasant Ave., Newark, N.J.
Guillemette, Joseph Dydime ('38) (OSTFLH), Engr., Jenks & Ballou, 2600 Indus. Trust Bldg., Providence 3; *for mail*, 53 Auburn St., Pawtucket, R.I.
Guillou, John C. (J'43) (HARW), Army of U.S.; 8512 Lilienthal Ave., Los Angeles 45, Calif.
Guillou, Rene ('46) (EGK), Engr., College of Engrg., Univ. of Calif.; *for mail*, 10810 Holman Ave., Los Angeles 24, Calif.
Guilloud, Lynn Henry (J'46), G. H. Lossing Mch. Works, 811 Exposition Ave., Dallas; *for mail*, Route 1, Pottsboro, Tex.
Guinan, John F. ('30; '36) (BCRS), Asst. Engr., Mech. Engrg. Dept., Consltd. Edison Co. of N.Y., 4 Irving Pl., New York; *for mail*, 58 Remsen St., Brooklyn, N.Y.
Guins, Sergei George (J'43) (BAHW), Mathematician, Propeller Div., Curtiss-Wright Corp., 880 Main St., Stamford, Conn.
Guins, Vsevolod Georgievich (J'45) (EHS), Design Engr., Joshua Hendy Iron Works, Sunnyvale; *for mail*, 951 Forest Ave., Palo Alto, Calif.
Guldberg, Dan Haigh ('21; '35) (BCDJMN), Plant Engr., Am. Cast Iron Pipe Co., Birmingham 2; *for mail*, 2842 Bush Blvd., Birmingham 8, Ala.
Gulick, Horace Silliman ('45) (RHACJS), Asst. to 1st V.P., N.Y. Air Brake Co., Watertown, N.Y.
Gulick, Lee Nelson ('23; '35) (JCL), Prof. Mech. Engrg., Asst. to Dean, Towne Sci. Sch., Univ. of Pa., 33rd & Locust Sts., Philadelphia 4, Pa.
Gulick, Walter Cox (J'43) (BJC), Indus. Engr., Ethyl Corp., Box 341, North Baton Rouge; *for mail*, 3242 Beech St., Baton Rouge, La.
Gulliksen, John Walter (J'26) (MJC), Asst. to Gen. Mgr., Worcester Pressed Steel Co., 100 Barber Ave., Worcester 6; *for mail*, 10 Perrot St., Worcester 2, Mass.
Gullikson, Albert Clarence ('25; '35) (GNME), Gen. Engrg. Instr., Univ. of Wash.; *for mail*, 4320 E. 56th St., Seattle 5, Wash.
Gumaer, Pierre Laertes ('20; '27; '35) (CF), Mech. Engr., Tex. Co., 135 E. 42nd St., New York 17; *for mail*, 90 Kent Ave., Hastings-on-Hudson 6, N.Y.
Gumprich, Wilbert C. (J'40), Estimator, Mar. Design & Estimating Div., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 799 E. 38th St., Brooklyn, N.Y.
Gunby, Frank M. ('15) (LOST), Assoc., Chas. T. Main, Inc., 201 Devonshire St., Boston 10; *for mail*, 12 Manchester Rd., Winchester, Mass.
Gunderson, Charles Kenneth (J'44) (ANKS), Ensign, U.S.N.R., Asst. Engrg. Officer, *Chikaskia*, AO-54, F.P.O., San Francisco, Calif.; *for mail*, Box 877, Grants, New Mex.
Gunderson, G. Charles ('26) (DHY), Mech. Engr., Bakelite Corp., River Rd., Bound Brook, N.J.
Gunderson, Lewis Olin ('45) (KRJT), Asst. V.P., Chem. Engrg. Research, Dearborn Chem. Co., 310 S. Michigan Ave., Chicago 4, Ill.
Gundrum, John Harry (J'40) (LDJ), Project Engr., Armstrong Cork Co., Liberty & Charlotte Sts.; *for mail*, 21 E. Lemon St., Lancaster, Pa.
Gung, George (J'40) (ABHMNR), Test. Engr., Radiographer, Hydro-Elec. Power Comm. of Ont., 8 Strachan Ave., Toronto, Ont., Can.
Gunkel, Kenneth Marvin (J'38) (B), Maj., 2nd Army Air Force, Colorado Springs, Colo.
Gunkel, Robert E. (J'41) (CM), 2679 E. 27th St., Brooklyn 29, N.Y.
Gunnell, Bruce Covington (J'32) (RSNJEF), Diesel Engr., So. Ry. System, 15th & K St., Washington 13, D.C.
Gunning, William A. ('36) (NZ), Mech. Engr., Head Draftsman, Am. Optical Co.; *for mail*, 119 Hartwell St., Southbridge, Mass.
Gunter, Addison Yancey ('45), Dir. of Devel., Alco Products, Inc., 30 Church St., New York; *for mail*, 29 Bonnett Ave., Larchmont, N.Y.

Gunther, Carl (J'44) (CMNJS), Asst. Ch. Engr., Pollak Mfg. Co., 541 Devon St., Arlington; *for mail,* 155 Union Ave., Rutherford, N.J.
Gunther, Charles Otto ('00; '05; '19) (BEKJFM), Prof. Math. & Ord. Engrg., Stevens Inst. of Tech., Box 822, Hoboken, N.J.
Gunther, Frederick J. (J'38) (AEP), 429 N. Grand Ave., Waukesha, Wis.
Guppenberger, Francis John (J'44) (KJHC), Rating Engr., Graham Mfg. Co., 4-12 Howard St.; *for mail,* 75 S. Main St., Batavia, N.Y.
Gupton, Theodore ('44) (ERKF), Dist. Mgr., Diesel Div., Am. Loco. Co., 1213 Commerce Bldg., Kansas City 6; *for mail,* 320 E. 68th Terrace, Kansas City 5, Mo.
Gurin, Herman M. (J'36) (BLOZC), Devel. Engr., Natl. Broadcasting Co., 30 Rockefeller Plaza, New York 20; *for mail,* 4877 Broadway, New York 34, N.Y.
Gurney, Wm. B. ('24; '35; '35) (CFK), Effic. Engr., Gulf States Utilities; *for mail,* 2631—100 Oaks Ave., Baton Rouge, La.
Gurvitch, Joseph Enoch (J'40) (WYAD), Plywood Engr., Plymold Corp., Lawrence; *for mail,* 87 Ft. Pleasant Ave., Springfield, Mass.
Gus, Charles E. ('28; '41) (BR), Exec. Secy., Prof. Engrg. Mechanics, College of Engrg., N.Y. Univ., University Heights, Bronx 53; *for mail,* 8801 Shore Rd., Brooklyn 9, N.Y.
Gusler, Denard Lee (J'44) (ES), Bassetts, Va.
Guss, Edward ('39; '45) (CGMB), Sales Engr., T. W. & C. B. Sheridan Co., 135 Lafayette St., New York, N.Y.; *for mail,* 85 Van Reypen St., Jersey City 6, N.J.
Gussack, Seymour Irving (J'45) (MLC), Product Designer, Grant Pulley & Hardware Co., Broadway & 57th St., Woodside; *for mail,* 30-29—150th Pl., Flushing, N.Y.
Gustafson, James Keith (J'44) (AZK), 30 W. Beacon, West Hartford 7, Conn.
Gustav, Cornelius Frederick (A'45) (CMDNBS), Dir. of Installations, Stand. Indus. Mgmt. Co., 1776 Broadway, New York 19; *for mail,* 136 Waverly Pl., New York 14, N.Y.
Gustavsen, Emil ('28; '32; '35) (LST), 264—10th St., Hoboken, N.J.
Gute, Harry (J'39) (CSHL), Field Engr., Allis-Chalmers Mfg. Co., 715 N. Van Buren St., Milwaukee 2, Wis.
Gutekunst, Ralph B. ('40) (CFKS), Effic. Engr., Generating Stas., Commonwealth Edison Co., 1111 W. Cermak Rd., Chicago 8, Ill.
Guthrie, Albert N. ('35; '41) (ELS), Supt. of Services, Libbey-Owens-Ford Glass Co.; *for mail,* 818 Congress St., Ottawa, Ill.
Guthrie, George Grant ('37) (SKF), Mech. Engr., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.; *for mail,* 711 Coolidge St., Westfield, N.J.
Gutleben, Chester (J'42), 1622-B Fernside Blvd., Alameda, Calif.
Gutleben, Donald Church (J'35) (CLS), Engr., Maint., Spreckels Sugar Co., Spreckels; *for mail,* 25 Grijalva Dr., San Francisco 12, Calif.
Gutowski, Edward Martin (J'41) (CDM), Sr. Tool Designer, Buick, Oldsmobile & Pontiac Divs., Gen. Motors Corp., Gen. Motors Bldg., Detroit 2; *for mail,* 7481 Woodmount Ave., Detroit 10, Mich.
Gutsch, Peter J. (CAKJM), 717 N. Encinita St., Temple City, Calif.
Guttormsen, Paul Andrew ('25) (FKSZ), Ch. Engr., Bird & Son, Inc., East Walpole, Mass.
Gutzwiller, Erwin Charles, Jr. (J'43) (BCS), Plant Engr., Charge of Engrg., G. Washington Coffee Refining Co., Hanover Ave., Morris Plains, N.J.
Gutzwiller, James Edward (J'41) (BNMJ), Worm Gear Engr., De Laval Steam Turbine Co., Nottingham Way, Trenton 2, N.J.
Guy, Henry Lewis ('30; F'44) (ABK), Secy., Institution of Mechanical Engineers, Storey's Gate, St. James Park, London, S.W. 1, England.
Guy, Hugh E. ('43) (CMZ), Lt. (j.g.), U.S.N.R., Prod. Engr., Inspr. of Naval Matls., 30 Church St., New York 7; *for mail,* 114-60—122nd St., Ozone Park, N.Y.
Guy, James M. ('38) (SKF), Ch. Engr., Erie City Iron Works, 1400 East Ave., Erie, Pa.
Guyette, Donald Francis (J'42) (YMCASJ), Jr. Prod. Engr., Vinylite Div., Bakelite Corp., River Rd., Bound Brook, N.J.; *for mail,* 231 Amory St., Fond du Lac, Wis.
Gwilliam, John ('27) (N), Pres., Gwilliam Co., 360 Furman St., Brooklyn 2, N.Y.; *for mail,* Box 86, Wilton, Conn.
Gygax, Edward Ernest (J'43) (KM), Lt. (j.g.), U.S.N., LCI (L)-454, F.P.O., San Francisco, Calif.; *for mail,* 5446 Claxton Ave., St. Louis 20, Mo.
Gygi, Blois Roger (J'40) (CTJM), Mgr., Compeco Dye Works, Inc., 801 E. 26th St.; *for mail,* 5000 Colonial Ave., Norfolk 8, Va.
Gysling, Manuel H. ('30) (FKS), c/o United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia, Pa.

H

Haack, Donald Carl (J'46), Instr., Ill. Inst. of Tech., 3300 Federal St., Chicago 16; *for mail,* 3815 Hollywood Ave., Hollywood, Ill.
Haack, Donald James (J'45) (LSZ), 811 Rossmore Ave., Pittsburgh 26, Pa.
Haacker, Carl William (J'45) (SKFZEC), Tech. Asst., Test Dept., Pub. Serv. Elec. & Gas Co., W. Pearl St., Burlington; *for mail,* 1235 Greenwood Ave., Trenton 9, N.J.
Haag, Herbert C. (J'37) (Y), Mech. Engr., Engrg. Research, Celanese Corp. of Am., 290 Ferry St., Newark; *for mail,* Hohokus, N.J.
Haag, Jos., Jr. ('31) (CKS), V.P., Todd Shipyards Corp., 1 Broadway, New York, N.Y.
Haag, Paul H. ('23; '25; '30), Ch. Mech. Engr., c/o São Paulo Tramway, Light & Power Co., Ltd., Caixa Postal 26-B, São Paulo, Brazil, S.A.
Haar, Selby ('12) (RBJDKN), Elec. Engr., Bd. of Trans., 250 Hudson St., New York 13; *for mail,* 25 W. 68th St., New York 23, N.Y.
Haaren, Craig Fordyce (J'41) (BODEFH), Lt., U.S.N.R.; 230 Corlies Ave., Allenhurst, N.J.
Haas, Charles Michael (J'39) (DNCB), Wire Rope Engr., Bethlehem Steel Co.; *for mail,* 691 Woodland Ave., Williamsport 19, Pa.
Haas, Edwin Henry (J'45), 34 Hawthorne Pl., Malverne, L.I., N.Y.
Habach, George Frederic ('29; '42) (HLBNGS), Ch. Engr., Centrifugal Engrg. Div., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 69 Osborne St., Glen Ridge, N.J.
Haber, Edward Harrison (J'40) (SC), Ensign, U.S.N.R.; *for mail,* 205 E. 23rd St., Wilmington 246, Del.
Haber, Harold E., Jr. ('32; '42) (CDL), Capt., U.S.A., Huntsville Arsenal, Huntsville, Ala.
Habicht, Ernst Rollemann ('26; '34; '35) (CLS), Plant Mgr., Belle Works, E. I. du Pont de Nemours & Co.; *for mail,* Box 98, Route 1, Charleston, W. Va.
Habicht, Frank Henry (J'45) (MJADC), Sales Dept., Marshall & Huschart Mchy. Co., 571 Washington Blvd., Chicago 6, Ill.
Hacker, John W. ('43) (C), Gen. Supt., Natl. Tube Co., Frick Bldg., Pittsburgh, Pa.
Hackett, Harold Nelson ('41) (SRLKFA), Constr. Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* R.D. 2, Ballston Lake, N.Y.
Hackett, Robert S. ('33; '42) (CLMORS), Engr., Personnel Dir., U.S. Guarantee Co., 90 John St., New York 7, N.Y.
Hacking, Chester ('28) (JM), Gen. Supt., Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Hackney, E. W. (J'39), Internatl. Harvester Co.; *for mail,* 816 Kinnaird Ave., Ft. Wayne, Ind.
Hackstaff, John Dudley ('01; '17) (OC), Cons. Engr., 520 Chapman Bldg., Los Angeles 14; *for mail,* 1661 S. Oxford Ave., Los Angeles 6, Calif.
Hacquet, Roland V. (J'35) (MCDZBJ), Asst. Ch. Engr., Universal Camera Corp., 28 W. 23rd St., New York 10, N.Y.; *for mail,* 8 Fairwood Rd., Madison, N.J.
Hadden, Arthur A. ('34) (CGMWL), Pres., McClure, Hadden & Ortman, Inc., 111 W. Washington St., Chicago 2, Ill.
Hadden, Cal. F. ('21), Secy., Treas., Stand. Supply & Hardware Co., P.O. Drawer 620, New Orleans 7, La.
Hadden, George Lewis ('45) (FKS), Prod. Supt., Ohio Pub. Serv. Co., Box 57, Shadyside, Ohio.
Haddock, Leander G., Jr. (J'41) (CDT), Ensign, U.S.N., Naval Air Sta., Moffett Field, Calif.
Haddy, John Frederick (J'45) (EJY), Jr. Engr., Engrg. Dept., St. Clair Processing Corp. Ltd., Sarnia, Ont., Can.
Hadley, G. Edwin (J'38) (BCD), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York, N.Y.
Hadley, Robert Campbell ('45) (SELOB), Mech. Engr., Stearns-Rogers Mfg. Co., 1720 California, Denver 2, Colo.
Hadley, Stanton Aaron ('21) (BEKOS), Partner, H & H Engrg. Serv., BMA Bldg., 215 W. Pershing Rd., Kansas City 8, Mo.
Hadley, William Alfred (J'42) (LKSBZY), Prof. Mech. Engrg., Pupin Bldg., Columbia Univ., New York 27, N.Y.
Hadlock, William Kenneth (J'46), 296 Pleasant St., Concord, N.H.
Hadlow, H. Ralph ('38) (SFKDHO), Ch. Engr., 715 Prospect Ave., Cleveland 15, Ohio.
Haentjens, Otto ('31) (HN), Pres. & Treas., Barrett-Haentjens & Co., Cedar St.; *for mail,* 51 James St., Hazleton, Pa.
Haentjens, Walter Dietrich (J'43) (HMBZ), Lt., Ord. Dept., Chief, Control Div., Cleveland Ord. Dist., 1036 Terminal Tower, Cleveland 13, Ohio.
Haesloop, Fred Louis, Jr. (J'40) (ABHCF), Stress Analyst, Douglas Aircraft Co., Inc., El Segundo; *for mail,* 3156 Hope St., Huntington Park, Calif.
Hagan, Albert Wilkin ('29; '35) (EBJ), Ch. Engr., Shoe Hardware Div., U.S. Rubber Co., 42 Brown St., Waterbury 89; *for mail,* 105 Plymouth St., Stratford, Conn.
Hagar, Edward Ford (J'41) (ABMCJD), Seaman, 1st Class, U.S.N.R., 1841 E. 4th St., Long Beach 4, Calif.
Hagblom, Edwin W. (J'39) (FKS), Serv. Engr., Foster Wheeler Corp., 165 Broadway, New York, N.Y.
Hage, S. Daniel (J'37) (AHK), Engr., Power Plant Analysis, Boeing Airplane Co.; *for mail,* 185—34th Ave. N., Seattle, Wash.
Hagemann, George Edward ('14; '19; '24) (CDMLJO), Engrg. Editor, Ronald Press Co., 15 E. 26th St., New York 10, N.Y.
Hagen, John F. ('30) (DCMTY), Prod. Engr., Callaway Mills, Inc., 295—5th Ave., New York 16; *for mail,* 405 W. 23rd St., New York 11, N.Y.
Hagensick, Robert L. (J'44) (KHBMNC), Design Engr., Engrg. Dept., Modine Mfg. Co., Racine, Wis.
Hagerman, Oliver Summers ('25; '35) (CFEBKJ), Box 1273, Charleston 25, W. Va.
Hagerty, Francis Willard (J'45) (AWY), Pres., Hagerty Co., Parker Ave.; *for mail,* Atlantic Ave., Cohasset, Mass.
Hagerty, Walter William ('05; '19), 2121 W. Market St., Pottsville, Pa.
Hagerty, William Walsh (J'45) (BAHN), Instr. in Engrg. Mechanics, College of Engrg., Univ. of Mich.; *for mail,* 2834 Whitewood St., Ann Arbor, Mich.
Haggerty, Charles ('41) (NBGM), Ch. Draftsman, New England Butt Co., 304 Pearl St., Providence 7; *for mail,* 197 Chace Ave., Providence 6, R.I.
Haggerty, Ridgeway T. (J'36), Indus. Engr., Ideal Novelty & Toy Co., 23-10—43rd Ave., Long Island City; *for mail,* 104-58—123rd St., Richmond Hill, N.Y.
Haglund, Gerhard O. (J'37) (BANCSZ), Sec. Head, Instrument Sec., Cornell Aero. Lab., Buffalo 5, N.Y.
Hagstrom, Jerome E. ('43) (BCJMN), V.P., Charge Engrg. & Research, Vendo Co., 1907 Grand Ave., Kansas City; *for mail,* 635 E. 66th St., Kansas City 5, Mo.
Hague, Charles Kenneth Felix ('32; '34) (CFS), Managing Dir., Babcock & Wilcox Ltd., Babcock House, Farringdon St., London, E.C. 4, England.
Hague, Floyd T. ('39) (BC), Mgr. of Engrg., Steam Div., Westinghouse Elec. Corp., Lester; *for mail,* 3700 Dermond Ave., Drexel Hill, Pa.
Hague, Robert Wilton ('45) (FKS), Project Engr., Tenn. Eastman Corp.; *for mail,* 3240 Ridgeview Rd., Kingsport, Tenn.
Hahn, Albert P. ('28; '35; '37) (ECSNDF), Sales Engr., Sterling Eng. Co., 2912 Russ Bldg., 235 Montgomery St., San Francisco 4; *for mail,* 3621 McLaughlin Ave., Los Angeles 34, Calif.
Hahn, Archie, Jr. (J'36) (DLS), Indus. Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail,* 816 Henry St., Herkimer, N.Y.
Hahn, Clifford Aylward ('20; '33) (CDLM), Pres., Holtzer Cabot Div., First Indus. Corp., 125 Amory St., Boston, Mass.
Hahn, Clifford Hitchcock (J'41) (CMDJ), 17 Chestnut St., Boston 8, Mass.
Hahn, Eugene ('29) (DFS), Pres., Hahn Equip. Corp., 30 Church St., New York 7, N.Y.
Hahn, Henry P. ('26; '35) (DEBRA), Export Mgr., Northwest Engrg. Co., 28 E. Jackson Blvd., Chicago 4, Ill.
Hahn, Mannel ('43) (GW), 370 Walnut St., Winnetka, Ill.
Hahn, Paul R. T. (J'38) (CJM), 515 Sherwood Pkwy., Westfield, N.J.
Hahn, Raymond P. ('33; '44) (KASWC), Sales Engr., Marley Co., Inc., 3001 Fairfax Rd., Kansas City, Kan.
Hahn, Robert S. (J'40) (BJNZ), Research Engr., Heald Mch. Co., 14 New Bond St., Worcester, Mass.
Hahn, William Francis ('40) (DJMR), Partner, W. F. Hahn Co., 1745 Blake St., Denver 2, Colo.
Haight, David LeRoy (J'45) (CDHKRS), Barnes Lane, White Plains, N.Y.
Haight, Harry Vercoe ('99; '07), 49 Portland Ave., Sherbrooke, Que., Can.
Haight, Robert Stanley ('39) (O), Ship & Engr. Surveyor, 90 West St., New York 6, N.Y.
Haigler, Edmund Derby ('37) (ZBLC), Mgr. Application Engrg., Foxboro Co., Foxboro, Mass.
Haile, William A. ('36) (CHKS), Sales Engr., Internatl. Div., Carrier Corp., 122 E. 42nd St., New York 17; *for mail,* Box 192, Scarborough, N.Y.
Hailer, Albert Joseph ('45) (EMS), Lt., Boiler Inspr., Bur. of Mar. Inspc. & Navigation, U.S.C.G., Appraisers Bldg., San Francisco; *for mail,* 25 Muirwood Dr., Daly City, Calif.
Hainer, Frederick William ('43) (SKF), V.P., Cleaver-Brooks Co., 326 E. Keefe Ave., Milwaukee 12, Wis.
Haines, Charles S., II ('43), Project Mgr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York 17; *for mail,* 150 Riverview Ave., Tarrytown, N.Y.
Haines, Edward Monroe (J'43) (CJM), Fred Goat Co. Inc., 314 Dean St., Brooklyn 17; *for mail,* 2 Guenther Court, Baldwin, L.I., N.Y.
Haines, Henry A. (J'40) (ACES), Indus. Serv. Rep., Stand. Oil Co. (Ind.), Box 30, St. Louis 8; *for mail,* 518 Taylor Ave., Flat River, Mo.
Hains, Charles F. (J'39) (HCOBW), U.S. Geological Survey, Box 178, Williamsburg, Ky.
Hainsworth, Bruce Alexander ('41) (LOKJDN), Engrg. Consultant, H. K. Ferguson Co., Hanna Bldg., Cleveland 15; *for mail,* 2924 Brighton Rd., Cleveland 20, Ohio.
Hait, J. M. ('39), Ch. Engr., Peerless Pump Div., Food Mchy. Corp., 301 W. Ave. 26, Los Angeles; *for mail,* 3560 Huntington Dr., San Gabriel, Calif.
Hajek, Ernest Emil (J'44) (KHCM), Chief Specialist (X) U.S.N.R., Aero. Engr., Div. N, Naval Air Sta., Moffett Field, Calif.

Hake, Robb Augustus ('28; '33) (CSB), Mgr., Lynn Sec., Serv. Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn, Mass.
Halbfass, Arnold Johannes ('45) (CMY), Cons. Engr., Apt. 20, 191 Adams St., Brooklyn 17, N.Y.
Halbmillion, Victor (J'43) (ABE), Pvt., 1st Class, Bldg. 322, Temperature Test Lab., Engr. Bd., Ft. Belvoir, Va.
Haldeman, Russell R. (J'33), Exper. Engr., De Laval Steam Turbine Co.; *for mail,* 833 Hamilton Ave., Trenton, N.J.
Haldeman, William Aquilla ('45) (CJK), Head, Design Gp., E. I. du Pont de Nemours & Co.; *for mail,* 38—3rd St., Ilion, N.Y.
Halden, Herbert Oliver ('25; '44) (EFS), Owner, H. O. Halden Co., 221 Plaza Theatre Bldg., Kansas City 2, Mo.
Hale, Floyd A. W. ('41) (CDBA), Engr., Louisville Cement Co., Speed, Ind.; *for mail,* 1236 Barret Ave., Louisville 4, Ky.
Hale, Herbert Zucker ('43) (WTCMOK), V.P., Astra Bent Wood Furniture Co., Inc., 61 W. 46th St., New York 19, N.Y.
Hale, Stephen C. ('22; '35) (SFTM), Ch. Engr., Fulton Bag & Cotton Mills, Box 1726, Atlanta 1; *for mail,* 1403 Emory Rd., N.E., Atlanta, Ga.
Hale, Wilburn ('45), Engrg. Supt., Sperry Gyroscope Co., Great Neck; *for mail,* 244-05 Van Zandt Ave., Douglaston, N.Y.
Haley, Robert L. (J'42) (CMOA), Processing Engr., McCormick Works, Internatl. Harvester Co., Chicago 8; *for mail,* 737 Hinman Ave., Evanston, Ill.
Hall, Albert Ernest ('44) (FSTWY), Dist. Engr., W. H. & L. D. Betz, Gillingham & Worth Sts., Philadelphia 24, Pa.; *for mail,* Box 406, Route 9, Birmingham 7, Ala.
Hall, Allen Strickland, Jr. (J'40) (BNHA), Instr. in Mech. Engrg., Mech. Engrg. Dept., Purdue Univ., Lafayette, Ind.
Hall, Alva Gage ('29; '35) (LY), Estimator, Dorr Co., Inc., 570 Lexington Ave., New York 22, N.Y.
Hall, Alvin Craig (J'44), Mech. Engr., Research & Devel. Lab., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail,* 62 N. Broad St., Woodbury, N.J.
Hall, Charles Milton, (J'40) (MNSBDJ), Bassick Co., 38 Austin St., Bridgeport; *for mail,* Box 103, Trumbull, Conn.
Hall, Clarence ('23; '30) (NMJLCF), Mech. Engr., Century Elec. Co., 1806 Pine St., St. Louis 3; *for mail,* 5618 Itaska St., St. Louis 9, Mo.
Hall, Clarence Albert (J'38) (EFAK), Engr., Ethyl Corp., 1600 W. Eight Mile Rd., Detroit 20, Mich.
Hall, Collier (J'39) (CJ), Engr., Mengel Co., 1122 Dumesnil St., Louisville 1; *for mail,* 1022 Rosemary Dr., Louisville, Ky.
Hall, Earl Eugene (J'46), 1196 Broadway, Beaumont, Tex.
Hall, Elmon L. ('25) (FL), V.P., Ch. Engr., Portland Gas & Coke Co., Pub. Serv. Bldg., Portland, Ore.
Hall, Frederick Folsom (J'45), Ensign, U.S.N.R., 1514 Beacon St., Brookline, Mass.
Hall, Gordon ('42) (MJLDK), Plant Engr., Wallace Barnes Co. Ltd., Sherman Ave. N., Hamilton; *for mail,* 29 Nelson Ave., Burlington, Ont., Can.
Hall, Harris Harlan (J'41) (BNJKOS), Engr. Asst., Design, Humble Oil & Refining Co., Box 950, Baytown; *for mail,* 626 Marion St., Port Neches, Tex.
Hall, Harry ('26; '32; '35) (EBJNMF), Asst. Ch. Engr., Engrg. Devel., Worthington Pump & Mchy. Corp., Clinton St. & Roberts Ave., Buffalo 6; *for mail,* Box 953, Buffalo 5, N.Y.
Hall, Herbert Hudson ('24; '35) (JDCALM), Aluminum Co. of Am., Gulf Bldg., Pittsburgh; *for mail,* 76 Altadena Dr., Mt. Lebanon, Pittsburgh 16, Pa.
Hall, Jesse Harbour (J'39) (AECBK), Asst. to Exec. Engr., Mech. Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport; *for mail,* 4309 W. 66th, Cleveland 9, Ohio.
Hall, John G. ('28; '35) (FKSC), Asst. to V.P., Combustion Engrg. Corp., Ltd., 69 Yonge St., Toronto 1, Ont., Can.
Hall, John L. (J'40) (AYZ), Asst. Project Engr., Propeller Div., Curtiss-Wright Corp., Caldwell, N.J.
Hall, John M. (A'41) (KRS), Dir., Loco. Inspc., U.S. Govt.; *for mail,* 7605 Morningside Dr., N.W., Washington, D.C.
Hall, John Robert ('23; '35) (FSKHZ), Foreman, Boiler House, Socony-Vacuum Oil Co., Inc., 3821 Indianapolis Blvd., East Chicago; *for mail,* 301 Park Ave., Lafayette, Ind.
Hall, Joseph Marchand ('33) (RLY), V.P., Cardwell Westinghouse Co., 332 S. Michigan Ave., Chicago 4, Ill.
Hall, Lewis Wray (J'44) (ZNKAS), Aviation Radio Technician, 3rd Class, U.S.N., Shore Patrol Hdq., Union Sta., McAlester; *for mail,* 1442 N. Elgin, Tulsa 6, Okla.
Hall, Norman Macleod ('21) (OSF), Prof. Mech. Engrg., Univ. of Manitoba, Winnipeg, Man., Can.
Hall, Otis Nathaniel (J'41) (SCFKHJ), Asst. Sta. Supt., Va. Elec. & Power Co., Hampton Power Sta.; *for mail,* Box 352, Route 2, Hampton, Va.
Hall, Peter P.-G. ('27), Pres., Hall Planetary Co., Fox St. & Abbotsford Ave., Philadelphia, Pa.
Hall, R. Benson ('19; '34) (MBNZ), Mech. & Elec. Engr., Hall Serv. Co., Box 3, Calumet City, Ill.; *for mail,* 6431 Euclid Ave., Hammond, Ind.
Hall, Ralph E. ('30) (FKLMST), Dir., Hall Labs., Inc., Box 1346, 323—4th Ave., Pittsburgh 30, Pa.
Hall, Richard Wax (J'42) (CJM), Asst. Foreman, Natl. Mchy. Co.; *for mail,* 91½ Main St., Tiffin, Ohio.
Hall, Robert Everett ('98; '05) (CHOS), Retired; 1017 Madison Ave., Plainfield, N.J.
Hall, Robert Louis (J'42) (MC), 1st Lt., U.S.M.C., Radar Officer, AWS-17, Cherry Point, N.C.; *for mail,* 7146 S. Benton, Kansas City 5, Mo.
Hall, Rodney Dennis ('02; '08) (HES), Asst. to V.P., Utilities, Worthington Pump & Mchy. Corp., Harrison, N.J.
Hall, Stanley Rowland ('35; '44) (CHMN), Mech., Design Engr., Lockheed Aircraft Corp.; *for mail,* 1004 N. Lincoln St., Burbank, Calif.
Hall, Walter Atwood ('11) (CLHSOB), Engr., Reconstruction Finance Corp., 10 Post Office Sq., Boston 9; *for mail,* 15 Puritan Rd., Swampscott, Mass.
Hall, William S. ('25; '33; '35) (C), Housing Mgmt. Adviser, Federal Pub., Housing Authority, Georgia Savings Bank Bldg., Atlanta 3; *for mail,* 184 Vidal Blvd., Decatur, Ga.
Halladay, Harry F. ('03; '13) (RJG), Mech. Engr., N.Y. Air Brake Co., Starbuck Ave.; *for mail,* 1012 Washington St., Watertown, N.Y.
Hallaman, Chas. G. (J'35), Magnolia Petroleum Co., Box 900, Dallas 1; *for mail,* Box 90, Route 1, Dallas, Tex.
Hallas, Leon Gary (J'44) (BCDMNO), Mech. Design Engr., West. Cartridge Co., East Alton; *for mail,* 604 W. High St., Edwardsville, Ill.
Hallberg, Elmer Edward ('44) (CHBJMN), Works Mgr., Rockford Mch. Tool Co., 2500 Kishwaukee St.; *for mail,* 1214—23rd Ave., Rockford, Ill.
Halldorsson, Gisli ('44) (BKM), Pres., Gisli Halldorsson, Ltd., Austurstraeti 14; *for mail,* Flokagata 6, Reykjavik, Iceland.
Hallenbeck, George Elliott ('08) (NHJCML), Pres., Baker Bros. Inc., 1000 Post St.; *for mail,* 2702 Parkwood Ave., Toledo 10, Ohio.
Hallenbeck, Thomas Lewis (J'37) (NBJHML), Plant Engr., Baker Bros., Inc., 1000 Post St., Toledo 10, Ohio.
Haller, Henry E. ('20) (BHJLS), Pres., Natl. Valve & Mfg. Co., 3101 Liberty Ave., Pittsburgh 1; *for mail,* 415 S. Pacific Ave., Pittsburgh 24, Pa.
Haller, Karl Raymond (J'23) (OCM), Owner, Keystone Auto Body Co., 1415 Spring Garden St., Philadelphia, Pa.
Haller, Louis G. ('17; '22; '28; F'45), (BKLSTY), Supt. of Engrg., Tenn. Eastman Corp., Kingsport; *for mail,* 703 E. Watauga Ave., Johnson City, Tenn.
Haller, Oliver Jacob ('36) (S), Project Engr., Pittsburgh Piping & Equip. Co., 10—43rd St., Pittsburgh 1, Pa.
Haller, Robert F. (J'41), Babcock & Wilcox Co., Candler Bldg., Atlanta, Ga.
Haller, Rudolf Vincent (J'35) (SFZ), Assoc. Mech. Engr., Tenn. Valley Authority, Power Bldg., Chattanooga; *for mail,* Route 1, Ooltewah, Tenn.
Hallinan, John C. (J'40) (BME), 605 Linden Ave., Riverton, N.J.
Halliwell, Arthur ('24; '35) (BCM), Office of Civilian Personnel, Cent. Pacific Base Command, A.P.O. 958, c/o Postmaster, San Francisco, Calif.
Hallock, Homan F. ('10; '16; '30) (LDF), Ch. Engr., Diamond Match Co., Oswego, N.Y.
Hallock, Richard Wheeler (J'46) (DLO), Engr., Gen. Elec. Co., Schenectady; *for mail,* 64 Vley Rd., Scotia, N.Y.
Halloran, Joseph Michael (J'45) (CMJZKN), Mgr., Furnace Div., A. F. Holden Co., 52 Richards St., West Haven 16, Conn.
Halloran, Ralph A. ('18; '35) (EF), Calif. Research Corp., 200 Bush St., San Francisco 4, Calif.
Hallum, Thomas E. (J'39) (CEJ), R.D. 4, Troy, N.Y.
Halpern, Benj. M. ('21; '26), Partner, Mgr., Gibson & Halpern, 15 Park Row, New York, N.Y.
Halpern, William (A'44) (NMB), Owner, Wm. Halpern & Co., 53 Park Pl., New York 7, N.Y.
Halpin, Carl L. ('29; '35) (CDMOZH), Cons. Engr., 1216 Harvard Rd., Grosse Pointe Park 30, Mich.
Halporn, Arnold Davidson ('44) (EFKS), Sales Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York; *for mail,* 310 W. 85th St., New York 24, N.Y.
Halsey, William Darrach ('16; '18; '23) (SFOKLZ), Ch. Engr., Boiler Div., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford 2, Conn.
Haltenhof, Harold Roy (J'45), Tech. Analyst, Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 700 E. 141st St., New York 54, N.Y.
Halvorsen, Robert Alfred (J'40) (CMJ), Secy., Treas., Sterling Die Casting Co., Inc., 743—39th St., Brooklyn 32, N.Y.
Halzel, George Charles (J'45) (SDFKZH), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 11 Colonial Ave., Dorchester 24, Mass.
Ham, Clarence W. ('16) (NBDMEH), Prof. Mch. Design, Univ. of Ill., 113 Trans. Bldg., Urbana, Ill.
Ham, Merrill Turner (J'44) (KEF), Sgt., Corps of Engrs.; 41 Western Ave., Augusta, Me.
Hamaker, Frank Maurice (J'44), 2230 Haste St., Berkeley 4, Calif.
Hamaker, John Donald (J'40) (WJMCD), 544 E. Jefferson Ave., Kirkwood 22, Mo.
Hamann, Charles Henry, Jr. (J'45) (SRZBKN), Jr. Engr., Union Elec. Power Co., 315 N. 12th St., St. Louis; *for mail,* 4663 Maffitt Ave., St. Louis 13, Mo.
Hamblet, Geo. W. ('21) (CDM), Propr., Hamblet Mch. Co., 30 Island St., Lawrence, Mass.
Hambleton, Warner VanMurray (J'43) (SFKJBO), Asst. Ch. Engr., Boiler Div., Henry Vogt Mch. Co., 1000 Ormsby St., Louisville 10; *for mail,* 1248 S. Jackson St., Louisville, Ky.
Hamel, Clarence G. (J'27) (SFHKZ), Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Place, New York 3; *for mail,* 45 Hawthorne Pl., Manhasset, N.Y.
Hamer, Ian Malcolm (J'42) (HACJNM), Ch. Engr., Dowty Equip. (Canada) Ltd., 999 Aqueduct St., Montreal 3, Que., Can.
Hamil, James Kenney ('44) (BCGLMN), Mgr. Mch. Devel., Remington Arms Co., Inc., 939 Barnum Ave., Bridgeport; *for mail,* 591 S. Compo Rd., Westport, Conn.
Hamil, John Clement (J'42) (CEM), Box 112, Troy, Ala.
Hamill, John Richard (J'37) (OLSCHK), Sales Engr., Worthington Pump & Mchy. Corp., 2905 N. Broad St., Philadelphia 32, Pa.
Hamill, John Stafford (J'38) (MCJ), Supervisory Machinist, Naval Gun Factory, Washington 3; *for mail,* 4617—5th St., N.W., Washington 11, D.C.
Hamill, Samuel M., Jr. ('41), Asst. Supt., Elec. Oper. Dept., Cincinnati Gas & Elec. Co., Box 356, Cincinnati 1, Ohio.
Hamill, Thomas (J'40) (LBKSZY), Engr., Farrel-Birmingham Co., Inc., 25 Main St., Ansonia; *for mail,* 50 Mayflower Pl., Milford, Conn.
Hamilton, Allan Robert (J'45) (CJSK), Mech. Engrg. Dept., Can. Wire & Cable Co., Ltd., Laird Dr., Leaside; *for mail,* 118 Armstrong Ave., Toronto 4, Ont., Can.
Hamilton, Arthur Stuart, Jr. ('35; '45) (NLMCBD), Cons. Engr., 83 Westland Ave., Rochester 13, N.Y.
Hamilton, Charles Andrew ('20; '24) (SFHOKW), Sr. Mem., Hamilton-Weeber, 354 Houseman Bldg., Grand Rapids 2, Mich.
Hamilton, Chester B., Jr. ('09; '14) (MJC), Pres., Hamilton Gear & Mch. Co., 62 Van Horne St., Toronto 4, Ont., Can.
Hamilton, Donald B. (J'36) (CM), 7819 S. Peoria St., Chicago 20, Ill.
Hamilton, Douglas Thomas ('16; '20) (BMN), Publicity Mgr., Fellows Gear Shaper Co., 78 River St.; *for mail,* 100 Summer St., Springfield, Vt.
Hamilton, Earl Baten (J'44), 229 E. Lewiston Rd., Dayton, Ohio.
Hamilton, Eugene Dale (J'45) (HLY), 1245 Everett St., El Cerrito, Calif.
Hamilton, Geo. S. (J'37) (ABG), Design Engr., Ford Instrument Co., Rawson St. & Nelson Ave., Long Island City; *for mail,* 113 Mowbray Pl., Kew Gardens, L.I., N.Y.
Hamilton, James C. (J'43) (CZDEMJ), Design Engr., Hardy Mfg. Corp., Pendleton; *for mail,* 1412 Meridian St., Anderson, Ind.
Hamilton, John Simpson, Jr. (J'45), c/o A. R. Melick, 2023 S. 6th, Allentown, Pa.
Hamilton, John Stuart (J'40) (ACM), Field Serv. Engr., Boeing Aircraft Co., Seattle 14; *for mail,* College Club, 6th & Spring, Seattle, Wash.
Hamilton, Myron B. (J'43) (KLEYWR), Sales Engr., Ross Heater & Mfg. Co., Inc., 1407 West Ave., Buffalo 13; *for mail,* 1679 Amherst St., Buffalo 14, N.Y.
Hamilton, Newell ('41) (BCDEFG), Supt., Steel Mill, Babcock & Wilcox Tube Co., Beaver Falls, Pa.
Hamilton, Robert Compton (J'42) (ACE), 616 Puritan Rd., Birmingham, Mich.
Hamilton, Samuel Lewis (J'41) (KOZHLC), Design Engr., Fla. Hill York Corp., 1225 S.W. 8th St., Miami; *for mail,* Apt. 2, 33 S.W. 11th Ave., Miami 36, Fla.
Hamilton, Stuart (J'42) (ESAKB), 240 Montclair Ave., Newark 4, N.J.
Hamilton, T. Hayden ('27), Box 113, Coytesville, N.J.
Hamilton, Thomas Patrick ('42) (HCMR), Planning Supvr., Hyd., N.Y. Air Brake Co.; *for mail,* 147 Thompson Blvd., Watertown, N.Y.
Hamilton, W. B. ('12) (HMBC), Sales & Prod. Engr., Hardie-Tynes Mfg. Co., Birmingham 1, Ala.
Hamilton, William Edward ('24) (FDEHS), Partner, Sanderson & Porter, 52 William St., New York, N.Y.
Hamilton, William Floyd ('27; '34; '35) (YKBT), Dir. of Engrg., Ensign-Bickford Co.; *for mail,* Massaco St., Simsbury, Conn.
Hamilton, William I. (J'35) (ENBFOC), Ch. Engr., Lanova Corp., 27-01 Bridge Plaza N., Long Island City 1; *for mail,* 53-146—63rd St., Maspeth, L.I., N.Y.
Hamilton, Wm. J. ('05) (C), Secy., Hendrick Mfg. Co., 51 Dundaff St., Carbondale, Pa.

Hamilton, William S. H. ('45) (REFK), Equip. Elec. Engr., N.Y. Cent. R.R., 466 Lexington Ave., New York 17; *for mail,* 65 Edgewood Ave., Larchmont, N.Y.
Hamlen, Earle Kenneth ('42) (NGMJ), 223 Birchwood Ave., Cuyahoga Falls, Ohio.
Hamlin, Charles P. (J'37) (CEF), Gas Engr., Gas Serv. Co., York Rite Bldg., Wichita, Kan.
Hamlin, Eugene E., Jr. (J'46), Box 227, Balboa, C.Z.
Hamlin, Perley Chamberlain (J'44) (CMDEF), Sr. Indus. Engr., Natl. Supply Co., Border St., Torrance; *for mail,* 10301 Mississippi Ave., Los Angeles, Calif.
Hamlin, William Fred (J'41) (JMNKYR), Ch. Engr., Model Engrg. Corp., Box 635, Bristol, Tenn.
Hamm, Hans W. (J'42) (BHNS), Design Engr., S. Morgan Smith Co.; *for mail,* 515 Linden Ave., York, Pa.
Hamm, James Robert (J'43) (BAFKSH), Design Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester; *for mail,* 23 Princeton Ave., Swarthmore, Pa.
Hammarlund, Arthur Ervin (J'43) (ABS), Ensign, U.S.N.R., U.S.S. *Gillis,* AVD-12, F.P.O., San Francisco, Calif.
Hammarstrom, Erik ('19) (SKLWHD), Pur. Engr., W. Va. Pulp & Paper Co., 230 Park Ave., New York 17, N.Y.
Hammell, Reeve Howard ('24; '31; '35) (EKNF), Mech. Engr., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail,* 198 Richey Ave., West Collingswood, N.J.
Hammer, Alexander ('44) (BKS), Research Engr., De Laval Steam Turbine Co., Nottingham Way, Trenton 2; *for mail,* 42 Maple Ave., Trenton 8, N.J.
Hammer, Edwin Wesley ('13) (ABLNRS), Cons. Engr., 80 John St., New York 7, N.Y.
Hammer, James E. (J'40) (CS), 1st Lt., 250th Field Arty. Bn., Camp Maxey, Tex.
Hammers, William Scott, Jr. (J'37) (MCDLR), 715 Webster St., N.W., Washington 11, D.C.
Hammershaimb, George T. ('40) (ERAFSB), Devel. Engr., Eddystone Plant, Baldwin Loco. Works, Philadelphia 42; *for mail,* 44 Green Valley Rd., Wallingford, Pa.
Hammerstein, Harold K. ('29; '35) (JBM), Mech. Engr., Broderick & Bascom Rope Co., 4203 N. Union Blvd., St. Louis 15, Mo.
Hammerstone, James E. (J'40), 257 Kleinhans Ave., Easton, Pa.
Hammett, George Robert ('20; '29; '35) (CHS), Mgr., A. M. Lockett & Co., Ltd., 308 Whitney Bldg., New Orleans 7, La.
Hammett, Philip Melancthon ('10) (RSH), Retired; P.O. Box, Mandarin, Fla.
Hammett, Urquhart Archer ('37) (SLECYZ), Dist. Sales Mgr., Permutit Co., 330 W. 42nd St., New York 18, N.Y.; *for mail,* 2605 Lorain Rd., San Marino 9, Calif.
Hammond, Caleb Dean, Jr. (J'38) (GC), Lt. (j.g.), U.S.C.G., U.S.S. *Hunter Liggett,* F.P.O., San Francisco, Calif.
Hammond, Edward K. ('19) (CDHJMN), West. Mgr., Indus. Press, 228 N. La Salle St., Chicago 1, Ill.
Hammond, Harold Marcus ('19; '23; '35) (FSZLCR), Gen. Sales Mgr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail,* 15706 Hazel Rd., East Cleveland 12, Ohio.
Hammond, Harry Parker ('37) (HBM), Dean, Sch. of Engrg., Pa. State College, State College, Pa.
Hammond, James David (J'45), Ensign, U.S.N.R., Naval Supply Operational Trng. Center, Naval Supply Depot, Bayonne Annex, N.Y. Naval Shipyard, Bayonne, N.J.; *for mail,* 542 Lafayette Ave., Palmerton, Pa.
Hammond, John R., Jr. (J'40) (HNSCEY), Mech. Designer, Office Engrg. Div., Dept. of Opera. & Maint., The Panama Canal, Balboa Heights; *for mail,* Box 431, Diablo Heights, C.Z.
Hammond, Nelson Browne (J'44) (ESKNFH), *Student Award,* '44; Jr. Engr., Philco Corp., C & Allegheny, Philadelphia; *for mail,* 4951 Walnut St., Philadelphia 39, Pa.
Hammond, S. Irving ('38) (FKL), Mech. Engr., N.J. Zinc Co.; *for mail,* 542 Lafayette Ave., Palmerton, Pa.
Hampton, Flournoy W. (J'35) (NBDCM), Pres., Hampton Enterprises, 5135 University Ave., Chicago 15, Ill.
Hampton, Leon N. ('23; '35), Engrg. Supvr., Bell Tel. Labs., Inc., 463 West St., New York 14; *for mail,* 1895 Grand Concourse, New York, N.Y.
Hanauer, Elbert A. (J'32), 92-16—195th Pl., Hollis 7, L.I., N.Y.
Hanauer, Sylvan Leonard ('25; '34; '35) (OYSKLC), V.P., Gen. Realty & Utilities Corp., 285 Madison Ave., New York; *for mail,* 315 E. 68th St., New York 21, N.Y.
Hancock, Clifford Harold ('42) (HBZANG), Supvr., Hyd. Lab., Newport News Shipbldg. & Dry Dock Co., Newport News, Va.
Hancock, James Edwin ('39) (COYG), Mgr., Maint. & Constr., Shawmut Realty Trust, 15 Norway St., Boston, Mass.
Handa, Chuni Lal ('45) (BDH), Rai Bahadur, I.S.E.M.I.E., c/o The Registrar, Punjab Irrigation Secretariat, Lahore, Punjab, India.
Handel, Richard Robert (J'43), Power Tube Standardizing Engr., RCA Victor Div., Radio Corp. of Am.; *for mail,* 415 S. Queen St., Lancaster, Pa.
Handlos, Alphonse A. (J'41) (MC), Prod. Engr., Vilter Mfg. Co., 2217 S. 1st St., Milwaukee 7; *for mail,* 1133 N. 24 Pl., Milwaukee 3, Wis.
Handman, Seymour (J'45) (BOS), 1710—86th St., Brooklyn 14, N.Y.
Handman, Stanley Edward (J'44), 1372 Bristow St., Bronx 59, N.Y.
Hands, Ronald Clive ('37) (CAM), Lt. Col., Ord. Dept., U.S.A., Mercantile Bldg., Rochester 4; *for mail,* 270 Fair Oaks Ave., Brighton 4, N.Y.
Haney, Glenn Earl ('35; '35) (SFEK), Supt., Power Plant, Ohio State Univ., N. High St., Columbus 10; *for mail,* 2005 Berkshire Rd., Columbus 8, Ohio.
Haney, Harold B. (J'40) (BCJLNY), Dist. Sales Engr., Haveg Corp., 550 Leader Bldg., Cleveland 14, Ohio.
Haney, Jiles William ('14; '21; F'36) (SEBAFK), *Manager,* '34-'37; Col., Hdq., 64th QM. Base Depot, A.P.O. 350, c/o Postmaster, New York, N.Y.; *for mail,* 2521 Capitol Ave., Cheyenne, Wyo.
Hangarter, Andrew Joseph (J'37) (CML), Exec. Asst., Andrew C. Roesch, Inc., 179-81 Powers St., Brooklyn; *for mail,* 47-06 Fresh Meadow Lane, Flushing, N.Y.
Hanger, Samuel Ryland ('30; '35) (SEFJZH), Jr. Engr., Asst. Supvr., Philadelphia Elec. Co., 2301 Market St., Philadelphia 3, Pa.
Hanger, Wallace Sherwood (J'39) (CJ), 2nd Lt., Field Arty. Reserve, U.S.A.; *for mail,* 1102 Pontiac St., Denver 7, Colo.
Hangey, Theodore Gross (J'45) (FJMKBD), Ch. Draftsman, Charge of Design, Roberts & Mander Stove Co., Jacksonville Rd., Hatboro; *for mail,* 6603 N. Opal St., Philadelphia 38, Pa.
Hango, John (J'40) (FKS), Serv. Dept., Foster Wheeler Corp., 165 Broadway, New York; *for mail,* 8918—133rd St., Richmond Hill, L.I., N.Y.
Hanhart, Ernest Henry ('33; '43) (OLMD), Cons. Engr., 1311 Ramblewood Rd., Baltimore 12, Md.
Hanke, Harold (J'39) (CFK), 225 N. Cuyler Ave., Oak Park, Ill.
Hanken, George Henry (J'45) (BJY), 593 Lexington Ave., Rochester, N.Y.
Hankes, Elmer Joseph ('41; '45) (ONMCSZ) Owner, Engrs. Northwest, 2620 Portland Ave. S., Minneapolis 7, Minn.
Hankins, Cyrus ('40) (RJB), V.P., Unitcast Corp., 2237 Waterworks Dr., Toledo 9, Ohio; *for mail,* 832 Munsey Bldg., Washington 4, D.C.
Hankison, Lewis E. ('16; '26) (SFBKZ), Supt., Effic. Dept., West Penn Power Co., 14 Wood St., Pittsburgh; *for mail,* 1301 Biltmore Ave., Pittsburgh 16, Pa.
Hanks, George Jason, Jr. (J'42) (EKYZJH), Tech. Asst., Carbide & Carbon Chemicals Corp., McCorkle Ave., South Charleston; *for mail,* 1595 Quarrier St., Charleston 1, W.Va.
Hanks, George Raymond ('24) (CJLRZB), Pres., Taylor-Wharton Iron & Steel Co., High Bridge, N.J.
Hanley, Edward (J'42) (BCH), Draftsman, Goodman Mfg. Co., 4834 S. Halsted St.; *for mail,* 6653 S. Claremont Ave., Chicago, Ill.
Hanley, William A. ('13; '20; F'36) (CLS), *Manager,* '27-'30, *Vice-President,* '30-'32, *President,* '41; V.P., Eli Lilly & Co., Indianapolis 6, Ind.
Hanna, John Holger (J'39) (ABH), Bechtel Bros. & McCone, 220 Bush St., San Francisco; *for mail,* 15 Arguello Blvd., San Francisco 18, Calif.
Hanna, John Hunter, Jr., (J'39) (SFCKJ), Plant Engr., Potomac Elec. Power Co., 10th & E Sts., N.W., Washington 4; *for mail,* 3009 Q St., N.W., Washington 7, D.C.
Hannah, Robert Bruce ('45) (AJN), Protection Engr., Mountain States Tel. & Tel. Co., 931—14th St., Denver 2; *for mail,* 637 S. Vine St., Denver 9, Colo.
Hannan, Raphael Q. ('22; '24; '35), Inspr., Iron & Steel, Richmond County, City of N.Y.; *for mail,* 360 Drumgoole Blvd., Staten Island 9, N.Y.
Hanneman, Frank (J'39) (JMR), Engr., Paasche Airbrush Co., 1909 Diversey Pkwy.; *for mail,* 1421 Sherwin Ave., Chicago, Ill.
Hanneman, Richard Louis (J'42) (A), Asst. Mfg. Engr., Hawthorne Works, West. Elec. Co., Inc., 23rd Pl. & Cicero Ave., Chicago; *for mail,* 3849 N. Pittsburgh, Chicago 34, Ill.
Hannewald, Burton August (J'26) (GJMC), Supt., Mch. Shop, Bemis Bros. Bag Co., 1940 Barth Ave., Indianapolis 7; *for mail,* 821 N. Audubon Rd., Indianapolis 1, Ind.
Hannold, John Robert (J'44) (CYMLN), Asst. to Pres., Welding Engrs., Inc., Box 391, Norristown, Pa.
Hannon, William W. (J'40) (GWTMC), Prod. Engr., West. Automatic Mch. Screw Co., Foster & Lake Aves.; *for mail,* 127 Harwood St., Elyria, Ohio.
Hannum, Charles M. (J'32) (ABM), Capt., Hdq. Depot 0-642, A.P.O. 582, c/o Postmaster, New York, N.Y.
Hannum, Joshua Eyre ('19; '35), (CMA), Dean of Engrg., Ala. Poly. Inst.; *for mail,* 425 E. Magnolia Ave., Auburn, Ala.
Hanny, Rupert M. ('36) (LSJB), Mgr., Sales & Engrg., Lenape Hyd. Pressing & Forging Co., Box 536, West Chester, Pa.
Hanrahan, Frank John ('42) (WBJADY), Deputy Dir. of Lumber Stands. Div., Natl. Lumber Mfrs. Assn., 1319—18th St., N.W., Washington 6, D.C.
Hanscom, Geoffrey Leonard ('39) (HBC), Devel. Engr., Stand. Mchy. Co., 1475 Elmwood Ave., Providence 7, R.I.; *for mail,* 8 Glendale Rd., Sharon, Mass.
Hanscom, William W. ('13) (BFL), Cons. Engr., 848 Clayton St., San Francisco 17, Calif.
Hansen, Alf ('28; '37) (SKFCO), Dist. Turbine Engr., Gen. Elec. Co., 235 Montgomery St., San Francisco 6, Calif.
Hansen, Anton ('14; '35) (MBCJ), Mech. Engr., Wm. Bergfels Co., 10-14 Orchard St., Newark 2; *for mail,* 818 DeGraw Ave., Newark 4, N.J.
Hansen, Edward H. ('36) (GCTMAY), Book Mfg. Dir., Crowell-Collier Publ. Co., 250 Park Ave., New York 17; *for mail,* 1—4th Rd., Great Neck, N.Y.
Hansen, Ellis Peter ('44) (NBHERK), Design Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., 1126 S. 70th St., Milwaukee 1; *for mail,* 2929 N. 72nd St., Milwaukee 10, Wis.
Hansen, George Albert (J'44) (FS), Steam Engr., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh, Pa.
Hansen, Hans (J'41) (ACM), Project Engr., Am. Type Founders, Inc., 200 Elmora Ave., Elizabeth; *for mail,* 134 E. 7th Ave., Roselle, N.J.
Hansen, Hans I. ('30; '37) (CBMNOD), Engr., Deere & Co., Moline, Ill.; *for mail,* 114 W. Central Park Ave., Davenport, Iowa.
Hansen, Herbert Nathon (J'42), 229 N. Grove Ave., Elgin, Ill.
Hansen, Holger Hilmer ('26) (SRBAC), Ch. Engr., Gen. Supt., Antonio Roig Sucrs.; *for mail,* Central El Ejemplo, P.O. Box 205, Humacao, P.R.
Hansen, Merlin ('30; '39) (BEJGN), Asst. Ch. Engr., John Deere Tractor Co.; *for mail,* 1123 Western Ave., Waterloo, Iowa.
Hansen, Robert (J'41) (ABSKJ), Draftsman Engr., Otis Elev. Co., 260—11th Ave., New York; *for mail,* 280 Parkside Ave., Brooklyn, N.Y.
Hansen, Sam S. ('38; '41) (EF), Ch. Indus. Engr., Gen. Petroleum Corp., 108 W. 2nd St., Los Angeles 12; *for mail,* 4551 Round Top Dr., Los Angeles 41, Calif.
Hansen, Viggo ('41) (SHBJ), Sr. Engr., Philadelphia Elec. Co., 17th Floor, 900 Sansom St., Philadelphia 5, Pa.
Hansen, Walter Roland ('45), Mech. Supt., U.S. Smelting, Refining & Mining Co., 1120 Newhouse Bldg.; *for mail,* 839 East South Temple, Salt Lake City, Utah.
Hansen, William Oliver (J'38) (BGHKNS), Mech. Engr., Creole Petroleum Corp., Apartado Postal 172, Maracaibo, Venezuela, S.A.
Hanson, Clifford George (J'42) (BAKCH), Prod. Engrg. Liaison, Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 17 Munroe St., Lynnfield, Mass.
Hanson, Frederick Herman, Jr. (J'44) (AFCSYB), Mech. Engr., Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field; *for mail,* 2449 Russell St., Berkeley 5, Calif.
Hanson, Harold Franklin ('28; '41) (BCNMJL), Mech. Engr., Metals Protection Co. of Pa., 7424 Tioga St., Pittsburgh; *for mail,* 623 Whitney Ave., Wilkinsburg 21, Pa.
Hanson, Hubert Chester (J'35) (SFZ), Capt., Trans. Corps, U.S.A., Ft. McDowell, Angel Island, Calif.
Hanson, Karl P. ('34; '39) (EFKS), Prof. Mech. Engrg., Univ. of Conn., Storrs, Conn.
Hanson, L. C. ('31; '35; '35) (BJL), Mech. Engr., Solvay Process Co.; *for mail,* 410 Prince George Ave., Hopewell, Va.
Hanson, Lennart Lars (J'42) (SZNCLO), Engr., Design, E. I. du Pont de Nemours & Co.; *for mail,* 531—25th St., Niagara Falls, N.Y.
Hanson, Milton Emmons ('19; '27) (GJKNR), Mech. Engr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., Cresmont & Hadden Ave., Camden; *for mail,* 120 Gill Rd., Haddonfield, N.J.
Hanson, Ralph A. (J'40) (AGM), Engrg. Draftsman, Boeing Aircraft Co., Georgetown Sta., Seattle; *for mail,* 11508—5th Ave., N.E., Seattle 55, Wash.
Hanson, Raymond F. ('25; '32; '35) (FCZKSO), Power Engr., Gen. Motors Corp., Gen. Motors Bldg., Detroit 2; *for mail,* 14253 Freeland Ave., Detroit 27, Mich.
Hanssen, Albert Jacob (J'39) (ZNMLCE), Asst. Ch. Engr., McAlear Mfg. Co., Automatic Control Div. of Climax Industries, 15 N. Cincinnati, Tulsa 1; *for mail,* 3803 E. 22nd St., Tulsa 4, Okla.
Hansson, Axel Sigfrid ('32) (BCNRSY), ASEA (Allmanna Svenska Elektriska Aktiebolaget), Västerås, Sweden.
Hanzlik, Henry John ('03; '10) (OSNKBJ), Cons. Engr., 315 Cornell Ave., Swarthmore, Pa.
Happel, Albert W. ('28) (BJM), Design Engr., Lynch Corp., Anderson, Ind.
Happel, Hermann E. (J'37) (BCJLMA), Warrant Officer (j.g.), 219th Ord. Heavy Maint. Co. (Tank), Camp Bowie, Tex.
Hara, Edward E. (J'41), 330 Riverview Ave., Drexel Hill, Pa.
Harakawa, Harvey H. (J'44) (AEF), 169 Paoakalani St., Honolulu 30, T.H.
Harazim, Stanley J. (J'27) (BCMNZ), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14, N.Y.
Harbeson, James Page, Jr. ('18; '35) (JENRSF), Ch. Engr., Camden Forge Co., Camden, N.J.

Harbin, Charles Ben ('42) (BJMNY), Pres., Acme Mch. & Welding Co., 327-29 Edgewood Ave., Atlanta 3; *for mail,* 852 Piedmont Ave., Atlanta, Ga.
Harbke, Harland Charles (J'43) (BJZS), Prod. Engr., Haggart Mfg. Co., Box 52, Route 2, Oregon City; *for mail,* 8602 S.E. Taylor St., Portland 16, Ore.
Harclerode, John Robert (J'45) (MONJC), Ch. Draftsman, Black, Sivalls & Bryson, Inc., 7500 E. 12th St., Kansas City 3; *for mail,* 2331 Oakley, Kansas City 1, Mo.
Hardaway, Henry Z. (J'46), Bell Tel. Labs., Whippany, N.J.
Hardaway, Warren Dunham ('29) (CHS), Supt., Hydroelec. Prod. & Transmission, Pub. Serv. Co. of Colo., 900—15th St., Denver 1, Colo.
Hardgrave, John Coyne ('30; '35) (M), Assoc. Prof. Mech. Engrg., Tex. Tech. College; *for mail,* 2418—20th St., Lubbock, Tex.
Hardgrave, Robert L. (J'37), So. Aircraft Corp., Garland; *for mail,* 6226 Revere Pl., Dallas 14, Tex.
Hardgrove, George A., Jr. (J'42), Lt. (j.g.), 10200 Unity Ave., Cleveland 11, Ohio.
Hardgrove, Ralph M. ('19; '25) (FSRK), Research Devel. Consultant, Babcock & Wilcox Co., 85 Liberty St., New York, N.Y.
Hardie, Philip Henry ('23; '29; '35) (SFKBE), Mech. Engr., Sanderson & Porter, 52 William St., New York 5, N.Y.
Hardin, Frank Hammond ('21) (CJR), Pres., Association of Manufacturers of Chilled Car Wheels, Rm. 1556, 230 Park Ave., New York 17, N.Y.
Hardin, James C., Jr. (J'42) (MCJW), Pres., Rock Hill Body Co., W. Main St., Rock Hill, S.C.
Harding, Adalbert (J'98), R.F.D. 2, Morristown, N.J.
Harding, Howard ('14; '35), Tech. Engr., Rochester Gas & Elec. Corp., 89 East Ave.; *for mail,* 29 Kingston St., Rochester, N.Y.
Harding, Louis Allen ('13; F'43) (BSKHE), 85 Cleveland Ave., Buffalo 9, N.Y.
Harding, Randolph Adolphis, Jr. (J'44) (ARS), 3508 Roland Ave., Baltimore 11, Md.
Harding, Robert Floyd (J'42) (YS), Jr. Devel. Engr., Gates Rubber Co., 999 S. Broadway, Denver 17; *for mail,* 3rd & Leon Sts., Delta, Colo.
Harding, Walter L. (J'40) (KSEFBL), Mar. Dept., Combustion Engrg Co., Inc., 200 Madison Ave.; *for mail,* 120 E. 31st St., New York 16, N.Y.
Harding, Warren Gamaliel (J'43) (MO), Lt., U.S.N.R.; *for mail,* 10 Montague St., Worcester 3, Mass.
Hardway, Jon Porter, Jr. (J'43), Lt., U.S.N.R., 1850 Montecito Way, San Diego 3, Calif.
Hardwick, James Berry (J'41) (ZST), Instrument Supvr., Joseph E. Seagram & Sons, Inc.; *for mail,* 31 E. Tate St., Lawrenceburg, Ind.
Hardy, Albert L. (J'40), Bldg. 41, Gen. Elec. Co.; *for mail,* 1274 Garner Ave., Schenectady, N.Y.
Hardy, Carroll Foster (M'43) (EFORS), Ch. Engr., Appalachian Coals, Inc.; *for mail,* 307 E. 4th St., Cincinnati 1, Ohio.
Hardy, George F. ('95) (HLS), Owner, Firm of George F. Hardy, Cons. Engr., 441 Lexington Ave., New York 17, N.Y.
Hardy, John Alexander ('28; '33) (OLWHS), Asst. Engr., George F. Hardy, Cons. Engr., 441 Lexington Ave., New York 17, N.Y.
Hardy, Norman Gladding ('13; '14; '19) (SFEKBC), Supt. of Power, Tex. Power & Light Co., Box 6331, Dallas 2, Tex.
Hardy, Rufe J. (J'43) (MC), Owner, South Branch Mch. Shop, Bartley; *for mail,* Long Valley, N.J.
Hardy, William A. ('35), Comdr., U.S.N., Retired; 12 Pryer Lane, Larchmont, N.Y.
Hardy, William Marion (J'46), 908 N. Mangum St., Durham, N.C.
Hare, Edward Taylor, Jr. (J'43) (SEMFKA), Indus. Serv. Rep., Standard Oil Co., 1300 Harmon Pl., Minneapolis 3; *for mail,* Box 232, Route 1, Mound, Minn.
Hare, Wilbur Emory (J'39) (RFDS), West. Sales Mgr., Allen, Sherman, Hoff Co., 225 S. 15th St., Philadelphia 2; *for mail,* Greene Manor Apts. 13, Greene & Johnson Sts., Philadelphia 44, Pa.
Hargest, William James ('42) (MRJ), Assoc. Editor, *American Machinist,* McGraw-Hill Publ. Co., 330 W. 42nd St., New York 18, N.Y.
Hargis, James R. ('40) (CDEKLO), Asst. Ch. Engr., Pure Carbonic, Inc., 3rd & Virginia Sts., Berkeley 2, Calif.
Hargrave, Russell William ('99; '14) (COYEA), Cons. Engr., 27 Manchester Rd., Poughkeepsie, N.Y.
Hargreaves, George (J'38) (SBH), Indus. Power Rep., B. C. Elec. Ry. Co., 600 Granville St., Vancouver; *for mail,* 1011 Hamilton St., New Westminster, B.C., Can.
Harkenrider, Robert Joseph (J'46) (ALR), Research Engr., Miller Waste Mills, Inc., 515 W. 3rd St.; *for mail,* Suite 3, 116 E. Broadway, Winona, Minn.
Harker, John Shields (J'41) (BHN), Mgr. Mch. Design, Kimble Glass Co., Division Ave.; *for mail,* Maple & Mayfair Ave., Vineland, N.J.
Harkins, H. Drake ('33) (SCLF), Dir. of Consultants, E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Harlan, J. Allen ('42) (CM), Textile-Finishing Mch. Co., Box 1716, Providence 1, R.I.
Harlan, John Harold (J'41), Box 772, Morgan City, La.
Harley, James Archibald (J'43) (CLNY), Asst. Works Mgr., Baker Perkins, Inc., Saginaw, Mich.
Harlow, James H. ('25; '35) (CFHSZ), Asst. Mech. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Harlow, Justin Edwards ('37) (DJL), Sales Rep., Stearns Magnetic Mfg. Co., 635 S. 28th St., Milwaukee 4, Wis.; *for mail,* 3519 Boudinot Ave., Cincinnati 11, Ohio.
Harlow, Loraine Charles (J'44), (JKM), Table Rock, Neb.
Harman, George A. ('30; '39) (HMS), Mech. Engr., Harworth Engrg., South Orange; *for mail,* 226 S. 11th St., Newark 7, N.J.
Harman, George Lloyd, Jr. (J'37) (SZEH), Mech. Engr., Bailey Meter Co., Rm. 1009, 1709 W. 8th St., Los Angeles 14, Calif.
Harman, John J. ('06; '09; '12) (JSR), 552 Twin Palms Dr., San Gabriel, Calif.
Harman, William Henry, Sr. (A'06) (HJOZC), Pres., William Sellers & Co., Inc., 1600 Hamilton St., Philadelphia 30, Pa.
Harmer, Robert Lewis ('38) (CLS), 8547 Dante Ave., Chicago, Ill.
Harmon, Robert Childs (J'44) (JGNOBC), Engr., Tractor Div., Catalog Dept., Isaacson Iron Works, 3917 E. Marginal Way, Seattle 4; *for mail,* 4311—12th, N.E., Seattle 5, Wash.
Harmon, Robert Harrison (J'42) (AH), Lt., Flight Engr., Army Air Forces, Officers' Mail Sec., Lowry Field, Denver, Colo.; *for mail,* 809 Talma St., Aurora, Ill.
Harmon, Wayne A. S. ('37) (LKSHBE), Box 927, Los Angeles 15, Calif.
Harmon, William Thomas (J'37) (KAM), Field Engr., Gen. Elec. Co., 222 N. Bank Dr., Chicago; *for mail,* 3 E. Oak Ave., La Grange, Ill.
Harms, Carl Frederick ('45) (AEFJRS), Mgr. Supercharger Dept., Elliott Co., Jeannette; *for mail,* 565 Shuey Ave., Greensburg, Pa.
Harmstad, J. Edwin (J'39) (SLKECW), Power Supvr., Chambers Works, E. I. du Pont de Nemours & Co., Deepwater; *for mail,* 1 Marlton St., Woodstown, N.J.
Harmuth, Joseph Thomas ('44) (CKS), Plant Engr., Am. Cyanamid & Chem. Corp.; *for mail,* 611 Winfield Dr., Bridgeville, Pa.
Harney, Doran Brice (J'40) (EHL), Ch. Tech. Sales Engr., Pacific Pumps, Inc., 5715 Bickett St.; *for mail,* 6215 Arbutus St., Huntington Park, Calif.
Harnsberger, Audley E. ('22; '28; '35) (LAEKSF), Chmn., Refinery Control Bd., Pure Oil Co., 35 E. Wacker Dr., Chicago 1, Ill.
Harold, Paul John (J'40) (SNLC), Mech. Power Design Engr., Electrochemical Dept., E. I. du Pont de Nemours & Co., Buffalo Ave. & Chemical Rd.; *for mail,* 2467 Willow Ave., Niagara Falls, N.Y.
Haroldson, Hugh W. (J'39) (ABRWSY), Assoc. Mar. Engr., U.S. Maritime Comm., Washington, D.C.; *for mail,* 1537 Key Blvd., Arlington, Va.
Harp, Terance Reginald ('44) (ABEKLZ), Mech. Engr., Engrg. Dept., Air Reduction Co., 60 E. 42nd St., New York 17; *for mail,* "Millwood," 507 Wall St., Huntington, L.I., N.Y.
Harper, Arthur C. ('20) (CNMGB), Pres., Wyomissing Poly. Inst.; *for mail,* 1124 Reading Blvd., Wyomissing, Pa.
Harper, Donald Raymond (J'42) (KEFLON), Propr., D. R. Harper Co., 1437 Schofield Bldg., Cleveland 15; *for mail,* 1469 E. 252nd St., Euclid 17, Ohio.
Harper, Edward Alvin ('34; '35) (ESP), Port Engr., Lykes Bros. S.S. Co., U.S. Natl. Bank Bldg., Galveston, Tex.
Harper, Ernest C. (J'29) (ELYH), Serv. Engr., Ingersoll-Rand Co., 11 Broadway, New York, N.Y.; *for mail,* 199 S. Main St., West Hartford 7, Conn.
Harper, George Brewster (J'41) (AC), Box 261, Harper Terminal, Inc., Branchville, N.J.
Harper, James Calvin (J'43) (CDJMB), Effic. Engr., Carborumdum Co., Buffalo Ave.; *for mail,* 352 Buffalo Ave., Niagara Falls, N.Y.
Harper, John Harlan ('25; '37) (LKJMC), Ch. Engr., Fansteel Metal. Corp., North Chicago, Ill.
Harper, Joseph J. ('45) (SCFJKD), Asst. Boiler Engr., Philadelphia Elec. Co., Front & Ward Sts., Chester; *for mail,* 142 Glentay Rd., Lansdowne, Pa.
Harper, Kennard W. (J'31) (BJM), Devel. Engr., Spencer Lens Co., 19 Doat St., Buffalo; *for mail,* 416 Oakwood Ave., East Aurora, N.Y.
Harper, Philip Strickland ('17; '45) (CMN), Pres., Harper-Wyman Co., 8562 Vincennes Ave., Chicago 20, Ill.
Harper, Robert S. (J'40) (CMJBDN), Geometric Tool Co., Blake & Valley Sts., New Haven; *for mail,* 146 Armory St., Hamden, Conn.
Harper, William Carrell, Jr. (J'45), Sales Engr., Stand. Pressed Steel Co., Jenkintown, Pa.; *for mail,* 147 Stonecrest Dr., Forestville, Conn.
Harrer, Theodore Michol (J'41) (ADM), Atlas Engrg. Co., 3421 Hollis St., Oakland 8, Calif.
Harrigan, Howard Hancock ('43) (SKFC), Plant Engr., Potomac Elec. Power Co., 1st & V Sts., S.W., Washington 4, D.C.; *for mail,* 343 Baltimore Ave., Takoma Park 12, Md.
Harrigan, William ('29) (FEAMOR), Cons. Technologist, Tex. Co., 135 E. 42nd St., New York 17, N.Y.; *for mail,* 31 E. Newell Ave., Rutherford, N.J.
Harriman, N. F. ('04), Retired; c/o A. H. Fetters, 4435 Conde Place, San Diego, Calif.
Harrington, Archie E. ('31; '35) (EFS), Ch. Engr., Charge Light, Heat & Power, E. I. du Pont de Nemours & Co., Lancaster St.; *for mail,* 11 Grove Terrace, Leominster, Mass.
Harrington, Carl C. ('43) (MCSD), Editor, Conover-Mast Corp., 205 E. 42nd St., New York 17, N.Y.
Harrington, Carlos Elmer ('26) (BLOAJD), Winfield H. Smith, Inc., 19 Elton St., Springville, Erie Co., N.Y.
Harrington, Cornelius John, Jr. (J'45) (KJEFHM), Ensign, U.S.N., U.S.S. *Carteret,* APA-70, F.P.O., San Francisco, Calif.; *for mail,* 55 Franklin St., Brookline 46, Mass.
Harrington, Earl Whittemore ('19; '25), V.P., Engr., Mfrs. Mutual Fire Ins. Co., 10 Weybosset St., Providence 1; *for mail,* 55 Columbia Ave., Cranston 5, R.I.
Harrington, Ferris T. ('30; '35) (HNAYDT), V.P., Sales, Vickers, Inc., 1400 Oakman Blvd., Detroit 26; *for mail,* 16800 Parkside St., Detroit 21, Mich.
Harrington, James V. (J'44), 28 Rogers St., New London, Conn.
Harrington, Robert Lee (J'42) (MCJ), Ensign, U.S.N.R.; 24 Woodlawn Rd., West Medford 55, N.J.
Harris, A. Emerson ('42) (BRK), Engr., Simonds Saw & Steel Co., Intervale Rd., Fitchburg; *for mail,* 1232 Main St., Leominster, Mass.
Harris, Anderson Winfield ('21; '28) (FKLS), Dept. Mgr., Combustion Engrg. Co., Inc., 1032 W. Main St., Chattanooga 2; *for mail,* 1204 Hanover St., Chattanooga 5, Tenn.
Harris, Carl Chester ('14) (CH), Pres., Rodney Hunt Mch. Co., Mill St.; *for mail,* Warwick Rd. Orange, Mass.
Harris, Charles O. ('43; '45) (BNY), Prof. of Mechanics, Ill. Inst. of Tech., 3300 Federal St., Chicago 16; *for mail,* 8540 Euclid Ave., Chicago 17, Ill.
Harris, Edgar Glenn (J'42), Ensign, U.S.N.R.; 2950—55th Ave., Oakland, Calif.
Harris, Ernest N. ('21; '35) (SKDHF), Cons. Engr., H. W. Beecher, Seattle 1; *for mail,* 2434—36th Ave. W., Seattle 99, Wash.
Harris, Ford Whitman ('13) (AB), Sr. Partner, Harris, Kiech, Foster & Harris, 1151 S. Broadway, Los Angeles, Calif.
Harris, George Newberg (J'43) (YCAWO), 333 Louisiana, Oak Ridge, Tenn.
Harris, George Sylvester ('39) (FKS), Asst. Elec. Engr., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington Bldg., Baltimore 3; *for mail,* 3315 Winterbourne Rd., Baltimore 16, Md.
Harris, H. Patterson (A'18) (CBAEMR), Asst. to Sales Mgr., Bryant Elec. Co., 1421 State St., Bridgeport 2; *for mail,* "Ivy Hill," Southport, Conn.
Harris, Harold C. (J'46), Worminster Rd., Hatboro, Pa.
Harris, Harry Ezekiel ('11; F'40) (OCMLND), Cons. Engr., 229 Thorme St., Bridgeport 6, Conn.
Harris, Henry S. ('18) (EFH), Secy., The Engineers' Club of Philadelphia, 1317 Spruce St., Philadelphia 7; *for mail,* 4 Greenwood Pl., Wyncote, Pa.
Harris, Herbert I. ('45) (BEP), V.P., Penfield Petroleum Products, Inc., 136-30—38th Ave.; *for mail,* 67-04—184th St., Flushing, N.Y.
Harris, J. Earl, Jr. (J'41) (ABF), 1628 Howarth St., Philadelphia, Pa.
Harris, James T. (J'43), Box 943, Avondale, Ariz.
Harris, LeRoy Stanford (J'46), 205 Le Carra Dr., Lansdowne, Pa.
Harris, Morton Foster (J'40) (AMOCN), Asst. Project Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* 106 Ayers Court, West Englewood, N.J.
Harris, Phil B. ('31) (CR), 538 S. Flower St., Los Angeles 13, Calif.
Harris, Robert Leighton (J'44) (BJMNCH), Ensign, (E)L, U.S.N.R., Box 1663, Santa Fe, New Mex.
Harris, Ruben Jay (J'45) (MSCJLD), Test Engr., Gen. Elec. Co., Philadelphia, Pa.; *for mail,* 532 Bradford St., Brooklyn 7, N.Y.
Harris, Sydney Preston (J'37) (JMZ), Mech. Engr., Schick Inc., 45 Garden St., Stamford; *for mail,* 7 Phillips Lane, Darien, Noroton Heights, Conn.
Harris, W. Eugene (J'33) (OSCL), Mech. Engr., DeLeuw, Cather & Co., 20 N. Wacker Dr., Chicago 6; *for mail,* 1305 N. Monitor Ave., Chicago 51, Ill.
Harris, William B. D., Jr. (J'43) (ZMG), R. D. 2, Doylestown, Pa.
Harris, William John, Jr. ('13) (FJ), Indus. Fuel Engr., Laclede Gas Light Co., 1017 Olive St., St. Louis 1; *for mail,* Route 2, Valley Park, Mo.
Harris, Zachary Nick (J'43) (CMNAKJ), Jr. Mech. Engr., Goodyear Aircraft Corp., Akron; *for mail,* 924 N. Howard St., Akron 10, Ohio.
Harrison, Chas. Geo. ('24) (SF), Owner, Chas. G. Harrison Co., 12075 Greenfield St., Detroit 27, Mich.
Harrison, Ernest Burton ('45) (SFEKZC), Prod. Supt., St. Joseph Light & Power Co., 112 S. 2nd St., St. Joseph 5, Mo.
Harrison, Frank Hugh ('44), Ch. Mech. Engr., South Australian Ry. Comm., Islington Ry. Workshops, Lower North Rd., Islington; *for mail,* "Urrbrae," Esplanade, Tennyson, South Australia.

Harrison, Harry ('13; '19) (CK), Mech. Engr., East. Cold Storage Insulation Co., 101 Park Ave., New York 17; *for mail,* 55 Winthrop Ave., Tuckahoe 7, N.Y.
Harrison, J. Houston ('23), Mech. Engr., Valuation, So. Ry. System, Rm. 940, McPherson Sq., Washington, D.C.
Harrison, John Large (J'33) (CFEBAM), Engrg. Asst., Harrisburg Gas Co., 14 S. Market Sq., Harrisburg, Pa.
Harrison, Joseph Wiley (J'39) (CJKMYZ), Ch. Inspr., Norge Mch. Products Div., Borg-Warner Corp., 2969 Lake Shore Dr., Muskegon, Mich.
Harrison, Leroy Wallace ('45) (DJLMNO), Sales Engr., Faitoute Iron & Steel Co., Inc., 182 Frelinghuysen Ave., Newark 5; *for mail,* 216 Fabyan Pl., Newark 8, N.J.
Harrison, Maurice Robert (A'43) (CJMNLO), Secy., Treas., Dir., Behringer Metal Works, Inc., 108-22 Jabez St., Newark 5, N.J.
Harrison, Robert Edward William ('27) (JNMRHA), V.P., Chambersburg Engrg. Co., Derbyshire St., Chambersburg; *for mail,* Route 1, St. Thomas, Pa.
Harrison, Thomas J. (J'38) (CJM), Indus. Engr., Republic Steel Corp., Alabama City; *for mail,* 436—2nd St. E., Gadsden, Ala.
Harrod, Charles Franklin (J'41) (MCZAB), Asst. Engr., Crescent Tool Co., 200 Harrison St.; *for mail,* 351 Foote Ave., Jamestown, N.Y.
Harrod, Raymond Johnson (J'37) (RCS), Lt., U.S.N.R., Trans. Officer, Naval Mag., Bangor, Wash.; *for mail,* 2009 Ash St., Terre Haute, Ind.
Harshman, Donald Guy (J'45) (EKSHWZ), 2811 Ellis St., Bellingham, Wash.
Harszy, Charles H. (J'37) (SZ), Sr. Asst. Engr., Cahokia Power Plant, Union Elec. Power Co., Monsanto; *for mail,* 21 N. 41st St., Belleville, Ill.
Hart, David Anderson (J'43) (BKF), Design Engr., Westinghouse Elec. Corp., Lester; *for mail,* 913 Lincoln Ave., Prospect Park, Pa.
Hart, David K. (J'41) (EFKA), Apt. 107, 2517 Mozart Pl., N.W., Washington 9, D.C.
Hart, Frank Joseph (J'42) (AYC), Pvt., Co. F, 85th Mt. Inf., A.P.O. 345, c/o Postmaster, New York, N.Y.
Hart, Fred W. ('19), Pres., Brookridge Farm, Inc., Littleton, Colo.
Hart, Gordon Cole (J'43) (AMCSY), 40 Quincy St., Medford 55, Mass.
Hart, Herman A. (J'46) (BJL), Design Engr., Wyatt Metal & Boiler Works, Dallas 1; *for mail,* 2839 W. Clarendon, Dallas 11, Tex.
Hart, Howard P. ('15; '25), Asst. to Mgr., Platt Bros. & Co., Box 1030; *for mail,* 36 Buckingham St., Waterbury, Conn.
Hart, James Max (J'42) (ABMNWZ), Asst. to Sec., Schwarz Paper Co., 1430 S. Canal St., Chicago; *for mail,* 215 Orchard Lane, Highland Park, Ill.
Hart, John Seward ('45) (OCZF), Chicago Dist. Mgr., Cent. Sta. Steam Co., 80 E. Jackson Blvd., Chicago 4, Ill.
Hart, Lawrence Henry ('29; '35) (RBCDEM), Supvg. Engr., Tech. Trng., Dept. of Rys., 19 York St., Sydney, N.S.W., Australia.
Hart, Simeon Thompson ('24) (CGLMN), Head, Admin. Engrg., Syracuse Univ.; *for mail,* 168 Westminster Ave., Syracuse 10, N.Y.
Hartburg, Herman Louis ('23; '28) (CDKLOS), Dist. Supt., Great West. Sugar Co., Box 5308, Terminal Annex, Denver 17, Colo.
Hartenberg, Richard S. ('28; '40) (ABNWRH), Asst. Prof. Mechanics, Tech. Inst., Northwestern Univ.; *for mail,* 2448 Lincolnwood Dr., Evanston, Ill.
Harter, Isaac ('08; '21) (FSCMJL), V.P., Babcock & Wilcox Co., Barberton, Ohio.
Harter, Robert Jackson (J'42) (BANJ), Stress Analyst, Kellett Aircraft Corp., Baltimore Pike & Chester Rd., Swarthmore, Pa.
Hartford, Ernest (A'18) (CLW), *Executive Assistant Secretary,* A.S.M.E., 29 W. 39th St., New York 18, N.Y.
Harthorn, Paul Dudley (J'45) (LCFD), Graduate Work, Rensselaer Poly. Inst., Troy, N.Y.; *for mail,* West Pembroke, Me.
Hartin, Frank Robert (J'35) (SECBNJ), Lt., U.S.N., Naval Oper. Base, Guantanamo Bay, Cuba; *for mail,* 42 Jerome Ave., West Newton 65, Mass.
Hartley, Arthur Clifford ('45) (BPS), Ch. Tech. Dir., Petroleum Warfare Dept., Ministry of Fuel & Power; Ch. Engr., Anglo Iranian Oil Co., Ltd.; *for mail,* Britannic House, Finsbury Circus, London E.C. 2, England.
Hartley, Harry Dwight ('10; '21; '34) (CFKSW), 4051 Washington Blvd., Indianapolis, Ind.
Hartman, Elmer Charles ('34; '45) (ACEMNO), V.P., Engr., Technifinish Lab., Inc., 38 Scio St., Rochester 4, N.Y.
Hartman, Elmer Edward ('39) (BFS), 825 E. 21st, North Kansas City, Mo.
Hartman, Frederich Vincent ('31) (DEJKN), Mech. Engr., Tech. Div., Aluminum Ore Co., 3300 Missouri Ave., East St. Louis, Ill.
Hartman, James Busse ('42) (BEHSJK), Dept. of Mech. Engrg., Lehigh Univ., Bethlehem, Pa.
Hartman, John H. (J'37) (CRMAEY), 2336 Linn St., Williamsport 24, Pa.
Hartman, John M. ('20; '30) (FKS), Engr., Charge of Research Lab., Kewanee Boiler Corp., Kewanee, Ill.
Hartman, Joseph A. (J'40) (CMJ), Comdr., U.S.N.R., 8502—48th Ave., Berwyn, Md.
Hartman, Lawrence R. (J'36) (BKS), Consltd. Gas, Elec. Light & Power Co., Madison St. Bldg., Baltimore; *for mail,* 48 S. Dundalk Ave., Dundalk, Md.
Hartman, Wilmer W., Jr. (J'39) (OMCS), Staff Engr., Day & Zimmermann, Inc., Philadelphia, Pa.; *for mail,* 34 Overbrook Dr., Colwick, Merchantville, N.J.
Hartmann, Carl ('21; '35) (BCMNRZ), Designer, 3798 River Rd., Cincinnati 4, Ohio.
Hartmann, Melvin J. (J'44) ABHKNM), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 4562 W. 174th St., Cleveland 11, Ohio.
Hartmann, Warren J. (J'42) (HAB), Ch. Engr., Natl. Metal Products Corp., 21 N. Loomis St., Chicago 7; *for mail,* 3507 W. 26th St., Chicago 23, Ill.
Hartnell, George Freeman (J'39) (AZYMEB), 2nd Lt., Project Engr., Armament Lab., Engrg. Div., Matériel Comd., Wright Field, Dayton, Ohio.
Hartnett, Edward John (J'43) (MJB), Asst. Communications Officer, U.S.S. *Kitkun Bay,* CVE-71, Naval Sta., Tacoma 2, Wash.; *for mail,* 210 Charles Ave., Staten Island 2, N.Y.
Hartridge, Alfred Lamar ('41) (CO), Asst. Mgr., Stone & Webster Engrg. Corp., 90 Broad St., New York 4, N.Y.
Hartshorn, Derick Sibley ('40) (BCJM), Asst. Works Mgr., Cameron Mch. Co., 61 Poplar St., Brooklyn; *for mail,* 279 Washington Ave., Brooklyn 5, N.Y.
Hartsig, Albert L., Jr. (J'37) (CLS), Defender Div., E. I. du Pont de Nemours & Co., Box 1009, Rochester 3; *for mail,* 72 Vassar St., Rochester 7, N.Y.
Hartwell, Arthur Edward ('11; '13; '22) (CJM), Pres., Gen. Mgr., Hartwell Iron Works, Inc., 224 Rains St., Houston 2; *for mail,* Box 105, Houston 1, Tex.
Hartwell, Cushman ('45), Norris & Elliott Resident Engr. at Harwood Mfg. Corp., Inc.; *for mail,* Royal Oak Apts., Marion, Va.
Hartwell, George Warren (J'43) (NBKRAS), Design Engr., Worthington Pump & Mchy. Corp., Appleton, Holyoke; *for mail,* 55 Maple St., Springfield, Mass.
Hartwell, Hiram Britton ('03; '39) (MNZJBL), Mech. Engr., Crosby Steam Gage & Valve Co., 10 Roland St., Charlestown, Boston; *for mail,* 9 Townsend St., Waltham 54, Mass.
Hartwell, John Nathan (J'46), 111 Cherry St., Gardner, Mass.
Hartwell, Terence Canada (J'41) (AM), 1st Lt. (Pilot), Army Air Forces, 377th Sqd., 309th Bomb. Gp., Army Air Base, Columbia, S.C.; *for mail,* 5 E. Bissell Ave., Oil City, Pa.
Hartwell, Thurston ('45) (CLMKJN), Ch. Engr., Barnstead Still & Sterilizer Co., Inc., 2 Lanesville Terrace, Roslindale 31; *for mail,* 20 Pinewood Rd., Needham, Mass.
Hartz, Marvin Edward (J'42) (ABJ), Contract Engr., Goodyear Aircraft Corp., Akron; *for mail,* 160 Russell Ave., Akron 11, Ohio.
Harvey, Albert Henry ('32) (SCTNJK), Managing Dir., Inniss & Riddle (China) Ltd., 43 Yuen Ming Yuen Rd., Shanghai, Kiangsu, China.
Harvey, Alexander (J'46), Mech. Designer, Rapid Transit Dept., Toronto Trans. Comm.; *for mail,* 49 Foxbar Rd., Toronto, Ont., Can.
Harvey, Bruce Eugene (J'41) (AHB), Engr., Draftsman "A", Lockheed Aircraft Corp., 1705 Victory St., Burbank; *for mail,* 306 N. Electric St., Alhambra, Calif.
Harvey, Cyrus Robinson (J'32) (YAG), Estimator, Westinghouse Elec. Corp., Lester; *for mail,* 1 Drexel Rd., Swarthmore, Pa.
Harvey, Edmund Lukens (J'37) (A), Pvt., 1st Class, Army Air Forces, Sqd. A, Lowry Field, Denver, Colo.; *for mail,* Bancroft Rd., Moylan, Pa.
Harvey, James Edward (J'43) (BZM), 2895 Elmwood Ave., Ann Arbor, Mich.
Harvey, James Edward, Jr. (J'40) (CBMEGD), Lt. Comdr., Trng. in Industry (Pur.), Air Reduction Co., Inc., Rm. 1036, 60 E. 42nd St., New York, N.Y.
Harvey, James H. (J'45), 1931 S. Grove Ave., Berwyn, Ill.
Harvey, John C. ('42) (COMH), Ord. Engr., Boston Ord. Dist., 140 Federal St., Boston 10; *for mail,* 154 Beacon St., Chestnut Hill 67, Mass.
Harvey, Maurice Eugene (J'37) (NSBH), Design Engr., Westinghouse Elec. Corp., Lester; *for mail,* Ardmimink Arms, City Line & Childs Ave., Drexel Hill, Pa.
Harvey, Thomas Norman, Jr. (J'38) (BJE), Engr., Bob Hipp Welding Works, 1460 El Camino Real; *for mail,* 955 Fremont St., Menlo Park, Calif.
Harvill, Henry Lloyd ('44) (BJM), Pres., H. L. Harvill Mfg. Co., Box 335, Vernon 11; *for mail,* Box 301A, Route 3, Anaheim, Calif.
Harwick, Harry K. ('42) (RAFJB), Assoc. Engr., Edw. G. Budd Mfg. Co., 24th & Hunting Park Ave., Philadelphia 32, Pa.
Harwood, Harry P. (J'43) (YL), 103 E. Walnut St., Alexandria, Va.
Harz, Joseph, III (J'40), Lt. (j.g.); 1104 Crawford St., Vicksburg, Miss.
Harza, Leroy Francis ('19) (H), Pres., Harza Engrg. Co., 205 W. Wacker Dr., Chicago, Ill.
Hasegawa, Akira (J'41) (HJK), Engrg. Draftsman, County Regional Planning Comm., 205 S. Broadway, Los Angeles 12; *for mail,* 707 N. State St., Los Angeles 33, Calif.
Hasegawa, Tatsuo ('35; '45) (CHJKMY), Design Engr., Acting Ch. Draftsman, Dri-Steam Products, Inc., 559 W. 35th St., New York 1; *for mail,* 75 Wadsworth Terrace, New York 33, N.Y.
Haselberger, Ray E. (J'35) (AZY), Project Engr., Aero. Div., Minneapolis-Honeywell Regulator Co., 2753—4th Ave., Minneapolis; *for mail,* 408 N. Wheeler St., St. Paul 4, Minn.
Hashagen, John B. (J'20), 4236 N. Mozart St., Chicago, Ill.
Haskell, J. Dennis ('21; '31), Secy., Engr., Dilts Mch. Works, Inc., Fulton, N.Y.
Haskell, M. E. ('32), 3 St. James Court, Marine Dr., Bombay, India.
Haskin, Joseph Robert, Jr. ('42; '45) (CSEO), Steam Engr., Monsanto Chem. Co., 1700 S. 2nd St., St. Louis 4; *for mail,* 419 Clark Ave., Kirkwood 22, Mo.
Haskin, Lawrence Howard, Jr. (J'45) (ABCJM), Champlain Co., Inc., 88 Llewellyn Ave., Bloomfield, N.J.
Haskins, George William ('21; '26; '35) (ABCEF), Lt. Col., Army Air Forces, Commanding, Mills Field, San Bruno, Calif.; *for mail,* 283—28th Ave., San Francisco 21, Calif.
Haslach, Joseph P. ('40) (GMT), Engr., Webendorfer & Wills, Inc., Mt. Vernon; *for mail,* 1594 Unionport Rd., New York, N.Y.
Hasse, Frank C. ('36) (RCJ), V.P., Mech. Dept., Oxweld R.R. Serv. Co., 230 N. Michigan Ave., Chicago 1, Ill.
Hasselberg, Frederick Crozier, Jr. (J'44) (NOMDC), Design Engr., Celery Harvester Co., Box 831, Sarasota, Fla.
Hassell, Howard Johannas ('45) (SOEFNB), Ch. Engr., Power Engrg. Co., Box 1777, Salt Lake City; *for mail,* 1519 S. 16th St. E., Salt Lake City 5, Utah.
Hassett, Robert J. (J'43) (BAN), Sr. Stress Analyst, Glenn L. Martin Co., Middle River, Baltimore 20; *for mail,* 24 Walkern Rd., Baltimore 21, Md.
Hassler, Frank Rogers ('24; '27; '35) (CM), Lt. Col., Ord. Dept., U.S.A., Chief, Readjustment Div., St. Louis Ord. Dist., 4801 Riverview Blvd., St. Louis, Mo.; *for mail,* Box 476, R.R. 13, Kirkwood 22, Mo.
Hassman, Fred Anthony ('30) (BHN), Design Supvr., Cincinnati Milling Mch. Co., Marburg St., Cincinnati; *for mail,* 824 Wakefield Dr., Cincinnati 26, Ohio.
Hastings, Chas. F. ('21), Retired; 92 Lakeview Ave., Lynn, Mass.
Hastings, Raymond George ('30) (SMCDOA), Valuation Engr., Jackson & Moreland, 31 St. James Ave., Boston 16; *for mail,* 132 Wellesley St., Weston 93, Mass.
Haswell, Henry Lawrence ('44) (NHG), Asst. Ch. Engr., Devel., Champagne Paper Corp., Box 300, Pisgah Forest; *for mail,* 357 Broad St., Brevard, N.C.
Hatch, Albert Marston (J'40) (BAS), Mech. Engr., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 9 Centre St., Brookline, Mass.
Hatch, Burton D. (J'37) (NBZCYJ), Mar. Sec. Leader, Gen. Elec. Co., Bldg. 81, 1 River Rd., Schenectady 5, N.Y.
Hatch, Charles Haight ('43) (COMJ), Mgmt. Engr., Decoy Farm, Rock Hall, Md.
Hatch, Gordon Hayes ('34; '44) (ZLJMGH), Tool & Prod. Engr., Platts Mills, Bristol Co., Waterbury; *for mail,* 116 Cliff St., Naugatuck, Conn.
Hatch, John P. (J'39) (CKS), Draftsman, Newport News Shipbldg. & Dry Dock Co.; *for mail,* 348—58th St., Newport News, Va.
Hatch, Theodore Frederick ('35) (BHMDC), Head, Mem. Relations, Industrial Hygiene Foundation, 4400—5th Ave., Pittsburgh 13, Pa.
Hatfield, Homer F. ('39) (SFD), Steam Engr., Pa. Power & Light Co., 901 Hamilton St., Allentown, Pa.
Hatfield, Robert Marcus, Jr. ('43) (SFCK), Asst. Gen. Sales Mgr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York, N.Y.; *for mail,* 39 Ashton Pl., Glen Rock, N.J.
Hattis, Robert Elijah ('43) (EKS), 28 E. Jackson Blvd., Chicago 4, Ill.
Hatton, Morton ('42) (OCSE), Rate Research Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Hatzfeld, George, Jr. (J'34) (RSKFHE), Test Engr., N.Y. City Transit System, 500 Kent Ave., Brooklyn; *for mail,* 176 Overlook Ave., Great Neck, L.I., N.Y.
Hau, Oscar Enrique (J'41) (BCDEFJ), Diez de Andino 1, Santurce, P.R.
Hauer, Edward Raymond ('42) (RJEFKB), Engr., M.P., Advis. Mech. Comm., 1038 Midland Bldg., Cleveland, Ohio.
Hauffe, Floyd H. (J'44) (EHS), Lt. Col., Hdq., 13th Abn. Div., A.P.O. 333, Ft. Bragg, N.C.
Haug, John Rowland (J'42), Lt., U.S.N.R.; 9509 Germantown Ave., Philadelphia 18, Pa.
Haug, John Sobieski ('27) (FJKBSE), Cons. Gas Engr., United Engrs. & Constructors Inc., 1401 Arch St., Philadelphia 5, Pa.
Hauger, Harry Hoyt, Jr. (J'44) (AKHBS), 8028 Blackburn Ave., Los Angeles 36, Calif.
Haughton, Frank Alricks ('03) (NJRYWH), Retired; 1084 Waverly Pl., Schenectady 8, N.Y.

Haupt, H. Harold ('30) (RSDF), Gen. Supt. M.P., Pa. R.R., Pa. Sta. Bldg., Pittsburgh 22; *for mail*, 591 Moorhead Pl., Pittsburgh 6, Pa.
Hauschildt, Maurice Richard (J'41) (SKFBW), Mar. Engr., Mch. Design, Bur. of Ships, Navy Dept., 17th St. & Constitution Ave., Washington 25, D.C.; *for mail*, 10109 Greenock Rd., Silver Spring, Md.
Hausel, Walter M. ('18; '30) (ACLM), 68 Hamilton Ave., Yonkers 5, N.Y.
Hauselt, J. Donald (J'38) (BKS), Atty. at Law, Ebenezer Oil Co. Bldg.; *for mail*, 132 Stevens St., Wellsville, N.Y.
Hauser, Cavour Herman (J'44) (ABKSCY), Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 1363 Lakewood Ave., Lakewood 7, Ohio.
Hauser, George Henry ('18; '31) (AC), V.P., Liberty Aircraft Products Corp., Farmingdale; *for mail*, 29 Hilton Ave., Garden City, L.I., N.Y.
Hauser, John Herman (J'41) (MNZJAC), Designer, Huck & Co., 420 Lexington Ave., New York; *for mail*, 48-10—43rd St., Woodside, L.I., N.Y.
Hausler, Walter Bertrand (J'41) (ZNBAJM), Spec. Asst. Engr., Office of Scien. Research & Devel., 1530 P St., N.W., Washington 25; *for mail*, 3726 Connecticut Ave., N.W., Washington 8, D.C.
Hausman, Moses ('28; '34; '35) (BDCM), Automotive Mech. Engr., War Dept., Union Guardian Bldg., Detroit 26, Mich.; *for mail*, 447 Ft. Washington Ave., New York 33, N.Y.
Hausman, Sidney ('23; '41) (OCMLDT), Cons. Engr., Westminster, Md.
Hausmann, Louis ('36) (BEFKLN), Comdr., U.S.N.R., Mchy. Design Sec., Bur. of Ships, Navy Dept., Washington; *for mail*, 3000 Gainesville St., S.E., Washington 20, D.C.
Hausmann, William (J'38) (CMJN), Plant Mgr., Schnefel Bros. Corp., 682 S. 17th St., Newark 3; *for mail*, 2143 Kay Ave., Union, N.J.
Havel, Frank Lawrence (J'40) (OMCLKJ), Chief, (Engr.), Charge of Modification Overseas Materiel Procurement, Office, Chief of Ord., U.S.A., Union Guardian Bldg., Detroit; *for mail*, 17127 Ontario Ave., Detroit 20, Mich.
Havemeyer, H. R. (J'44), 1833 E. 13th St., Brooklyn, N.Y.
Havemeyer, Henry Osborne, Jr. ('30; '42) (RF SK), Exec. Asst. to Pres., Brooklyn Eastern Dist. Terminal R.R., 111 Broadway, New York 6, N.Y.
Havens, Kenneth Barton (J'35) (RNBH), Mech. Designer, Sharples Corp., 2300 Westmoreland St., Philadelphia 40; *for mail*, 7271 Guilford Rd., Upper Darby, Pa.
Havens, Louis Arthur ('44) (CMFD), V.P., Mgr., Louis Sherry, Inc., 30-30 Northern Blvd., Long Island City; *for mail*, 107 Union St., Valley Stream, N.Y.
Haver, Ralph L. (J'36) (ACM), Project Engr., Ryan Aero. Co.; *for mail*, 4658 Adams, San Diego, Calif.
Haverstick, John Sterling ('44) (SBKE), Asst. Ch. Engr., Gas Turbine Div., De Laval Steam Turbine Co., Trenton 2; *for mail*, R.D. 2, Stockton, N.J.
Havey, Edward Joseph, Jr. (J'44) (ABJ), 220 N. Regent St., Port Chester, N.Y.
Havlik, Maurice Daniel (J'42) (MLJYEO), Ensign, U.S.N.; *for mail*, Box 254, Moulton, Tex.
Havrylak, Joseph (J'45) (MCDOFY), Ensign, U.S.N.R., Personnel Officer, Staff, Comdr. Amphibious Gp. 3, F.P.O., San Francisco, Calif.
Hawke, Clarence Ewart ('27) (FLKSJC), Gen. Sales Mgr., Refractory Div., Carborundum Co., Perth Amboy, N.J.
Hawkins, Albert E. (J'40) (BDL), Ch. Engr., J. M. Lehmann Co., Inc., 550 New York Ave., Lyndhurst; *for mail*, Apt. E2, 550 Piaget Ave., Clifton, N.J.
Hawkins, Edward Carroll (J'41) (ABHNW), Layout Designer, Glenn L. Martin Co., Baltimore 3; *for mail*, 13 S. Beechwood Ave., Baltimore 28, Md.
Hawkins, George Andrew ('30; '38) (KBZ), *Pi Tau Sigma Medallist*, '40; Prof. Thermodynamics, Purdue Univ.; *for mail*, 701 Crestview, West Lafayette, Ind.
Hawkins, George W. ('13), 316 Greenway Rd., Ridgewood, N.J.
Hawkins, Milton Gregory (J'44) (OCNM), Sales Engr., Barnes & Reinecke, 230 E. Ohio St., Chicago 11; *for mail*, 4529 N. Greenview, Chicago 40, Ill.
Hawkins, Robert Barton (J'43) (ESK), Field Engr., Sealed Power Corp., 5-164 Gen. Motors Bldg., Detroit 2, Mich.
Hawkins, Robert Dawson ('39) (BAH), Prof. Applied Mechanics, College of Engrg., Univ. of Ky., Lexington, Ky.
Hawkins, Wilford Judson ('43) (BCMNO), Works Mgr., Loading Plant, Atlas Ansonia Co., Stiles Lane, North Haven; *for mail*, 145 E. Rock Rd., New Haven 11, Conn.
Hawks, Arthur S. ('04; '09), 4510 Ridge St., Chevy Chase 15, Md.
Hawley, Charles D. ('45) (EFAC), Asst. to Gen. Mgr., Ethyl Corp., 16600 W. Eight Mile Rd., Detroit 20, Mich.
Hawley, Charles Freeman ('39) (FKS), Asst. Ch. Mech. Engr., Riley Stoker Corp., Worcester; *for mail*, Walnut Ave., Holden, Mass.
Hawley, Ransom Smith ('16) (SCF), Chmn., Dept. of Mech. Engrg., Univ. of Mich., Ann Arbor, Mich.
Hawley, William Parker ('03; A'10) (NBJM WG), Retired; 320 N. Ridgeland Ave., Oak Park, Ill.
Haworth, Henry Lowell ('39) (CLS), Ch. Engr., Fox Paper Co., Lockland; *for mail*, 130 Springfield Pike, Wyoming, Cincinnati 15, Ohio.
Hawthorne, Charles Wilson, Jr. ('43) (FLS), Supt. of Rolling Mills Maint., Kaiser Co., Inc., Fontana, Calif.
Hay, Arthur Donald (J'42) (HBNS), Asst. Prof., Project Dir., Sch. of Engrg., Princeton Univ., Washington Rd.; *for mail*, 120 Prospect Ave., Princeton, N.J.
Hay, Barclay William ('27; '45) (CFJKLS), Supt. Gen. Utilities, Tenn. Eastman Corp.; *for mail*, Apt. A-3-121, Oak Ridge, Tenn.
Hay, Earl Downing ('18; '20) (NBGMLY), Prof. Mech. & Indus. Engrg., Univ. of Kan.; *for mail*, 1653 Indiana St., Lawrence, Kan.
Hay, Robert Franklin (J'46) (CJM), 12146 Kentucky Ave., Detroit 4, Mich.
Hay, William Oscar, Jr. ('19; '35) (BCJSZ), Pres., Elec. Securities Corp., 570 Lexington Ave., New York 22, N.Y.
Hayden, Ralph Thomas (J'44) (SFCTMK), Power Engr., Indus. Rayon Corp.; *for mail*, R.F.D. 2, Fairlawn, Covington, Va.
Hayden, William F. (J'40) (CDS), Ensign, R.R. 1, Muscatine, Iowa.
Haydock, John ('34) (MJCADR), Exec. Editor, *American Machinist*, McGraw-Hill Publ. Co., 330 W. 42nd St., New York 18, N.Y.
Haydon, A. William ('40) (ABCNSZ), Pres., Treas., Ch. Engr., A. W. Haydon Co., 711 Lilley Bldg., 111 W. Main St., Waterbury 32, Conn.
Hayek, Arthur Foster (J'39) (NB), Engr., Research, Gen. Precision Lab., Inc., Bedford Rd., Pleasantville, N.Y.; *for mail*, 142 Oakwood Ave., Bogota, N.J.
Hayes, Crawford (J'42) (JMCBN), Design Engr., Remington Arms Co., Ilion, N.Y.
Hayes, Edwin G. (J'37), Draftsman, Marnall Steel Products, Inc., 501—5th Ave., New York; *for mail*, 7615 Colonial Rd., Brooklyn, N.Y.
Hayes, Elmer B. ('41) (FKS), Engr., Charge of Constr., Nashua Gummed & Coated Paper Co., Franklin St.; *for mail*, 27 Concord St., Nashua, N.H.
Hayes, Frank O. (A'45) (CJM), Engrg. Dept., E. I. du Pont de Nemours & Co., du Pont Bldg., Wilmington, Del.
Hayes, John A. ('37) (EF), Lt., U.S.N., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail*, 1318 N. Illinois St., Arlington, Va.
Hayes, Lawrence W. ('17; '35) (EHSY), Treas., Hayes Pump & Mchy. Co., 125 Purchase St., Boston 10, Mass.
Hayes, William T. ('23) (CFP), Local Mgr., Maint. & Repair Div., War Shipping Admin., 1055 Dexter Horton Bldg., Seattle 4, Wash.
Hayleck, Charles Raymond, Jr. (J'43), Asst. Engrg. Instr., College of Engrg., Univ. of Md., College Park, Md.
Haynes, Hasbrouck ('12; '19; '28) (CM), 1st Natl. Bank Bldg., Chicago 3, Ill.
Haynes, James LeRoy ('14; '23) (ORDCNM), Div. Engr., Hyatt Bearings Div., Gen. Motors Corp., 332 S. Michigan St., Chicago 4, Ill.
Haynes, James Morton (J'44) (CE), Witte Eng. Works Div., Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Haynes, Paul Drew (J'44) (CHS), Aviation Radio Technician, 2nd Class, Gen. Delivery, Oceana, Va.
Haynes, Thomas Edwin (J'44) (MCDNAJ), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail*, 146 Hanover Ave., Hampton, Va.
Haynes, Wm. E. ('23; '26; '35) (JNALG), 1st Asst. Supvr., Sch. Shops, Woodrow Wilson Vocational High Sch., Bd. of Education, 92-23—170th St., Jamaica 3, N.Y.
Hays, Charles Raymond (J'45) (BJNDMC), Designer, Chemist, Nanaimo Fdy. & Engrg. Works Ltd., Comax Rd., Nanaimo, B.C., Can.
Hays, Frank Bellchambers ('46), Comdr., U.S.N.R., Planning Supt., Indus. Comd., Naval Repair Base; *for mail*, 2922 Ocean View Blvd., San Diego, Calif.
Hays, Fred Negley ('45) (CSKMFO), Engr., Spec. Assignment, Carnegie-Ill. Steel Corp., 1431 Carnegie Bldg., Pittsburgh, Pa.
Hays, Lawrence Clayton (J'41) (JFESND), Research Fellowship, Lubrizol Corp., Cleveland; *for mail*, 11320 Knowlton Ave., Cleveland 6, Ohio.
Hays, Lewis Tompkins ('16; '18) (ADN), Sales Engr., Columbia Steel Co., Box 3416, Portland, Ore.
Hayward, John Gamble (J'39) (SGB), Assoc. Mar. Engr., Navy Dept., D/O Bldg. 21, U.S. Naval Drydock, South Boston; *for mail*, 200 Middlesex Ave., Wilmington, Mass.
Hayward, Judson ('15; '26) (D), Pres., Treas., Gen. Mgr., Hayward Co., 50 Church St., New York 7, N.Y.
Haywood, Joseph (J'39), Bethlehem Pike & Linderwold Ave., Ambler, Pa.
Hazard, Charles Sprague ('34) (ZAN), Ch. Engr., Neptune Meter Co., 50 W. 50th St., New York 20, N.Y.
Hazard, Charles Stephen (J'42) (AEM), Lt.; 52 Fairway, Bloomfield, N.J.
Hazard, Geoffrey Cornell ('23; '38) (BCT), 97 Ivy Way, Port Washington, N.Y.
Hazard, Herbert R. (J'39) (FKS), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Hazelton, Charles Henry ('19) (MC), Gen. Supt. Prod., Foote Burt Co., 13000 St. Clair Ave., Cleveland 8; *for mail*, 3502 Edison Rd., Cleveland Heights 21, Ohio.
Hazen, Deane Stanley ('21; '26; '35) (CMLJ), Treas., Mgmt. Engr., U.S. Slicing Mch. Co., La Porte, Ind.
Hazen, Frank D. ('37) (EKSZJ), Gen. Sales Mgr., Am. Arch Co., Inc., 1430 Park Bldg., Pittsburgh 22, Pa.
Hazen, Joseph (J'44) (COBFHJ), Operas. Supvr., Shell Oil Co., Inc., 420 Oakland St., Brooklyn 22; *for mail*, 32-51—77th St., Jackson Heights, N.Y.
Hazlett, William A. (J'37) (CHS), 3904 N. Charles St., Baltimore 18, Md.
Head, Francis ('06) (SE), 129 E. 10th St., New York, N.Y.
Head, Victor (J'41), Research Engr., Fluids Engrg. Dept., Fischer & Porter Co., County Line Rd.; *for mail*, 24 Elmwood Ave., Lacey Pk., Hatboro, Pa.
Headley, Lewis M. ('41) (J), Tech. Adviser, Fed. Mch. & Welder Co.; *for mail*, Apt. F, 2629 South St., Warren, Ohio.
Headman, Sasha Sam ('25) (SEWOMF), Partner, Headman, Ferguson & Carollo, 325 Ellis Bldg., Phoenix, Ariz.
Heald, Henry Townley ('40) (BC), Pres., Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.
Heald, Royal H. ('19; '21; '35), Assoc. Physicist, Natl. Bur. of Stands., Connecticut Ave. & Van Ness St.; *for mail*, 6201 Utah Ave., N.W., Washington, D.C.
Heald, Willard R. ('20; '30) (CJM), Mgr., Wilmington Shops Div., E. I. du Pont de Nemours & Co.; *for mail*, 14 Granite Rd., Wilmington 284, Del.
Healey, Edward Aldrich ('27; '32; '35), Equip. Design, Am. Cyanamid Co., 30 Rockefeller Plaza, New York 20, N.Y.
Healy, George F. (J'41) (AJK), 202 S. Augusta Ave., Baltimore, Md.
Healy, James Joseph ('28; '35) (OHFCNS), Project Engr., Am. Rolling Mills Co., Curtis St., Middletown; *for mail*, 261 Yankee Rd., Middletown 3, Ohio.
Healy, John M. ('23; '35), Design Engr., Chas. Hartmann Co., 985 Dean St.; *for mail*, 1758 Madison Pl., Brooklyn, N.Y.
Healy, William Joseph ('42) (ES), Sales Engr., De Laval Steam Turbine Co., Trenton, N.J.; *for mail*, 111 Arborlea Ave., Morrisville, Pa.
Heard, M. Earl ('42) (TKYZ), Dir., Research, West Point Mfg. Co., West Point, Ga.
Hearn, Harold Edwin (J'43) (AE), Asst. Engr., Refinery Maint. Sec., Trinidad Leaseholds, Ltd., Pointe-à-Pierre, Trinidad, B.W.I.
Hearty, Frank J. ('20; '35) (SEK), Owner, F. J. Hearty & Co., Archs. Bldg., 816 W. 5th, Los Angeles 13; *for mail*, 2620 Prospect Ave., La Crescenta, Calif.
Heater, Charles L. ('42) (RC), V.P., Engrg., Am. Steel Fdys., 410 N. Michigan Ave., Chicago 11, Ill.
Heath, Armour Roy, Jr. (J'38) (ACJ), Project Engr., Titeflex Metal Hose Co., 500 Frelinghuysen Ave., Newark; *for mail*, Apt. 1D, 18 Meadowbrook Village, Plainfield, N.J.
Heath, Charles O., Jr. (J'43) (SKJ), Asst. Prof. Mech. Engrg., Rutgers Univ., New Brunswick, N.J.
Heath, Clifford John (J'43), Natl. Magnesium Corp. of Elkton, Md., 74 Trinity Pl., New York, N.Y.; *for mail*, 255 Churchill Rd., West Englewood, N.J.
Heath, Westcott, Jr. (J'39) (ZYJL), Sr. Test Engr., Wright Aero. Corp., Beckwith Ave., Paterson; *for mail*, 410—16th St., Union City, N.J.
Heavilon, Ernest Baker ('28; '45) (COMAJK), Cons. Engr., Charge of Assigned Field Work, Trundle Engrg Co., Bulkley Bldg., 1501 Euclid Ave., Cleveland; *for mail*, 4211 Albertly, Parma, Ohio.
Hebard, James Currier, Jr. (J'44) (BDJLNZ), Mech. Engr., Research & Devel., J. W. Greer Co., 119 Windsor St., Cambridge A39; *for mail*, 450-B William St., Stoneham 80, Mass.
Hebberd, Loren L. ('12; '15; '24), Ch. Engr., Ramtite Co., 2563 W. 18th St., Chicago 8, Ill.
Hebenstreit, Charles George (J'41) (YKLBJ), Design Engr., Bakelite Corp., River Rd., Bound Brook; *for mail*, 9 Barkley Ave., Clifton, N.J.
Hebert, A. J. G. ('40) (DHSBMJ), Asst. to Ch. Engr., Plessisville Fdy.; *for mail*, 129 St. Calixte St., Plessisville, Que., Can.
Hebler, Wm. O. ('33; '35) (ZEFS), Owner, Wm. O. Hebler Co., Central & Long Aves., Hillside, N.J.
Hebley, Henry Frederick ('24; '32) (CFLRS), Dir. of Research, Pittsburgh Coal Co., Box 146, Pittsburgh 30, Pa.
Hebrank, Eugene Ferdinand (J'43) (CMNGJB), Lt. (j.g.), U.S.N.R., Instr., Mar. Engrg. Dept., U.S. Naval Acad.; *for mail*, 302 Dreams Landing, Annapolis, Md.
Hechler, F. G. ('15; '29) (EKS), Dir., Engrg. Exper. Sta., Pa. State College, State College, Pa.
Hecht, Edward E. ('43) (CJM), Die & Tool Engr., Dodge Main Plant, Chrysler Corp., 7900 Jos. Campau Ave., Detroit 11, Mich.

Hecht, Herbert (J'42) (A), Pvt., Corps of Engrs., A.U.S., Separation Sec., Officers Div., Army Serv. Forces Trng. Center, Hdq. Co., NE-22, Ft. Belvoir, Va.; *for mail,* 180 Riverside Dr., New York 24, N.Y.
Heck, John A. (J'40) (BEF), Test Engr., Baldwin Southwark Div., Baldwin Locomotive Works, Eddystone; *for mail,* 958 N. 5th St., Philadelphia, Pa.
Heck, John William (J'41) (JOHN), Asst. Plant Chief, Bertha Mineral Div., N.J. Zinc Co.; *for mail,* Austinville, Va.
Heck, John Wilson (J'36), Surveyor, Am. Bur. of Shipping, 47 Beaver St., New York; *for mail,* 86-10—34th Ave., Jackson Heights, N.Y.
Heck, Robert C. H. ('06; F'43) (SFEKBR), *Life Member for Distinguished Service,* '22; Emeritus Prof., Mech. Engrg., Rutgers Univ.; *for mail,* 51 Adelaide Ave., New Brunswick, N.J.
Hecker, Arthur E. ('21; '29) (LCDFYZ), Ch. Engr., Am. Cyanamid Co., 30 Rockefeller Plaza, New York 20, N.Y.
Hecker, Harvard K. (J'39), Designing Mech. Engr., Medart Co., 3500 DeKalb St., St. Louis; *for mail,* 31 Brentmoor Pk., Clayton 5, Mo.
Hecker, Roland Fred (J'45) (BHJMC), Design Engr., Lodge & Shipley Mch. Tool Co., 3055 Colerain Ave., Cincinnati 23, Ohio.
Heckman, David Allen (J'41) (BMJ), Glen Gardner, N.J.
Heckman, James C. ('20), Engr., Stevenson, Jordan & Harrison, 19 W. 44th St., New York 18; *for mail,* 8 Bayard St., Larchmont, N.Y.
Heckman, Thomas Paul (J'41) (MLDC), 708 N. 35th St., Seattle 3, Wash.
Hedberg, Harold Frederick (J'21) (TCMN), Asst. Gen. Supt., Albany Felt Co., 1373 Broadway, Albany 1, N.Y.
Hedberg, John Viking (J'45), 58—79th St., Brooklyn, N.Y.
Heddaeus, Ray Lester, Jr. (J'40) (NJMA), Jr. Engr., Voss Mchy. Co., 2882 W. Liberty Ave., Pittsburgh 16; *for mail,* 703 Klemont Ave., Pittsburgh 2, Pa.
Heddell, Douglas ('23; '33; '35) (KHCBOG), Quarterman, Htg. & Plumbing Shop, Dept. of Pub. Works, Naval Air Sta., Alameda, Calif.
Hedeen, Ernest William, Jr. (J'44) (NMCDJ), Seaman, 1st Class, U.S.N.R., Ward 22N, Naval Hospital, Great Lakes; *for mail,* 7649 Luella Ave., Chicago 49, Ill.
Hedges, Selby E. ('18; '24), Retired; 922 S. Galena Ave., Dixon, Ill.
Hedman, Sverker Nils Fredrick ('45) (KRSBJN), Designing Engr., Auxiliary Turbine Engrg. Div., Gen. Elec. Co., 166 Broad St., Fitchburg, Mass.
Hedstrom, Kenneth John (J'41) (CNSKLY), Lt., U.S.N.R., Asst. Officer in Charge, Ship's Spares Branch, Naval Supply Center, Guam, Navy 926, F.P.O., San Francisco; *for mail,* 2110—42nd Ave., Oakland 1, Calif.
Heebner, Edmund Richard (J'44), Gen. Delivery, Kearns, Utah.
Heehs, Richard Arthur (J'44), Lt. (j.g.), U.S.N.R., U.S.S. *Gen. J. C. Breckenridge,* AP-176, F.P.O., San Francisco, Calif.; *for mail,* Route 5, Perkins Rd., Memphis, Tenn.
Heekin, Daniel Molloy ('17; '23) (CMR), Secy., Treas., Heekin Can Co., 435 New St., Cincinnati 2, Ohio.
Heenan, John Nelson Dundas ('19; '25) (AOJKFB), Aero. & Gen. Cons. Engr., Heenan Winn & Steel, 29 Clarges St., Piccadilly, London, W.1., England.
Heeren, Duane Wesley (J'38) (HEFK), Constr. & Maint. Engr., San Joaquin Valley Pipeline Dept., Tide Water Associated Oil Co., Box 1101, Coalinga, Calif.
Heffelfinger, Heaton Bennet (J'43) (YRBJ), Rubber Sales Engr., Stanco Distributors, Inc., Stand. Oil Co. of N.J., 26 Broadway, New York 4; *for mail,* 95 Marble Hill Ave., New York 63, N.Y.
Heffernan, John T. ('94), 707 White Bldg., Seattle 1, Wash.
Heffernan, William Henry ('29) (LCMNSK), Mech. Engr., Am. Cyanamid Co., Linden, N.J.; *for mail,* 276 Florence Rd., Walthan 54, Mass.
Hefner, Charles Frank (J'43) (EMC), V.P., Charge of Sales, Hefner Elec. Co., 6 Lafayette Ave., Brooklyn 17, N.Y.
Heftler, Paul ('45) (AR), 1051 Berkshire Rd., Detroit 30, Mich.
Heftler, Victor R. ('14), 1051 Berkshire Rd., Grosse Pointe Park, Mich.
Hefty, Paul Milton (J'39) (LE), Ch. Draftsman, Natl. Gasoline Dept., Phillips Petroleum Co., Bartlesville; *for mail,* 102 S. Wyandotte St., Dewey, Okla.
Hege, Jeremiah Collins (J'43) (ABW), 1st Lt., Army Air Forces, Propeller Lab., Wright Field, Dayton, Ohio.
Hegenbarth, Francis ('23) (KFN), Mgr., Sugar Mchy. Div., Vulcan Iron Works, Box 570, Wilkes-Barre, Pa.
Heger, Erwin Frank (J'40) (MOCL), Mech. Engr., Constr. & Repair, Grasselli Chem. Div., E. I. du Pont de Nemours & Co., Grasselli; *for mail,* 144 Murray St., Elizabeth 2, N.J.
Hegge, Joseph Irving, Jr. (J'45) (ABE), Sr. Test Engr., Wright Aero. Corp., Beckwith & Madison, Paterson; *for mail,* 142 Gordonhurst Ave., Upper Montclair, N.J.
Heggen, Odvar (J'38) (CLM), Lt., O-V (S), U.S.N.R., Naval Shipyard, Brooklyn; *for mail,* 60 Heartt Ave., Cahoes, N.Y.
Heglund, Floyd W. (J'40) (AKS), Box 26, Muroc, Calif.
Hehemann, Fred H. (J'14) (BJSML), Ch. Engr., Lunkenheimer Co., Beekman & Waverly Ave., Cincinnati 14, Ohio.
Hehemann, Robert Frederick (J'43) (BJK), 1st Lt., Ord. Dept., U.S.A., Physicist, Lab. Div., Frankford Arsenal, Philadelphia; *for mail,* 4717 Rising Sun Ave., Philadelphia 20, Pa.
Heidersbach, Fritz George (J'32) (BFJKE), Mech. Designer, M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.
Heidinger, Fritz (J'26) (BDH), Assoc. Engr., U.S. Bur. of Reclamation, New Custom House; *for mail,* 1771 Pennsylvania St., Denver, Colo.
Heidmann, Marcus Frank (J'45) (ABM), Mech. Research Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 2100 Arthur Ave., Lakewood 7, Ohio.
Heigl, Carl H. (J'37) (NJMGCB), Gen. Mgr., Superior Products Co., 5504 Superior Ave.; *for mail,* 628 E. 96th St., Cleveland 3, Ohio.
Heil, Alfred Louis (J'43) (DEFS), Jr. Methods Engr., Fairchild Camera & Instrument Corp., 88-06 Van Wyck Blvd., Jamaica; *for mail,* 242-01 Merrick Rd., Rosedale 10, N.Y.
Heilberg, Herbert (J'44) (BEK), 214 S. Pearl St., Natchez, Miss.
Heilig, Charles Earl, Jr. (J'44) (JKMNRY), Sr. Engr., Design & Devel., Philco Corp., C St. & Allegheny Ave., Philadelphia 34; *for mail,* 1105 Roosevelt Dr., Upper Darby, Pa.
Heilig, William Emrich ('27; '35) (JCBKSE), V.P., Charge Engrg., Wm. Powell Co., 2525 Spring Grove Ave., Cincinnati 22; *for mail,* 3214 Woodburn Ave., Cincinnati 7, Ohio.
Heilman, Russell Howard ('19; '25; '30) (KB), *Junior Award,* '22 & '24; Sr. Fellow, Mellon Inst., 4400—5th Ave., Pittsburgh 13, Pa.
Heim, Alfred Henry (J'44) (BGH), Route 1, Turtle Ridge, Beloit, Wis.
Heim, William (J'41) (BJM), Mech. Engr., S. S. White Dental Mfg. Co., Prince Bay, S.I.; *for mail,* 107-03—86th Ave., Richmond Hill, L.I., N.Y.
Heimberger, Oscar W. ('23; '31) (KSELBR), Asst. Ch. Engr., Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.
Heimbrock, Jos. H. ('38) (FKS), Ch. Engr., Columbia Power Sta., Cincinnati Gas & Elec. Co., Columbia Park; *for mail,* 7250 Overcliff Dr., Cincinnati 33, Ohio.
Hein, Harrison Peckham (J'45) (CMAJ), O.K. Tool Co., 33 Hull St., Shelton, Conn.
Hein, Howard Arthur (J'45) (JMCN), Ch. Sales Engr., Cleveland Automatic Mch. Co., 2269 Ashland Rd., Cleveland 3; *for mail,* 1500 Lyndhurst Rd., South Euclid 21, Ohio.
Hein, Jerome Joseph (J'37) (LNMZDO), Sr. Engr., Engrg. Dept., Corn Products Refining Co., Argo; *for mail,* 249 Herrick Rd., Riverside, Ill.
Heine, Francis A. ('34) (HCZ), Ch. Engr., Bur. of Water, City Hall, Reading, Pa.
Heine, Gregor H. ('18; '24) (FSZ), Mech. Engr., Power Plant Constr., Gibbs & Hill, Inc., 326 Kentucky Ave., Indianapolis 6; *for mail,* Hotel Barton, Indianapolis 4, Ind.
Heineman, John ('35) (AY), Factory Rep., Vulcanized Rubber & Plastics Co., 261-5th Ave., New York 16, N.Y.; *for mail,* 116 Daniels Ave., Rutherford, N.J.
Heinen, Fred C. (J'21), Camel Sales Co., 500—5th Ave., New York, N.Y.
Heiney, Lewis Ernest (J'41) (AJM), Cpl., Army Air Forces, Sqd. 5, 4000th Base Unit (AB), Wright Field, Dayton, Ohio; *for mail,* Box 147, Bloom, Kan.
Heinrich, Chester Joseph ('45) (CMNODJ), 1355—20th St., Fair Lawn, N.J.
Heintz, Ralph Morell ('42) (ABJ), V.P., Charge Engrg., Jack & Heintz, Inc., Cleveland, Ohio.
Heintze, Arthur Louis ('23; '34) (EANSJL), Staff Engr., Sinclair Refining Co., 630—5th Ave., New York 20, N.Y.; *for mail,* 8 McKnight Lane, Clayton 5, Mo.
Heinz, Winfield Bernard ('30; '39) (ZLTKBA), Middlebrook Rd., Bound Brook, N.J.
Heinze, William A. ('33; '35) (CDL), 4415 Alta Canyada Rd., La Canada, Calif.
Heisler, Robert Walter (J'44) (HBNMY), Research Engr., Johns-Manville Corp., Manville; *for mail,* 523 Winsor St., Bound Brook, N.J.
Heisserman, Robert E. (J'40) (D), Engr., Export Div., Link-Belt Co., 233 Broadway, New York 7, N.Y.
Heitz, Robert L. (J'32) (FJ), Lt., U.S.N.R., Naval Inspr., U.S.N., Field Prod. Div., Naval Inspector's Office, Todd Shipyard, Hoboken, N.J.
Helander, Linn ('16; '24) (SKEBFA), *Manager,* '39-'42, *Vice-President,* '45-'46; Prof. & Head, Dept. of Mech. Engrg., Kansas State College, Manhattan, Kan.
Helbig, Robert Walter (J'37) (EJSYDB), Indus. Serv. Rep., Stand. Oil Co. of Ind., 205 Janes St., Saginaw; *for mail,* Box 403, Bay City, Mich.
Helbush, William Walker ('28; '35) (HBSA), Mech. Engr., Charge Design, San Francisco Pub. Utilities Comm., 425 Mason St., San Francisco 1, Calif.
Heldack, John Martin (J'40) (AKBHC), Sales Engr., Gen. Elec. Co., 212 N. Vignes St., Los Angeles, Calif.
Heldmann, Ernest John (J'38) (BCM), 27 King St., Hartford 6, Conn.
Helfter, Franklin S. ('34) (SCFKED), Asst. Supt., C. R. Huntley Stas., Buffalo Niagara Elec. Corp., 535 Washington St., Buffalo 3, N.Y.
Helin, Walter Engleberthe (J'45) (KHOF), Engrg. Asst. to Direct Sales, Carrier Corp., 405 Lexington Ave., New York 17, N.Y.; *for mail,* 684 Chestnut Ave., Teaneck, N.J.
Hella, Robert (J'35) (SCLZFE), Power Engr., Combined Locks Paper Co., Combined Locks, Wis.
Hellekson, Owen Hellek (J'44) (ZKF), Sales Engr., Minneapolis-Honeywell Regulator Co., 35 Webster St., Hartford, Conn.
Hellenberg, Clare Ernest (J'41) (HAM), Jr. Engr., Vickers, Inc., 1400 Oakman Blvd., Detroit 32; *for mail,* 20085 Rosemont, Detroit 19, Mich.
Heller, Edgar W. ('23) (HCB), Pres., Lecourtenay Co., 5 Maine St., Newark 5, N.J.
Heller, Elmer W. (J'43), Project Engr., Stratos Corp., 122-126 W. Main St., Babylon; *for mail,* 724 Mador Court, Far Rockaway, L.I., N.Y.
Heller, Harold P. (J'37) (YZJKL), 7631 W. Berwyn Ave., Chicago, Ill.
Heller, Lewis W. ('18; '23; '35) (SFK), Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Heller, M. Mendell ('24; '35; '35), United Gas Pipe Line Co., Box 1407, Shreveport, La.
Heller, Milton (J'42), Lt., 70th Ferrying Sqd., Berry Field, Nashville, Tenn.
Heller, Paul R. (J'38) (BNSCKJ), Mech. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* Pennsylvania Ave. Ext., Irwin, Pa.
Heller, William E. (J'41) (ABM), Lt., 878th Bomb Sqd., 499th Bomb Gp., A.P.O. 237, c/o Postmaster, San Francisco, Calif.
Helliwell, Wilfred (J'39), 25 Horse Shoes Lane, Sheldon, Birmingham 26, England.
Hellman, Wilbert M. (J'41), Lt. (j.g.), c/o Dir. of Philippine Bur., Coast & Geodetic Survey, Manila, P.I.
Helm, John Leslie (J'45), 106 Morningside Dr., New York 27, N.Y.
Helm, Paul F. ('38) (CFKS), Asst. to V.P., Indianapolis Power & Light Co., 17 N. Meridian St., Indianapolis 6; *for mail,* 4760 Washington Blvd., Indianapolis 5, Ind.
Helmer, Clifford Andrew (J'42) (AMC), Ensign, U.S.N.R., Div. Head, Aircraft Maint., Bachelor Officers' Quarters 153, Naval Air Sta., Glenview, Ill.
Helmer, Nicolas Arthur ('18) (LCMKZS), Helmer Co., Box 54, Plainfield, N.J.
Helmick, Walter E. ('20; '24; '35) (BHES), *Student Award,* '19; Sr. Engr., Exploitation Dept., Shell Oil Co., Inc., 1008 W. 6th St., Los Angeles 14; *for mail,* 920 Roxbury Rd., San Marino 9, Calif.
Helmig, Ralph Henry ('45), M.M., Eicor, Inc., 1501 W. Congress St., Chicago; *for mail,* 2624 N. Harding Ave., Chicago 47, Ill.
Helmrich, George Bernard ('41) (CSO), Schedule Engr., Constr. Engrg., Detroit Edison Co., 2000—2nd Ave., Detroit '26; *for mail,* 32920 Lahser Rd., R.R. 5, Birmingham, Mich.
Helmstaedter, William Edward ('15) (YL), Mech. Engr., Plastics Div., Celanese Corp. of Am., 290 Ferry St., Newark 5; *for mail,* 46 St. Paul Ave., Newark 6, N.J.
Helpbringer, James N. ('17; '21) (SEKO), Ch. Elec. Engr., City Ice & Fuel Co., 3638 Olive St., St. Louis; *for mail,* 512 Tuxedo Blvd., Webster Groves 19, Mo.
Helquist, John E. (J'35) (HKS), c/o Naugatuck Chem. Co., Naugatuck, Conn.
Heltzel, William G. ('24; '35) (BCEH), Exec. V.P., Stanolind Pipe Line Co., Box 591, Tulsa 2; *for mail,* Ambassador Hotel, 7 W. 14th St., Tulsa, Okla.
Helvig, William J. (J'41), Turbine Mechanic, Gen. Elec. Co., Philadelphia; *for mail,* 647 Alter St., Hazleton, Pa.
Helwig, Alfred ('16) (SROFEK), Ch. Engr., Bush Terminal Co., Foot of 43rd St., Brooklyn 32; *for mail,* 176 Winthrop St., Brooklyn 25, N.Y.
Hem, Eilif Sigurdsen (J'19) (SHEF), Serv. & Erection Engr., Allis-Chalmers Mfg. Co., 50 Church St., New York 7; *for mail,* 702—45th St., Brooklyn 20, N.Y.
Hem, H. O. ('09), Ch. Engr., Toledo Scale Co.; *for mail,* 3009 Kenwood Blvd., Toledo, Ohio.
Hembree, Paul Pittman (J'43) (ABC), 3215 Avenel Ave., Nashville 11, Tenn.
Hemenway, Henry H. (J'41) (SKFLJH), Asst. to Dir. of Engrg., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Hemenway, Seymour H. ('23; '35) (BSOAJN), Supvr. of Sub-Contrg., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 610 Strath Haven Ave., Swarthmore, Pa.
Hemingway, Everard Luther ('44), 4125 Yuma Dr., Madison, Wis.
Hemman, Robert Edgar (J'42), (SKFZ), Tech. Engr., Boston Edison Co., 39 Boylston St., Boston; *for mail,* 171 Florence St., Roslindale 31, Mass.

Hempel, Edward H. ('41) (CDL), Asst. Prof., Indus. Engrg. Dept., Columbia Univ., 116th St. & Broadway, New York, N.Y.
Hempel, Herbert W. ('32; '42) (BCMJNO), Mech. Engr., Marsh Stencil Mch. Co., 707 E. B St.; *for mail,* Box 150, Belleville, Ill.
Hemsarth, John Henry, III (J'40) (SON), Lt. (Eng.), U.S.N.R., Bur. of Ships, Navy Dept., 7 Aberdeen Pl., Woodbury, N.J
Henderer, Willard Everett, II (J'43) (MSK), Jr. Mech. Engr., A. L. Henderer's Sons, Wilmington 55; *for mail,* 1213 Delaware Ave., Wilmington 19, Del.
Henderson, Bruce Doolin (J'43) (CAJMY), Pur. Agt., Westinghouse Elec. Corp., 95 Orange St., Newark 1; *for mail,* 14 Reeve Circle, Millburn, N.J.
Henderson, Claude Earl ('45) (BEFKS), Mar. Surveyor, U.S. Maritime Comm., War Shipping Admin., Rm. 911, 39 Broadway, New York 6; *for mail,* 97 Elm St., Staten Island 10, N.Y.
Henderson, Curtis L. ('23) (CE), Pres., Vickers Petroleum Co., 201 Wheeler Kelly Hagny Bldg., Wichita, Kan.
Henderson, Douglas ('31) (SFEO), V.P., Fuel Engrg. Co. of N.Y.; Mem. of Firm, Gould, Henderson & Brunjes, 215—4th Ave., New York 3, N.Y.
Henderson, Edgar Frank (J'48) (KECY), 1828 W. Riverside Ave., Spokane 9, Wash.
Henderson, Frederick Robert (J'42) (CMD), Assoc. Prof. Indus. Engrg., Northeastern Univ., Boston 15, Mass.
Henderson, George Andrew (J'40) (JMNGLC), Asst. Prof. Mech. Engrg., Engrg. Hall, Vanderbilt Univ., Nashville 4, Tenn.
Henderson, George Louis ('42) (HKLZ), Engr., West. Gear Works, 417—9th Ave. S., Seattle 4; *for mail,* 312—31st Ave., Seattle 22, Wash.
Henderson, George T. ('30; '41) (CMDNBL), Plant Mgr., Hinde & Dauch Paper Co., 5110 Penrose St., St. Louis 15, Mo.
Henderson, Gordon Kenneth (J'44) (KMNHCD), Process & Devel. Engr., Am. Optical Co., Mechanic St., Southbridge, Mass.
Henderson, Henry Hardy (J'45) (BKHLS), Struc. Design Engr., Austin Co., 16112 Euclid Ave., Cleveland 12, Ohio.
Henderson, Herbert ('14; '16) (EL), V.P., Engrg. Dept., Gulf Oil Corp., Box 1166, Pittsburgh 30, Pa.
Henderson, Jack Robert (J'45) (CMO), 611 Homestead Ave., Peoria 3, Ill.
Henderson, Jas. Mashburn (J'42) (JMBONC), Design Engr., West Fabricating Plant, Aluminum Co. of Am., Alcoa; *for mail,* 298 Gamble St., Maryville, Tenn.
Henderson, John Robinson ('33; '43) (BKS), Aircraft Gas Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 9 Sheridan Rd., Swampscott, Mass.
Henderson, Robert Dale (J'39) (BKN), Engr., Design Sec., Puget Sound Naval Shipyard; *for mail,* Apt. A, 934 Highland, Bremerton, Wash.
Henderson, Robert H. ('07), Mgr., Henderson Elec. Co., Ampere; *for mail,* 181 Greenwood Ave., East Orange, N.J.
Henderson, Robert William (J'37) (BKNYAH), Leader, Engrg. Gp., War Dept. Contract, Univ. of Calif., Box 1663, Santa Fe, New Mex.
Hendrich, Henry Alfred (J'30) (BNR), Mech. Designer, Internatl. Business Mchs. Corp., North St.; *for mail,* 512 South St., Endicott, N.Y.
Hendrick, Wallace M. (J'12), Pres., Atlantic Concrete Pipe, Inc., W. John St.; *for mail,* Box 247, Hicksville, N.Y.
Hendrickson, Elmer L. ('44), Sears, Roebuck & Co., 3625 E. 15th St., Kansas City; *for mail,* 717 E. 71st Terrace, Kansas City 5, Mo.
Hendrickson, G. A. ('26; '32; '35), Dean, College of Engrg., Lawrence Inst. of Tech., Highland Park; *for mail,* 7500 Normile, Dearborn, Mich.
Hendrickson, George S. ('20; '35) (ZSNC), V.P., Republic Flow Meters Co., 2240 Diversey Blvd., Chicago 47, Ill.
Hendrickson, Ole Quist (J'35) (CDKO), Sunshine Ave., Hydes, Md.
Hendrickson, Robert Leroy (J'42) (ABSEHR), Engr., Turbine Div., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 21 Concord St., Scotia 2, N.Y.
Hendrie, Charles Francis ('21; '29; '35) (SKLC), Mgr., Condenser & Ejector Sec., Steam Power Div., Worthington Pump & Mchy. Corp., Harrison, N.J.
Hendrix, Houston William ('44) (CEFS), V.P., Mackenzie Engrg. Co., Broad St. Sta. Bldg., Philadelphia 3; *for mail,* 148 Trent Rd., Philadelphia 31, Pa.
Hendry, Wickliffe B. (J'40), Mech. Engr., Univ. of Minn., Minneapolis; *for mail,* Box 101, Excelsior, Minn.
Heneghan, Sherald F. (J'46), Design Engr., Stand. Oil Co. (Ind.); *for mail,* 451 Hamilton Ave., Wood River, Ill.
Henke, Werner (J'38) (CE), Henke Engrg. Co., Box 754, Lafayette, La.
Henley, Knowles H. (J'39), West. Elec. Co., Inc.; *for mail,* 1704 Homestead Ave., Atlanta, Ga.
Henn, Russell M. (J'43) (EKS), 43 Oak St., Yonkers, N.Y.
Hennessey, Woodrow Lawrence (J'42) (FKS), Lt., Corps of Engrs., 8th Engr. Sqd., A.P.O. 201, c/o Postmaster, San Francisco, Calif.; *for mail,* 471—44th St., Brooklyn, N.Y.
Hennig, Fritz Otto ('34; '45) (BSNKEH), Ch. Designer, Research Dept., Elliott Co., Jeannette; *for mail,* R.D. 3, Greensburg, Pa.
Hennigan, Francis A. ('42) (CLMO), Indus. Engr., Universal Atlas Cement Co., 135 E. 42nd St., New York, N.Y.
Henning, H. Clay ('25) (SFH), Div. Engr., Charge Pumping Water, St. Louis Water Div., 1640 S. Kingshighway, St. Louis 10, Mo.
Henninger, Frederick Albert ('43) (AJP), Rulane Tank Co., Box 29, Charlotte 1, N.C.
Henninghausen, Louis Kemp (J'40) (ACS), Maj., Hdq., 2nd Bn., 409th Inf., A.P.O. 470, c/o Postmaster, New York, N.Y.
Henofer, John P. ('20) (SF), Mech. Engr., Ebasco Internatl. Corp., 2 Rector St., New York 6, N.Y.; *for mail,* 104 Wood St., Rutherford, N.J.
Henriksen, Paul F. (J'45) (CJM), 1024 N. Lawler Ave., Chicago 51, Ill.
Henrikson, Karl G. (J'40) (BCMNS), Lub. Engr., Socony-Vacuum Oil Co., Inc., Hanna Bldg., Cleveland 15; *for mail,* 3254 Berkeley Ave., Cleveland Heights 18, Ohio.
Henrikson, Wm. (J'36), Design Engr., Keystone Mfg. Co., 151 Hallet St., Boston; *for mail,* 35 Ponkapoag Way, Canton, Mass.
Henry, Alex. S., Jr. ('16; '25; '35), Estimator, Brunswick-Kroeschell Co., 203 Fulton St.; *for mail,* 300 Central Park West, New York, N.Y.
Henry, Howard J. (J'37) (BES), Design Engr., Chicago Pneumatic Tool Co.; *for mail,* 305—8th St., Franklin, Pa.
Henry, Jacob Schermerhorn ('43), Mgr., New England Dist., Safety Car Htg. & Ltg. Co., 230 Park Ave., New York, N.Y.
Henry, John Adams ('45) (SCEM), Assoc. in Mech. Engrg., Univ. of Ill., 205 Mech. Engrg. Lab., Urbana, Ill.
Henry, Jos. S. ('11), Cons. Engr., Austral Malay Tin Dredging Ltd.; *for mail,* Barrington, Hargreaves St., Castlemaine, Victoria, Australia.
Henry, Otto H. ('20; '25; '35) (BJL), Assoc. Prof. Metal. Engrg., Poly. Inst. of Brooklyn, 99 Livingston St., Brooklyn, N.Y.
Henry, Wm. M. (J'17), Ch. Oper., Waterside Stas., Consltd. Edison Co. of N.Y., Inc., 666—1st Ave.; *for mail,* 300 Central Park West, New York, N.Y.
Hense, John A. (A'42) (EHK), Rm. 417-E, 30 Church St., New York, N.Y.
Hensel, Frank Gerard ('42) (BHKZ), Ch. Centrifugal Test Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1; *for mail,* Manlius Rd., Fayetteville, N.Y.
Henseler, William James (J'36) (LO), Lt. (j.g.), U.S.N.R., U.S.S. *Vogelgesang,* DD-862, F.P.O., New York, N.Y.; *for mail,* 30 Cooper Pl., Weehawken, N.J.
Henshall, P. P. ('15; '20; '35), Centre Hall, Pa.
Henshaw, Charles Norton ('26; '35) (NBSEJ), Research, Design Engr., Pal Blade Co., Inc., South River St.; *for mail,* Box 304, Plattsburgh, N.Y.
Hensinger, Claude E. (J'46), 1349 Turner St., Allentown, Pa.
Henslee, Homer Elbert (J'44) (OCL), Contract Engr., Foster Wheeler Corp., 165 Broadway, New York 6; *for mail,* 56-49—141st St., Flushing, N.Y.
Henszey, Roy O. ('17; '35) (CKLS), Ch. Engr., Carnation Co.; *for mail,* 115 Woodland Lane, Oconomowoc, Wis.
Henwood, George Lee (J'42) (EFK), Ensign, O-V (S), U.S.N.R.; *for mail,* Syracuse, Ind.
Henwood, James B. ('34; '42) (AFL), Devel. Engr., Selas Corp. of Am., Erie Ave. & D St., Philadelphia; *for mail,* 301 Kent Rd., Bala-Cynwyd, Pa.
Henze, Otto C. W. (MNWYZ), Cons. Engr., Designer, 601 W. 148th St., New York 31, N.Y.
Henze, Philip S. (J'41) (ACR), Liaison Engr., Hamilton Stand. Propellers Div., United Aircraft Corp., 400 S. Main St.; *for mail,* 119 Naubuc Ave., East Hartford 8, Conn.
Hepburn, James William (J'29) (EHS), Application Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 52 Edgar Pl., Nutley, N.J.
Hepburn, Norman Clickner ('43) (EFS), Chief, Plumbing & Refrig. Sec., Asst. to Chief of Engrg. Sub-Div., Constr. Serv., U.S. Veterans Admin.; *for mail,* 1610 Webster St., N.W., Washington, D.C.
Hepke, Wm. C. ('30) (CGL), Ch. Engr., Thomas M. Royal & Co., 5800 N. 7th St., Philadelphia; *for mail,* 1407 Dorset Lane, Overbrook Hills, Pa.
Heppenheimer, Herman (J'34), Pvt., 13th Inf., U.S.A., Ft. Jackson, S.C.
Herb, Charles Oliver ('28; '35) (MJC), Managing Editor, *Machinery,* 148 Lafayette St., New York 13, N.Y.
Herb, James Hamilton (J'42) (DFBSLN), Design & Application Engr., Pa. Crusher Co., Liberty Trust Bldg., Philadelphia; *for mail,* 1415 Navahoe Dr., Pittsburgh 16, Pa.
Herbert, Arthur Royland ('44) (BNSJDK), Dist. Mgr., Philadelphia Gear Works, Inc., 111 W. Washington St., Chicago 2; *for mail,* 318 S. Delphia Ave., Park Ridge, Ill.
Herbert, Charles Graham ('00) (YNOE), Retired; 2659 N. Tanoble Dr., Altadena, Calif.
Herbert, Frederick Davis ('99; '07; F'41) (A), Sr. Partner, Kearfott Engrg. Co., 117 Liberty St., New York 16, N.Y.
Herbert, James Stewart (J'44) (CDM), Pvt., 311 N. Main St., Milltown, N.J.
Herbert, Leslie Elmer ('30; '42) (CMNSKB), 1512 Pentridge Rd., Baltimore 12, Md.
Herbst, Robert See (J'43) (CKHMGA), Design Engr., Product, Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.
Hering, Harold E. (J'41) (MNJC), Ensign, U.S.N.R., U.S.S. *Salerno Bay,* CVE-110, F.P.O., New York, N.Y.; *for mail,* 2715 W. 23rd Pl., Chicago 8, Ill.
Hering, Henry Hall (J'41) (NBAG), Mech. Design Engr., Gen. Elec. X-Ray Corp., 2012 W. Jackson Blvd., Chicago 12; *for mail,* 1633 E. 83rd St., Chicago 17, Ill.
Herkimer, Harold (J'42) (SEKFCH), Project Mgr., Design Engr., Raisler Corp., 129 Amsterdam Ave., New York 23; *for mail,* 140 Riverside Dr., New York 24, N.Y.
Herkimer, Herbert ('43) (CEFKLM), Ch. Engr., Herkimer Engr. Associates, 140 Riverside Dr., New York, N.Y.
Herking, Paul William (J'43) (SCMEJ), 1st Lt., Ord. Dept., Ravenna Arsenal, Apco; *for mail,* 4300—34th Ave., Cincinnati 9, Ohio.
Herlin, Robert G. (J'38) (LBNJCH), Sqd. Leader, Engrg. Dept., Tex. Co., Box 712, Port Arthur, Tex.
Herman, E. Olney ('22; '35), V.P., Mgr., Tiffany Enameled Brick Co., Box 41; *for mail,* Box 32, Momence, Ill.
Herman, Kenneth Ray ('44) (HNASBM), V.P., Gen. Mgr., Vickers, Inc., 1400 Oakman Blvd., Detroit 32, Mich.
Herman, Rudolph Vincent ('45) (CDO), Mech. Engr., Briggs Mfg. Co., 11631 Mack, Detroit 14; *for mail,* 33524 Jefferson St., St. Clair Shores, Mich.
Herman, Sydney Keith (J'45) (MDCWYB), 61 Lyndhurst Ave., Toronto, Ont., Can.
Herman, T. August (J'38) (HMS), Test Engr., Worthington Pump & Mchy. Corp., 401 Worthington Ave., Harrison; *for mail,* 15 Myrtle Ave., Newark, N.J.
Hermann, Clarence Carl ('43) (CDMNO), Cons. Engr., 1342 Harvard Rd., Grosse Pointe Park 30, Mich.
Hermann, Philip (J'45) (NHKJZL), Lt., U.S.N.; 621 N. Poinsettia Pl., Los Angeles 36, Calif.
Hermes, Richard Manning ('46) (BGO), Asst. Prof., Univ. of Santa Clara, Santa Clara, Calif.
Hermes, W. Dan (J'41) (CMJND), Lt. Comdr., U.S.N.R., Project Engr., Bur. of Ord., Navy Dept., Naval Ord. Plant, Center Line, Mich.; *for mail,* 315 N. Chatham St., Janesville, Wis.
Hernandez A, Roberto Cesar (J'43) (BRS), Oficina de Talleres, FF. CC. Nales-Pacifico, Chipichape, Cali, Valle, Colombia, S.A.
Herndon, Luther Joe, Jr. (J'46) (JP), Box 966, Duncan, Okla.
Herndon, Lyle Kermit ('44) (LHKFCD), Assoc. Prof. Chem. Engrg., Ohio State Univ., Columbus 10; *for mail,* 2303 Onandaga Dr., Columbus 8, Ohio.
Hernried, Erwin George (J'42) (KHLBSJ), Engr., Golden State Co., Ltd., 425 Battery St., San Francisco 11, Calif.
Herod, Wm. R. (J'20), 570 Lexington Ave., New York, N.Y.
Herold, Richard (J'41) (ESRTAB), Pres., Sulzer Bros., Ltd., New York, 50 Church St., New York 7, N.Y.
Herr, Benjamin Musser ('15) (HKOS), Owner, Herr-Harris Co., 545 William Penn Pl., Pittsburgh 19, Pa.
Herr, Frederick Williams (J'42) (CLM), 318 W. 7th St., Auburn, Ind.
Herr, John Alexander ('45) (DCBN), Asst. Ch. Bridge & Tramway Engr., Am. Steel & Wire Co., Box 1709, New Haven 7; *for mail,* 51 Spring Garden St., Hamden, Conn.
Herr, John G. (J'40) (AGC), 322 Pine St., Middletown, Pa.
Herr, Wm. A. ('22; '28) (FLS), Sr. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5; *for mail,* 122 Tyson Ave., Glenside, Pa.
Herrick, Carl Albert ('12) (BNWY), Prof. Math. & Mechanics, Univ. of Minn., Minneapolis 14; *for mail,* 4324 W. 44th St., Minneapolis 10, Minn.
Herrick, Daniel A. ('15; '19; '24) (CSZ), Mgr., Sales, d'Este Div., Am. Chain & Cable Co., Tulpehocken St., Reading; *for mail,* 1427 Cleveland Ave., Wyomissing, Pa.
Herrick, Edson P. ('18; '20; '35) (CDJLMN), Sales, Stevens Engrg. Co., 7 S. Main St., West Hartford; *for mail,* R.F.D. 2, Manchester, Conn.
Herrick, Gerard Post ('39) (A), Pres., Vertoplane Devel. Corp., 25 E. 76th St., New York 21, N.Y.
Herrick, Robert William (J'42) (BDH), 625 W. Hillcrest, Monrovia, Calif.
Herrick, Thomas J. (J'36) (AB), Asst. Prof. Aero. Engrg., Purdue Univ., Lafayette, Ind.
Herrmann, Edward Matthew (J'45) (YOCKMD), Lt., U.S.N.R., S(E1), David W. Taylor Model Basin, Carderock; *for mail,* 5104 Eugene Ave., Baltimore, Md.
Herrmann, John F. ('18; '22; '35), Plant Mgr., Mech. Engr., Herrmann & Grace Co., 671-89 Bergen St., Brooklyn; *for mail,* Box 402, Brightwaters, L.I., N.Y.
Herron, James Hervey ('97; '05; F'36) (FHJLSY), *Manager,* '22-'25, *Vice-President,* '34-'36, *President,* '37; Pres., James H. Herron Co., 1360 W. 3rd St., Cleveland 13, Ohio.

Herron, William L. ('21; '32) (HNZO), Sales Engr., Worthington-Gamon Meter Co., 296 South St., Newark 5, N.J.; *for mail*, Box 234, Oswego, N.Y.

Herschmann, Arthur J. ('00; '05) (CJS), Mech. Engr., A. J. Herschmann, 50 Church St., New York, N.Y.

Hersey, Donald Samuel ('45) (ABCEKS), Engrg. Asst. to Chief of Devel., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St.; *for mail*, 160 Holland Lane, East Hartford 8, Conn.

Hersey, Mayo Dyer ('12; '14; '19; F'38) (BKNMJZ), Research Assoc., Mech. Engrg., Bldg. 31, Mass. Inst. of Tech., Cambridge 39, Mass.

Hershey, Albert Eby ('44) (KEFABH), Research Engr., Research Labs., Westinghouse Elec. Corp., East Pittsburgh, Pa.

Herst, Richard Jacob (J'45) (BKN), Devel Engr., S.A. Platt, 325 W. Huron St., Chicago 19; *for mail*, 1926 Berteau Ave., Chicago 13, Ill.

Herstedt, Harold B. R. (J'43) (ABHNSC), Mech. Engr., Constr. Sec., Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field; *for mail*, 2521 Wilbur St., Oakland 2, Calif.

Hertel, Crawford W. (J'40) (AMYSC), 221 Center St., East Lansing, Mich.

Hertenstein, Edgar William, Jr. (J'42) (MC), Engr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore 24, Md.

Herty, Frank B. ('32; '35) (BLS), Prof. Mech. Engrg., Univ. of S.C.; *for mail*, 814 Poinsettia Rd., Columbia 50, S.C.

Hertz, David Bendel (J'43) (MCJKO), Devel. Engr., Celanese Corp. of Am., 290 Ferry St., Newark 5, N.J.; *for mail*, 410 E. 57th St., New York 22, N.Y.

Hervey, Eugene ('37; '45) (CJOSLM), Ch. Engr., Natl. Smelting Co., 6700 Grant Ave., Cleveland 5, Ohio.

Herwald, Seymour Willis (J'39) (ABZLSO), Cent. Engr., Devel., Westinghouse Elec. Corp., Braddock Ave., East Pittsburgh; *for mail*, 5557 Hobart St., Pittsburgh 17, Pa.

Herzfeld, Siegfried L. (J'45) (ZKBOHN), 815 W. 106th St., New York 25, N.Y.

Herzog, Al. R. (J'40), Asst. Supt., Kettle River Co., Edwardsville, Ill.

Herzog, John Henry ('37) (BEFOS), Ch. Engr., San Francisco Brewing Corp., 490—10th St., San Francisco, Calif.

Herzog, Michael S. (J'38), 317 W. 89th St., New York, N.Y.

Hescheles, Charles A. ('25; '32; '35) (SFELZK), Mech. Engr., U.S. Rubber Co., 1230—6th Ave., New York 20; *for mail*, 1710 Ave. K, Brooklyn 30, N.Y.

Heslop, Paul Loveridge ('31) (HJBE), Hyd. Engr., Ebasco International Corp., Caixa Postal 883, Rio de Janeiro, Brazil, S.A.

Hess, Ernest Elliott ('17; '25; '35) (BJNOWY), Route 3, Lufkin, Tex.

Hess, Frederic Otto ('45) (KFNOMC), Pres., Sales Corp. of Am., Erie Ave. & D St., Philadelphia 34; *for mail*, Kenilworth, Philadelphia 44, Pa.

Hess, John R. (J'37) (ARNYCG), Designer Draftsman, Edw. G. Budd Mfg. Co., Red Lion & Veree Rds., Philadelphia; *for mail*, 880 Carver St., Philadelphia 24, Pa.

Hess, John Strider ('27) (KHSEFD), Ch. Engr., Refineries, Sinclair Refining Co., 630—5th Ave., New York 20, N.Y.

Hess, Paul Dale (J'37) Engr., Hyd. Dept., Allis-Chalmers Mfg. Co., Milwaukee, Wis.

Hess, Robert Graves (J'43) (CDR), Dir. of Wages, Methods & Budget, N.Y. Air Brake Co., Starbuck Ave.; *for mail*, 630 Sherman St., Watertown, N.Y.

Hess, Robert Joseph (J'42) (KLS), Ensign, Naval Air Sta., Junior Officers' Quarters 175, Sand Point, Seattle, Wash.; *for mail*, 1 Frothingham Ave., Jeannette, Pa.

Hess, Stanley E. (J'38) (CMBDS), Lunkenheimer Co., Cincinnati 14; *for mail*, 5113 Paddock Rd., Cincinnati 29, Ohio.

Hess, Walter John, Jr. (J'45) (ZLEJW), Asst. Engrg. Officer, U.S.S. *Charles Lawrence*, APD-37, F.P.O., San Francisco, Calif.

Hess, William Boyer ('44) (HSJBEC), Test Engr., Safe Harbor Water Power Corp., R.D. 2, Conestoga, Pa.

Hesse, Walter Kasper (J'40) (BJM), 95 Hilton Ave., Maplewood, N.J.

Hessellund, Regnar ('30; '37) (BES), Ch. Power Plant Engr., Cerro de Pasco Copper Corp.; *for mail*, Hotel Junin, Oroya, Peru, S.A.

Hession, John William, Jr. ('43), Research Engr., Columbia War Research, Natl. Defense Research Com., 668 Chandler, Columbia Univ., New York, N.Y.; *for mail*, Edgehill Rd., Darien, Conn.

Hesslein, Robert Jacques (J'45) (NMCDS), Ensign, D (L), U.S.N.R., U.S.S. *Puget Sound*, CVE-113, F.P.O., San Francisco, Calif.; *for mail*, Box 645, Montgomery 1, Ala.

Hestand, Rue S. (J'37) (CEMN), Asst. Ch. Engr., Am. Mfg. Co. of Tex., Box 710, Ft. Worth 1, Tex.

Heston, Robert B. (J'39), 1st Lt., Inf., U.S.A., Co. B, 1213th Reception Center, Ft. Niagara, N.Y.

Hetherington, Carl Frederick ('42) (DJ), Draftsman, Alvey Conveyor Mfg. Co., 3201 S. Broadway, St. Louis 18, *for mail*, 5643 Bartmer Ave., St. Louis 12, Mo.

Hetherington, Irvine J. (J'43) (LYKZC), U.S. Navy; 540 Locust St., Bristol, Pa.

Hettinger, Carl (J'39) (EKS), Lt., U.S.A., Camp Blanding, Starke, Fla.; *for mail*, 2608 Goodwood Rd., Baltimore 14, Md.

Hettinger, Charles ('32; '35) (CDLMNS), Plant Engr., Horn & Hardart Co., 600 W. 50th St., New York, N.Y.; *for mail*, 1208 The Strand, West Englewood, N.J.

Hettrick, Ames Bartlett ('34; '35) (CLSD), Works Mgr., Calco Chem. Div., Am. Cyanamid Co., Piney River, Va.

Hetzel, Lowell Hayward (J'34) (FDCMJ), Supply Supt., Stand. Lime & Stone Co., Millville, W. Va.

Hetzel, Theodore Brinton ('29; '37) (ENLSOA), Asst. Prof., Haverford College; *for mail*, 768 College Ave., Haverford, Pa.

Heumann, John P. ('32; '35) (NMJBLY), Devel. Engr., Remington Arms Co., Inc., 939 Barnum Ave., Bridgeport 2; *for mail*, 128 Smith St., Fairfield, Conn.

Heuser, Fred ('38; '45) (BCHM), Devel. Engr., Sterling Eng. Co., 1270 Niagara St.; *for mail*, 72 Delham Ave., Buffalo, N.Y.

Heuser, Henry Vogt (J'36) (CMSF), V.P., Henry Vogt Mch. Co., Louisville 10, Ky.

Heverly, Earl L. ('37) (BCM), Gen. Mgr., Norton Door Closer Co., 2900 N. Western Ave., Chicago 18, Ill.

Hewey, Robert Warren (J'40) (BCMN), Mech. Engr., Singer Mfg. Co., 803 E. Washington Ave.; *for mail*, 195 Shelton St., Bridgeport 8, Conn.

Hewitt, Arthur R. (J'40) (CNMG), Ord. Engr., Frankford Arsenal, Bridge St. & State Rd., Philadelphia 37; *for mail*, 6203 Erdrick St., Philadelphia 35, Pa.

Hewitt, Edward R. ('18) (EFP), Cons. Engr., 48 Gramercy Park, New York 10, N.Y.

Hewitt, Ellis Eberle ('44) (NRAHMZ), Ch. Engr., Westinghouse Air Brake Co., Wilmerding, Pa.

Hewitt, Frederick McCracken ('45) (NCM), Dist. Engr., Bijur Lub. Corp., 4301—22nd St., Long Island City 1, N.Y.; *for mail*, 34016 Oakland Rd., Farmington, Mich.

Hewitt, George Edward (J'43) (CNYBJM), Design Engr., Radar Div., Gilfillan Bros., Inc., 1815 Venice Blvd., Los Angeles 6; *for mail*, 1011-B Western Ave., Glendale 1, Calif.

Hewitt, John Vance, Jr. (J'39) (CA), Lt. Comdr., U.S.N.R., U.S.S. *White Marsh*, F.P.O., New York; *for mail*, 970 Park Ave., New York 28, N.Y.

Hewitt, Reginald William (J'27) (CMDBJY), c/o Automatic Tel. & Elec. Co., Ltd., Liverpool 7, England.

Hewitt, William Barker ('20) (CMNY), Devel. Engr., North & Judd Mfg. Co.; *for mail*, 15 Emmons Pl., New Britain, Conn.

Hewitt, William Henry (J'37) (CJS), Direct Engrg. & Factory Rep. (Field), Jas. Morrison Brass Mfg. Co., Ltd., 276 King St. W., Toronto 2; *for mail*, 11 Abbott Ave., Islington, Ont., Can.

Hewlett, Allen Morton (J'41) (FGKLSZ), Factory Supt., Hutchinson Sugar Plantation Co. Ltd., Naalehu, Kau, Hawaii, T.H.

Heymann, Clifford Donald (J'41) (MNTCOA), Seaman, 2nd Class, U.S.N.R., Instr., Naval Acad. Prep. Sch. Crew, Naval Trng. Center, Bainbridge, Md.; *for mail*, 8802—104th St., Richmond Hill, L.I., N.Y.

Heyward, T. C., Jr. (J'39) (BCS), Dist. Engr., for T. C. Heyward, 1408 Independence Bldg., Charlotte 2; *for mail*, 1812 Winthrop Ave., Charlotte 3, N.C.

Heyward, Theo. C. ('15; '19) (FKS), Dist. Sales Mgr., Combustion Engrg. Co., Inc., 1408 Independence Bldg.; *for mail*, 2448 Mecklenburg Ave., Charlotte, N.C.

Heywood, Harold Lester ('41) (NJMHB), Supvr., Devel. Engrg., Kearney & Trecker Corp., 6784 W. National Ave., West Allis 14, Wis.

Hibbeler, Charles Leroy (J'41) (JMC), Plant Engr., John Nooter Boiler Works Co., 1400 S. 2nd St., St. Louis 4; *for mail*, 128 W. Etta, Lemay 23, Mo.

Hibbeler, Gletner Leonard (J'41) (LKDCZ), Design Engr., Monsanto Chem. Co., 1700 S. 2nd St., St. Louis; *for mail*, 5330 Pershing Ave., St. Louis 12, Mo.

Hickerson, Fred Rodrigo (J'43), Ridgewood Ave., Mt. Tabor, N.J.

Hickman, Charles Devitt ('14) (SFEO), Cons. Engr., Otis Bldg., Philadelphia 2; *for mail*, 2416 Bryn Mawr Ave., Philadelphia 31, Pa.

Hickman, Hally B. (J'34) (CEL), E. I. du Pont de Nemours & Co., Belle; *for mail*, 304 Main Dr., Charleston, W. Va.

Hickox, Charles M. ('40) (SFDKHZ), Mech. Engr., H. K. Ferguson Co., 1650 Hanna Bldg., Cleveland 15; *for mail*, 19627 Winslow Rd., Shaker Heights 22, Ohio.

Hickox, George Harold ('44) (HBSK), Sr. Hyd. Engr., Hyd. Lab., Tenn. Valley Authority, Norris, Tenn.

Hicks, Earl Junior (J'43) (ABHN), Stress Analyst, Structures Dept., Beech Aircraft Corp., Wichita 1; *for mail*, 341 N. Hillside Ave., Wichita 8, Kan.

Hicks, James Ryan ('32; '39) (ZSLH), Tech. Editor, Socony-Vacuum Oil Co., Inc., 26 Broadway, New York; *for mail*, 120 Gorey Brook Rd., North Tarrytown, N.Y.

Hicks, Melvin Leroy (J'45), Instr., Mech. Engrg. Lab., Univ. of Kan.; *for mail*, 1827 Massachusetts St., Lawrence, Kan.

Hicks, Robert William (J'42) (ABE), Ensign, U.S.N.R.; *for mail*, 1008 N. Walnut St., Brazil, Ind.

Hickson, William McLean ('45) (COS), Dist. Supt. of Erection, Babcock & Wilcox Co., 2511 Carew Tower, Cincinnati 2, Ohio.

Hider, George Turner ('11; '16; '35) (DES), Pres., Gen. Mgr., Hider Bros., Inc., Box 386, Lake Providence, La.

Hidley, Robert Wendell (J'42) (AMN), Mech. Engr., Design, Gen. Elec. Co., Schenectady; *for mail*, 227 S. Main Ave., Albany 3, N.Y.

Hiebeler, Harry G. ('19; '25; '35) (SECF), Asst. Supt., Power, Houston Ltg. & Power Co., Elec. Bldg., Houston; *for mail*, 5335 Blythewood, Houston 4, Tex.

Hiers, George O. ('28) (JNBOML), Metallurgist, Research Labs., Natl. Lead Co., 105 York St., Brooklyn 1, N.Y.

Hiersch, Frederick Alexander (A'43) (KBE), Sr. Project Engr., Al-Fin Corp., 184-10 Jamaica Ave., Jamaica 1; *for mail*, 112-07—177th St., St. Albans 12, L.I., N.Y.

Higbie, Vinton ('31; '35) (CARBJK), Pres., Gen. Mgr., Atlantic Casting & Engrg. Corp., 721 Bloomfield Ave., Clifton; *for mail*, 12 Llewellyn Rd., Montclair, N.J.

Higginbotham, Oscar ('26; '33) (AKS), Central Esparia, Perico, Cuba.

Higgins, Aldus Chapin ('16), Chmn., Exec. Com., Norton Co., 1 New Bond St., Worcester 6, Mass.

Higgins, Alexander ('36) (SEFMLN), Cons. Engr., 3620-8 A St. W., Calgary, Alta., Can.

Higgins, Bradley Carter (J'42) (CJMDBA), Asst. Plant Engr., Worcester Pressed Steel Co., 100 Barber Ave., Worcester; *for mail*, 12 Somerset St., Worcester 2, Mass.

Higgins, Charles Lewis (J'43) (NBZHMK), Lt., Design Sec., Supersonic Wind Tunnels Lab., Aberdeen Proving Ground, Md.; *for mail*, 4 Verdon St., North Plainfield, N.J.

Higgins, Edwin Melbourne (A'30) (NCLT), Ch. Engr., Master Lubricants Co., Meadow & Jackson Sts., Philadelphia, Pa.; *for mail*, 18 Hovey St., Watertown 72, Mass.

Higgins, John T. ('43) (SEFC), Asst. Ch. Engr., Prudential Ins. Co. of Am., 763 Broad St., Newark, N.J.

Higgins, John Woodman ('02; '13) (BCGJLM), Pres., Treas., Worcester Pressed Steel Co., 100 Barber Ave., Worcester 6, Mass.

Higgins, Nathan Bert ('26) (CHS), Pres., Safe Harbor Water Power Corp., Lexington Bldg.; *for mail*, Box 2097, Baltimore 3, Md.

Higgins, Roland Seymour ('44) (CZSFJK), 16620 Baylis, Detroit 21, Mich.

Higgins, Stephen Preston, Jr. (J'44), Mass. Inst. of Tech., Cambridge; *for mail*, 338 High St., West Medford 55, Mass.

Higgins, Theodore J., Sr. ('22; '27; '35) (CFS), Div. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York, N.Y.; *for mail*, 184 Jefferson Ave., North Plainfield, N.J.

Higgins, Theodore John, Jr. (J'41) (SKFHCN), Jr. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York, N.Y.; *for mail*, 184 Jefferson Ave., North Plainfield, N.J.

Higginson, Edmund Elder ('34) (FKLOST), Ch. Engr., Olson Rug & Carpet Mfrs., 2800 N. Crawford Ave., Chicago 41, Ill.

Higginson, Thomas H. ('21) (BCHO), Managing Dir., Automatic Sprinkler Co. of Can. Ltd., Jeanne Mance, Montreal, Que., Can.

Highum, Orvie (J'37) (CDNSEC), Mech. Engr., Riverside Cement Co., Riverside, Calif.

Higley, Frank Raymond ('18; '25; F'44) (E), Pat. Lawyer, 1020 Leader Bldg., Cleveland 14; *for mail*, 3009 Fairfax Rd., Cleveland Heights 18, Ohio.

Higman, James (J'38) (EMS), Mar. Engr., Mar. Dept., Union Oil Co. of Calif., Wilmington; *for mail*, 8736 Shoreham Dr., West Hollywood 46, Calif.

Hilands, William H. (J'36), Comdr., U.S.N., 4807—43rd Pl., N.W., Washington 16, D.C.

Hilbert, Wm. Michael ('31) (BDFJKS), Ch. Engr., U.S. Metals Refining Co., Carteret; *for mail*, 15 Munsee Dr., Cranford, N.J.

Hilbish, Frederick George (J'45) (CMNOYH), Ensign, U.S.N.R., Engrg. Officer, U.S.S. *Caperton*, DD-650, F.P.O., New York, N.Y.; *for mail*, Piney River, Va.

Hilborn, William Dwight (J'41) (SZL), Serv., Sales Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio; *for mail*, 8250 Park Place Blvd., Houston 12, Tex.

Hildebrand, H. Edw. ('20; '25; '35), V.P., Dir. of Engrg., Continental Baking Co., 630—5th Ave., New York, N.Y.

Hildenbrand, Charles Frederick (J'35) (SCMF), Engr., Power Prod., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington Bldg., Baltimore 3; *for mail*, 639 Regester Ave., Anneslie, Baltimore 12, Md.

Hilderbrand, Ewing Andrew (J'41) (EFJ), Field Engr., RCA Serv. Co., Inc., Radio Corp. of Am., Bldg. 5-6, Camden 2, N.J.

Hildestad, Harold Lyle (J'44), Devel. Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester; *for mail*, 313 Dartmouth Ave., Swarthmore, Pa.

Hildreth, William Osgood ('33; '30) (DABJMN), Engr., Lamson Corp., Lamson St., Syracuse 1; *for mail*, 122 Rigi Ave., Syracuse 6, N.Y.

Hilera, Pedro (J'45) (CMS), Engr., P.R. Pulp & Paper Corp., P.O. Box 2072, San Juan, P.R.

Hiles, Elmer K. ('00; '06), Gulf Oil Corp., 458 Gulf Bldg., Pittsburgh, Pa.
Hilgartner, George Henry ('13) (BFHRS), Retired; 3013 Brook Rd., Richmond 22, Va.
Hilgert, Adolph John (J'42) (BKSZ), Design Engr., Johnson Serv. Co., 507 E. Michigan St., Milwaukee 2; *for mail,* 1720-A W. Meinecke Ave., Milwaukee 6, Wis.
Hilke, John Lewis (J'43), Student Test Engr., Gen. Elec. Co., Schenectady; *for mail,* 140 Otsego St., Canajoharie, N.Y.
Hilker, Harold William (J'42) (BCMNSZ), Mech. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail,* 1832 Willowhurst Rd., Cleveland 12, Ohio.
Hill, Albert J. ('32; '41) (CMS), Ch. Engr., Haytian Am. Sugar Co., Port-au-Prince, Haiti, W.I.
Hill, Arthur Lefevre ('24; '37) (MZTCFY), V.P., Natl. Fuse & Powder Co., 3801 Delgany St., Denver 5; *for mail,* 1033 Humboldt St., Denver 3, Colo.
Hill, Arthur Middleton ('39) (SEK), Prof. Heat Engrg., Tulane Univ., New Orleans 15, La.
Hill, Charles F. (J'41) (CJM), Lt. (j.g.), U.S.N.R., 10646 Ave. F, Chicago, Ill.
Hill, Chas. H. (A'07) (B), Retired; 628 Foss Ave., Drexel Hill, Pa.
Hill, Charles St. John (J'39) (ASR), Sr. Test Engr., Wright Aero. Corp., Beckwith Ave., Paterson 3; *for mail,* 152 Valley View Ave., Ridgewood, N.J.
Hill, Charles Strunk, Jr. ('45) (MDJOWZ), Cons. Engr., 301 S. 15th St., Philadelphia 2, Pa.
Hill, E. Rowland ('07) (ERS), V.P., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.
Hill, Ebenezer ('00; '07), c/o A.S.M.E., 29 W. 39th St., New York, N.Y.
Hill, Edgar Goodspeed ('21) (CEO), V.P., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Hill, Erland Victor (J'45) (ESJWRN), Co-Partner, Foreman, Hill Timber Co., Battle Ground, Wash.
Hill, Erle Gladstone ('45) (JCFK), Dir. of Metal. & Devel., Wheeling Steel Corp., Wheeling Steel Bldg., Wheeling, W. Va.
Hill, Frank (J'32) (FKOST), Co-Partner, Firm of Geo. Hill, Engr., Box B, Wellford; *for mail,* Duncan, S.C.
Hill, Fred Carlton ('21; '35) (CFER), Supt., Coke Plant, N.Y. Power & Light Corp., Troy, N.Y.
Hill, Fred M. ('38) (BCMNS), Serv. Mgr., Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport; *for mail,* 203 Edison Rd., Bridgeport 14, Conn.
Hill, G. A. ('40), Supt. M.P. & Cars, Wheeling & Lake Erie Ry. Co., Brewster, Ohio.
Hill, George Blood (J'36) (CMBJ), Designing Engr., Dain Mfg. Co., Vine St.; *for mail,* 238 S. Adella St., Ottumwa, Iowa.
Hill, George Mitchell ('44) (CMSRHY), Supvr., Civ. Sec., Pub. Works Design, Naval Shipyard, Terminal Island; *for mail,* 238 Coronado Ave., Long Beach 3, Calif.
Hill, George William ('45) (ERS), Application Engr., Westinghouse Elec. Corp., 708 Masonic Temple Bldg.; *for mail,* 2800 Audubon St., New Orleans, La.
Hill, Harold Gordon ('35; '43) (KFE), Sub-Lt., Engr., Royal Canadian Naval Volunteer Reserve, Naval Serv. Hdq., Ottawa; *for mail,* 175 Clendenan Ave., Toronto, Ont., Can.
Hill, Harold Otto ('41) (HJMO), Asst. Ch. Engr., Bethlehem Steel Co.; *for mail,* 201 E. Goepp St., Bethlehem, Pa.
Hill, J. Lawrence, Jr. ('45) (GHC), Assoc. Prof. Mech. Engrg., Univ. of Rochester, Rochester 3, N.Y.
Hill, James Thomas ('32; '35) (BEFKS), Power Supt., Clinton Labs., Monsanto Chem. Co., Box 1991, Knoxville 11, Tenn.
Hill, John Cantwell, Jr. (J'41) (ACMBNJ), 6925 Ottawa Ave., Chicago 31, Ill.
Hill, Myron Francis ('32) (HYAKSZ), Pres., Gerotor Corp.; Propr., Durotor Lab., Edgewater Hillside, Westport, Conn.
Hill, Rowland Chester ('40) (CJL), Plant Engr., Simonds Saw & Steel Co., Intervale Rd., Fitchburg, Mass.
Hill, Rowland F. ('14; '23) (EJ), Pat. Engr., Natl. Supply Co., Bishop St.; *for mail,* 1951 Richmond Rd., Toledo, Ohio.
Hill, William Edwin (M'43) (COMD), V.P., Turck, Hill & Co., Inc., 10 Rockefeller Plaza, New York 20, N.Y.
Hill, William Epps (J'42), Wyatt Metal & Boiler Works, Dallas 2, Tex.
Hill, William Holcombe ('29; '35) (CKMS), Pres., Baldwin-Hill Co., 501 Klagg Ave., Trenton 2, N.J.
Hill, William Plummer ('33; '41) (BFHJKS), Asst. Supt., Mech. Dept., Bethlehem Steel Co.; *for mail,* 806 C St., Sparrows Point 19, Md.
Hill, William Steinert ('18; '35) (BFHKSW), Examiner, War Assets Corp., 340 Cent. Bldg., Seattle 1; *for mail,* 5716—31st Ave., N.E., Seattle 5, Wash.
Hillenbrand, Charles William (J'42) (FJP), Pvt.; 445 Edgemont St., Pittsburgh, Pa.
Hillendahl, Wesley H. (J'42) (AK), Naval Airship Trng. & Exper. Comd., Naval Air Sta., Lakehurst, N.J.
Hiller, James C. (J'40) (CMDLJA), Works Cost Engr., Columbia Steel Co.; *for mail,* Box 56, Torrance, Calif.
Hiller, John Dean (J'42), Ch. Engr., Ariz. Equitable Rating Office, 710 Title & Trust Bldg.; *for mail,* Box 2831, Phoenix, Ariz.
Hiller, Jos. L. (A'07) (BFJ), Cons. & Contrg. Engr., Hiller Beaters, Pine Island Rd., Mattapoisett, Mass.
Hiller, Nicolai Henry ('02) (K), Liquidating Trustee, Carbondale Mch. Co., 27 N. Main; *for mail,* 68 Laurel St., Carbondale, Pa.
Hillgartner, Harry Leonard (J'45) (JSC), c/o Wm. Page, 110 Grant Ave., Hamilton, Ont., Can.
Hillier, Ronald Grundy ('44) (CJMS), W. D. Beath & Son Ltd., 394 Symington Ave.; *for mail,* 199 Kingswood Rd., Toronto, Ont., Can.
Hillman, Fred Worsdale ('44) (FSZKE), 212 Harvard Pl., Ontario, Calif.
Hills, Frederic Wheeler ('35) (COMDJL), Consultant in Mgmt., Rm. 330, 551—5th Ave., New York 17, N.Y.; *for mail,* 74—4th Ave., East Orange, N.J.
Hills, Leslie W. ('21; '37) (BMCNAK), V.P., Hills Bros. Coffee, Inc., 2 Harrison St., San Francisco 19, Calif.
Hills, Robert LaRue (J'43), 76 Pleasant St., Bolivar, N.Y.
Hillsley, John Jacob, Jr. (J'42), 1st Lt., Ord. Dept., Ord. Sch., Aberdeen Proving Ground, Md.
Hilmer, Otto E. ('16; '18) (SLFEOD), Partner, Fosdick & Hilmer, Union Trust Bldg., Cincinnati 2; *for mail,* 436 Westcliff Lane, Cincinnati 20, Ohio.
Hilpert, Meier George ('26) (BRW), Lt. Comdr., U.S.N.R., Retired; *for mail,* 33 W. Church St., Bethlehem, Pa.
Hilprecht, Robert Charles, Jr. (J'40) (BKFEAS), Apt. 37G, Mars Estates, 1628 Rickenbacker Rd., Baltimore 21, Md.
Hilstrom, Hollis Robert (J'43) (MCLJEO), Pres., Driv-Lok Pin Co., 565 W. Washington Blvd., Chicago 6; *for mail,* 3902 Central Ave., Western Springs, Ill.
Himes, Walter Hugh ('37) (MN), Prod. Engr., Elec. Boat Co., Groton; *for mail,* Box 503, Noank, Conn.
Hinch, Ralph J. ('39) (CFS), Asst. Supt. Constr., Commonwealth Edison Co., 72 W. Adams St., Chicago, Ill.
Hinckle, Don Albert (J'45) (BFK), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1; *for mail,* 1706 Wyandotte Rd., Columbus 8, Ohio.
Hind, John Henry, Jr. (J'38) (SOE), 1525 Sanborn Ave., Los Angeles 27, Calif.
Hind, Thomas Wilkinson ('40) (SMFKL), Expeditor & Inspr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 120 Vermilyea Ave., New York 34, N.Y.
Hindle, Norman Frederick ('44) (JNCBKD), Tech. Dir., American Foundryman's Association, Inc., 222 W. Adams St., Chicago 6; *for mail,* 5 W. Warren St., Calumet City, Ill.
Hindman, W. L. ('14; '35) (BCM), Tool Engr., Chrysler Corp., Detroit; *for mail,* 16159 Lilac Ave., Detroit 21, Mich.
Hindman, William P. ('44) (CMR), Works Mgr., Am. Car & Foundry Co., Milton, Pa.
Hine, Russell Curtis ('18; '35) (SFEK), Asst. to Supt. Power Prod., Consltd. Gas, Elec. Light & Power Co., Lexington Bldg., Baltimore 1; *for mail,* 307 Underwood Court, Baltimore 12, Md.
Hineline, Burton Clark (J'43) (KLS), 10 Terrace Pk., Rochester, N.Y.
Hines, George Ernst ('41) (EFHKOS), Partner, Russell B. Moore Co., 1456 N. Delaware St., Indianapolis; *for mail,* Country Club Pk., Lebanon, Ind.
Hines, Vernon A. (J'41) (FKS), 751 Ardmore Ave., Akron 2, Ohio.
Hinkle, James Oliver (J'42) (CJM), Lt. (j.g.), U.S.N.R.; 761 N. 12th St., Reading, Pa.
Hinkle, Rolland Theodore (J'36) (NB), Asst. Prof., Mch. Design Dept., Cornell Univ., Ithaca, N.Y.
Hinkle, William Paul (J'44) (OMBKNJ), Naval Arch., Naval Shipyard, Charleston, S.C.; *for mail,* Box 305, Route 1, Thomasville, N.C.
Hinkley, Ray Alfred ('44) (BDKN), Mech. Engr., Corning Glass Works, Corning, N.Y.
Hinnant, Clarence Henry, Jr. ('40; '45) (KTC), Maj., Corps of Engrs., Post Engr., Army Serv. Forces Depot, Richmond 12; *for mail,* 1023 W. 45th St., Richmond 24, Va.
Hinton,, Charles Louis, Jr. ('45) (EHNSLK), Mech. Engr., Head of Pump Gp., M. W. Kellogg Co., 225 Broadway, New York; *for mail,* 17 McIntyre St., Bronxville, N.Y.
Hinton, William Arthur, ('41) (BKS), Asst. Prof. Mech. Engrg., College of Engrg., Duke Univ., Box 261, College Sta., Durham, N.C.
Hires, John Edgar ('22; '26) (MCO), V.P., Charles E. Hires Co., 206 S. 24th St., Philadelphia 3, Pa.
Hirsch, Bertram H. ('35; '41) (CM), Engr., Geo. S. May Co., 235 Montgomery St., San Francisco 4; *for mail,* Box 957, Route 2, Alto, Mill Valley, Calif.
Hirsch, Charles Edward ('23; '39) (LSCDNK), Dist. Engr., Great West. Sugar Co., Box 5308, Terminal Annex, Denver 17, Colo.
Hirsch, Curt Walter (J'41) (NMCHBD), Electronic Technician's Mate, 2nd Class, U.S.N., Separation 3, Navy Separation Center, Shoemaker, Calif.; *for mail,* 76-33—174th St., Flushing, L.I., N.Y.
Hirsch, Edward Felix, Jr. (J'45) (COS), Ensign, U.S.N.R., U.S.S. *Baynton,* DE-1, F.P.O., New York, N.Y.
Hirsch, Fred (J'41) (BKS), 1264 Rhinelander Ave., New York, N.Y.
Hirsch, Gustav ('13) (BCOS), Cons. Engr., 810 W. Broad St., Columbus 8, Ohio.
Hirsch, Joseph (J'37) (CBJ), Ord. Engrg. Sec., Research & Devel. Dept., Naval Ord. Test Sta., Inyokern; *for mail,* Box 284, Naval Ord. Test Sta., Inyokern, Mojave, Calif.
Hirsch, Sylvan Roos ('28; '45) (MKNCLZ), Ch. Engr., Brunner Mfg. Co., 1821 Broad St., Utica 1; *for mail,* 2626 Sunset Ave., Utica 4, N.Y.
Hirschberg, Charles Austin (A'17) (WHEBCJ), Pres., Howard H. Samuel, Inc., 200 Valentine St., Hackettstown, N.J.
Hirschland, Franz Herbert ('06; '16) (CJLR), Chmn., Bd. of Dirs., Metal & Thermit Corp., Box 255, Rahway, N.J.
Hirshfeld, Egon Henry (J'44) (BNMK), Mech. Engr., Fish-Schurman Corp., 230 E. 45th St., New York 17; *for mail,* 156-08 Riverside Dr. W., New York 32, N.Y.
Hirshfeld, James Floyd (J'37) (CMJN), Asst. Gen. Mgr., Stran-Steel Div., Great Lakes Steel Corp., 3750 Penobscot Bldg., Detroit 26, Mich.
Hirt, John Harrison (J'43) (KJFZOH), Sales Engr., James H. Knapp Co., 4920 Loma Vista Ave., Los Angeles 11; *for mail,* 718 De La Fuente, Monterey Park, Calif.
Hitch, Robert Arthur (J'39) (ABKCEF), Engr., Aircraft Gas Turbine Research, De Laval Steam Turbine Co., Trenton 2, N.J.; *for mail,* 3000 Van Buren St., Wilmington, Del.
Hitchcock, E. A. ('98), Dean Emeritus, College of Engrg., Ohio State Univ.; *for mail,* 348 W. 8th Ave., Columbus, Ohio.
Hitchcock, John H. ('31; '37) (JBN), Dir. of Research, Morgan Constr. Co., 15 Belmont St., Worcester 8, Mass.
Hitchman, William Raymond (J'43) (FKS), 4002 Stuanton Ave., Charleston 4, W. Va.
Hite, Merrill W. (J'37) (JMW), Draftsman, Washington Iron Works, 1500—6th Ave. S.; *for mail,* 17860 Midvale Ave., Seattle, Wash.
Hitz, Robert E. (J'39), 1182 Warren Rd., Lakewood 7, Ohio.
Hixon, Chas. R. ('21) (FJS), Head Prof. Mech. Engrg., Ala. Poly. Inst., 111 Mitcham Ave.; *for mail,* Box 329, Auburn, Ala.
Hjerpe, Carl W. (J'39), Factory Rep., H. A. Thrush & Co., 38 Cabot St., Newton, Mass.; *for mail,* 149 Roxbury Rd., New Britain, Conn.
Hjerpe, Norman F. ('32; '41) (DFS), 149 Roxbury Rd., New Britain, Conn.
Hoag, Wilton Folsom ('30) (BCJMN), Owner, Wilton F. Hoag & Co., 13 N. River St., Batavia, Ill.
Hoagland, Cecil Normand ('29; '36) (CMNDBJ), Prod. Engr., Bullard Co., Canfield Ave., Bridgeport; *for mail,* 119 Sterling Pl., Bridgeport 4, Conn.
Hoagland, Frank Oscar ('12) (BHJN), *Vice-President,* '38; M.M., Pratt & Whitney Div., Niles-Bement-Pond Co., Charter Oak Blvd., West Hartford 1, Conn.
Hoagland, Jack Charles (J'42) (ACFHJK), Ensign, U.S.N., Tech. Asst. to Officer-in-Charge, Bur. of Ord., Navy Dept., Johns Hopkins Univ., Silver Spring, Md.; *for mail,* 1228 N. Sierra Bonita, Pasadena 7, Calif.
Hoagland, John E. ('30; '35) (BCM), Foreman, Gillette Razor Co., Boston; *for mail,* 88 Kilburn Rd., Belmont, Mass.
Hoar, John C. ('14; '20) (EFOS), Asst. Ch. Mech. Engr., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19; *for mail,* 376 Vernon St., New Kensington, Pa.
Hoban, Charles Owen ('44) (SKFE), Engr., Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3; *for mail,* 299 Vernon Ave., Glencoe, Ill.
Hobart, Frank G. ('90; F'41) (E), Cons. Engr., Fairbanks, Morse & Co.; *for mail,* 732 Hobart Pl., Beloit, Wis.
Hobart, Henry M. ('15) (ERS), Cons. Engr., 10 Balltown Rd., Schenectady, N.Y.
Hobbs, Elmer E. ('20; '35), Foreman, Power Plant, U.S. Naval Acad., Annapolis, Md.
Hobbs, Franklin W. ('11) (TBCEFZ), Pres., Arlington Mills, 78 Chauncey St., Boston, Mass.
Hobbs, George Stevenson (J'45) (SMNJHB), Lt. (j.g.), U.S.N.R., Ch. Engr., LST-755, F.P.O., San Francisco, Calif.
Hobbs, James Clarence ('12; '16; '20; F'41) (SHLABC), 60 Wood St., Painesville, Ohio.
Hobbs, William Stevenson (J'28) (KHSF), Mech. Engr., Engrg. Div., Mfg. Dept., Sun Oil Co., 1608 Walnut St., Philadelphia 3; *for mail,* Box 269, Swarthmore, Pa.
Hoblit, Frederic Maltby (J'39) (ABES), Stress Analyst, Lockheed Aircraft Corp., Burbank; *for mail,* 817 S. Madison Ave., Pasadena 5, Calif.
Hoblitzell, William Henry, Jr. (J'45), 3963 Abington Ave., Cincinnati 29, Ohio.
Hobson, Robert Raymond (J'41) (AYK), 525½ Smithson Ave., Lawrence Park, Erie, Pa.
Hoch, Andrew Wilbur (J'45), 40-16—168th St., Flushing, N.Y.

Hoch, Fred W. ('30; '35) (BCG), Pres., Mgr., Fred W. Hoch Associates, Inc., 461—8th Ave., New York, N.Y.
Hoch, Thornley Searles (J'43) (CMN), Lt., U.S.N.R., Mchy. Inspr., Safety Officer, c/o Supvr. of Shipbldg., U.S.N., Consltd. Steel Corp., Orange, Tex.
Hochman, Eugene (J'38) (ELWYZS), Mar. Engr., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail*, Apt. 208, 3912 Elbert Ave., Alexandria, Va.
Hochmuth, Frank William (J'41) (SFKLEW), Engr., Paper Mill Equip., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Hochner, Royal Monroe (J'42) (YMTCR), Firestone Tire & Rubber Co., Akron; *for mail*, 1060 W. Jefferson St., Akron 2, Ohio.
Hochuli, John H. ('28; '35) (SFLDEJ), Mech. Engr., Gibbs & Hill, Inc., Pa. Sta., New York 1; *for mail*, 8535—90th St., Woodhaven 21, L.I., N.Y.
Hockema, Frank C. ('22; '27) (YSE), Exec. Dean, Secy. to Bd. of Trustees, Purdue Univ., Lafayette; *for mail*, 832 Main St., West Lafayette, Ind.
Hodara, Ralph Leon (J'45) (TCYSML), Jr. Engr., Barnes Textile Associates, 10 High St., Boston 10; *for mail*, 3 Ames St., Cambridge, Mass.
Hodes, Lewis E. (J'40) (HBESJK), Lt. (j.g.), U.S.N.R., Bachelor Officers' Quarters, Lewis Park, Naval Oper. Base, Norfolk, Va.
Hodge, Byron T. ('37), Natl. Carbon Co. Inc., Fostoria; *for mail*, Route 1, Alvada, Ohio.
Hodge, Chas. A. ('18) (EJS), Instr., City of Yonkers, 33 Seminary Ave., Yonkers, N.Y.
Hodge, Gordon N. (J'41) (CHMDS), Devel. Engr., Worthington Pump & Mchy. Corp., Harrison, N.J.
Hodge, John Frederick ('45) (CLMEF), Plant Engr., Celanese Corp. of Am., Box 148, Bishop; *for mail*, Box 804, Kingsville, Tex.
Hodges, John Little ('24; '26; '31) (SFE), Deputy Smoke Abatement Engr., Dept. of Smoke Regulation, Hudson County Court House; *for mail*, 2787 Hudson Blvd., Jersey City 6, N.J.
Hodges, Kenneth Raymond ('35; '43) (CFE DM), Asst. Editor, *Industry & Power*, Maujer Publ. Co., 420 Main St., St. Joseph, Mich.
Hodgkinson, Francis ('02; F'39) (EKS), *Vice-President*, '39-'41; *Holley Medallist*, '38; Cons. Engr., 138 E. 36th St., New York, N.Y.
Hodgkinson, George Arthur (J'27) (FSKE), Fuel Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Hodgson, Alec Wilberforce ('10) (SBCE), Retired; 92 Mountain Ave., Summit, N.J.
Hodgson, Grant Baker (J'40) (CEAKHN), Staff Engr., Research & Product Devel., Solar Aircraft Co., 2200 Pacific Highway, San Diego 12; *for mail*, 3529 Ingraham, San Diego 9, Calif.
Hodgson, Ronald Hugh Cleveland (J'41) (CAE), Apt. 2, 1514 MacKay St., Montreal 25, Que., Can.
Hodson, Walter David ('29) (JL), Pres., Gen. Mgr., Hodson Corp., 5301 W. 66th St., Chicago 38, Ill.
Hodson, Walter Henry (A'43), V.P., Gen. Sales Mgr., Hodson Corp., 5301 W. 66th St., Chicago 38, Ill.
Hoefer, Elmer George ('09; '30) (NSEFBK), Prof. Mech. Engrg., N.C. State College; *for mail*, F-3 Raleigh Apt., Raleigh, N.C.
Hoelke, Howard Alex (J'42), Lt. (j.g.), U.S.N.R., 4908 Westway Dr., Washington 16, D.C.
Hoell, George S. ('39) (BHN), Dir., Mech. & Civ. Engrg. Div., Franklin Inst., 20th St. & Parkway, Philadelphia 3; *for mail*, 449 W. Bringhurst St., Philadelphia 44, Pa.
Hoelscher, Randolph Philip ('45) (GNM), Prof. Gen. Engrg. Drawing, 307 Trans. Bldg., Univ. of Ill.; *for mail*, 1107 S. Busey Ave., Urbana, Ill.
Hoernes, Helmut H. ('31; '35) (NBE), Designing Engr., W. E. Hall Ptg. Co., 4600 Diversey, Chicago; *for mail*, 6230 Dorchester Ave., Chicago 37, Ill.
Hoerter, Harry Eugene (J'42) (AEW), c/o K. V. Knapp, Box 81, Ft. Myers, Fla.
Hoevel, K. O. (J'40), National Supply Co., Tulsa, Okla.
Hoey, Clyde R., Jr. ('25; '35; '35) (SKZF WH), Steam Specialist, Champion Paper & Fibre Co., Main St.; *for mail*, Box 826, Canton, N.C.
Hoff, Nicholas John ('44) (BAYW), Assoc. Prof. Aero. Engrg., Poly. Inst. of Brooklyn, 85 Livingston St.; *for mail*, 131 Joralemon St., Brooklyn 2, N.Y.
Hoffberg, Howard J. (J'40) (BHK), 98 Hancock St., Cambridge 39, Mass.
Hoffer, Howard Anthony ('18; '24; '27) (CMH JDB), Asst. Gen. Sales Mgr., U.S. Pipe & Fdy. Co., Burlington, N.J.
Hoffman, Albert A. ('18), 42 The Uplands, Berkeley 5, Calif.
Hoffman, Carl (J'42), 87 Martin Ave., Hempstead, L.I., N.Y.
Hoffman, Carl (J'42) (NMJ), Tool Designer, Grumman Aircraft Engrg. Corp., Bethpage, L.I.; *for mail*, 2657 Bedford Ave., Brooklyn 10, N.Y.
Hoffman, David (J'31) (CDM), Plant Layout Engr., Westinghouse Elec. Corp., Lester; *for mail*, 5060 City Ave., Philadelphia 31, Pa.
Hoffman, E. Edward ('38) (BCJLMO), Dir., Indus. Finishes Div., Natl. Paint, Varnish & Lacquer Assn., 1500 Rhode Island Ave., N.W., Washington 5, D.C.
Hoffman, Edward Lee (J'44) (AEH), Apt. 4, 2404 Chesapeake Blvd., Hampton, Va.
Hoffman, George A. ('30) (FS), Gen. Supt., Pumping Stas., St. Louis Water Div., 34 E. Grand Blvd., St. Louis 7, Mo.
Hoffman, George Anthony (J'45) (BN), 3342 Dellwood Rd., Cleveland Heights 18, Ohio.
Hoffman, Harry Burnett (J'44), Lt. (j.g.), U.S.N.R., Senior Officers' Quarters, U.S. Naval Hospital, Mare Island, Calif.
Hoffman, Howard T. ('26; '35), 1331 Chardon Rd., Euclid, Ohio.
Hoffman, Irwin R. ('45) (FCOSZR), Fuel Engr., Sales Co-ordinator, Cent. Coal Co., Inc., 215 E. 149th St., New York; *for mail*, 14 St. George Rd., Great Neck, L.I., N.Y.
Hoffman, J. Roy ('16; '26) (EJY), V.P., Treas., Smith-Booth-Usher Co., 2001 Santa Fe Ave., Los Angeles 54, Calif.
Hoffman, John Ernest ('15; '22; '35) (CLDS FO), Dir. of Purchases, Cuban Am. Sugar Co., 120 Wall St., New York 5, N.Y.
Hoffman, Paul C. (J'39), College of Engrg., Univ. of Detroit, Detroit, Mich.
Hoffman, Ralph N. ('21; '35), Morenci, Ariz.
Hoffman, Robt. J. ('20), V.P., Prest-O-Lite Co., Inc., 30 E. 42nd St., New York, N.Y.
Hoffman, Russell Richard (J'42) (SKJ), Foreman, Power & Light, Interlake Iron Corp., 108th St. & Calumet River; *for mail*, 10959 Ave. B, Chicago 17, Ill.
Hoffman, Wray M. (J'41) (CAJME), Liaison Quality Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail*, 231 Haverford Ave., Swarthmore, Pa.
Hoffmann, Alexander Francis ('45) (BDFKOS), Erection & Oper. Engr., Internatl. Combustion Ltd., 19 Woburn Pl., London, W.C. 1, England.
Hoffmann, Emil H. (J'37) (HBSEJK), Elec. Engr., Ambursen Engrg. Corp., 800 Hermann Bldg., 204 Travis St.; *for mail*, 720 Bomar Ave., Houston, Tex.
Hoffmann, James M. (J'39) (BYDSM), 1st Lt., Corps of Engrs., U.S.A., Co. B, 115th Engrs. Bn. (C), A.P.O. 40, c/o Postmaster, San Francisco, Calif.; *for mail*, 4101 N. Kilbourn Ave., Chicago 41, Ill.
Hofft, M. A. (A'37) (DSW), Pres., M. A. Hofft Co., 441 W. Georgia St., Indianapolis, Ind.
Hofmann, Charles Sebastian (J'44) (BJNZ), Research Asst., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Hofmann, Edward Henry (J'42) (SNACEH), Designer, Bell Tel. Labs., 463 West St., New York 14; *for mail*, 112 Hale Ave., Brooklyn 8, N. Y.
Hofmann, Gideon (J'42) (BNZMJ), Research Engr., U.S. Time Corp., Waterbury 91, Conn.
Hofmann, Robert Thomas (J'45), 2226 Kenilworth Ave., Berwyn, Ill.
Hofstein, Lawrence L. (J'39) (BHSLNE), 3381 Sedgewick Ave., Bronx, N.Y.
Hofstetter, Elias T. Cruse ('27; '30; '35) (BCD EFJ), Asst. to Supt., Consltd. Gas Elec. Light & Power Co., Lexington Bldg., Baltimore 3; *for mail*, 1208 Round Hill Rd., Baltimore 18, Md.
Hogan, Curt Leland ('42) (MZBS), Retired; 1218 Sherman St., Alameda, Calif.
Hogan, John Joseph ('46), V.P., Gen. Mgr., Brunner Mch. & Tool Co., Inc., 214 William St., New York 7; *for mail*, 27 Cambridge Ave., White Plains, N.Y.
Hogan, John Philip ('26) (ORS), Partner, Parsons, Brinckerhoff, Hogan & MacDonald, Engrs., 142 Maiden Lane, New York, N.Y.
Hogan, Mervin B. ('30; '43) (BJK), Prof. Mech. Engrg., Univ. of Utah, Salt Lake City 1; *for mail*, 1166—2nd Ave., Salt Lake City 3, Utah.
Hogan, William Edward ('43) (ACM), Head Procurement Inspr., Army Air Forces, Matériel Comd., Sikorsky Aircraft Corp., South Ave., Bridgeport; *for mail*, 215 Salem St., Bridgeport 6, Conn.
Hoge, Frederick H. ('21) (BCDN), Pres., W. A. Jones Fdy. & Mch. Co., 4401 Roosevelt Rd., Chicago 24, Ill.
Hoge, Wallace W. ('26; A'34) (CRAY), Buying Dept., Morgan Stanley & Co., 2 Wall St., New York 5, N.Y.; *for mail*, 50 Bedford Rd., Summit, N.J.
Hogg, Allan Douglas (J'40) (BZSK), Asst. Research Engr., Hydro-Elec. Power Comm. of Ont., 620 University Ave., Toronto 2, Ont., Can.
Hogg, James Vanderbilt (J'40) (MSONCE), 32 Grant Ave., Amsterdam, N.Y.
Hogg, John Webb ('18; '26) (SHF), E. I. du Pont de Nemours & Co., 3500 Grays Ferry Ave., Philadelphia 46; *for mail*, Sproul Rd., Broomall, Pa.
Hogin, Philip Edward (J'43) (MCAWBR), Ensign, U.S.N.R., 36 Axtell Dr., Scarsdale, N.Y.
Hoglund, Nils Ove (J'44) (BJM), 210-07 Richland Ave., Flushing, N.Y.
Hohenstein, Frederick Carl (J'44) (KCEFSO), Mech. Engr., Heat Transmission Sales Dept., Elliott Co., Jeannette, Pa.
Hohl, Leonard Louis ('31) (HBYKDS), Engr., Am. Rubber Mfg. Co., 1145 Park Ave., Oakland 8; *for mail*, 30 Beverly Rd., Berkeley 8, Calif.
Hoirils, Holger ('43) (BML), Asst. Plant Engr., Moore Dry Dock Co., Foot of Adeline St., Oakland; *for mail*, 851 Regal Rd., Berkeley 8, Calif.
Hoke, Arnold ('13) (CDF), V.P., Whitehall Cement Mfg. Co., 160 Main St., Cementon, Pa.
Holbreich, Milton (J'41) (CDM), Lt., Signal Corps, U.S.A., 552nd Sig. Depot Co., Pusan Det., A.P.O. 901, San Francisco, Calif.; *for mail*, 275 Pennsylvania Ave., Freeport, L.I., N.Y.
Holbrook, Dio Lewis ('99) (H), Retired; 305 Lincoln Ave., Point Pleasant, N.J.
Holbrook, Frank Martyn ('44) (LKS), Mech. Engr., Alpha Steam Specialty Co., 15 Park Row, New York 7, N.Y.
Holbrook, Gordon E. (J'39) (ASE), Sr. Project Engr., Allison Div., Gen. Motors Corp., Indianapolis 6, Ind.
Holcomb, Normand P. (J'40) (GJM), Tool Designer, Colt's Pat. Firearms Mfg. Co., Inc., 17 VanDyke Ave., Hartford; *for mail*, Spring St., Warehouse Point, Conn.
Holcombe, Amasa Maynard (A'16) (BJM), Pat. Lawyer, 438 Munsey Bldg., Washington 4; *for mail*, 3024 Tilden St., N.W., Washington 8, D.C.
Holdcraft, Harvey J., Jr. (J'41) (ABP), Lt., U.S.N.R., Ground Sch., U.S. Naval Air Sta., Pensacola, Fla.
Holdcraft, Joseph William (J'43) (KSA), Lt. (j.g.), S-E(4), U.S.N.R., Naval Shipyard, Boston, Mass.; *for mail*, 1247 Langham Ave., Camden, N.J.
Holden, Albert Frederick (J'38) (SZBR), Asst. Engr., Elec. Operas., Pacific Gas & Elec. Co., 245 Market St., San Francisco 6, Calif.
Holden, Edward Allen ('18; '25; '29) (FKES MZ), Ch. Engr., Engr. Co., 75 West St., New York 6, N.Y.
Holden, Frederick Raymond (J'42) (ABE), 6312 Morton St., Philadelphia 44, Pa.
Holden, Irving Harold ('43) (NMCOJB), Mech. Design Engr., Philharmonic Radio Corp., 528 E. 72nd St., New York 21; *for mail*, 1440 East Ave., New York 62, N.Y.
Holden, John M. (J'40) (CM), Prod. Engr., Security Engrg. Co., Inc., 108 W. Whittier Blvd.; *for mail*, 517 S. Comstock St., Whittier, Calif.
Holden, Paul E. ('26) (CMBDL), Prof. Indus. Mgmt., Graduate Sch. of Business, Stanford Univ., Stanford University, Calif.
Holding, John Brewster ('33; '45) (JCBL), Research Engr., Battelle Memorial Institute, 505 King Ave., Columbus 1, Ohio.
Holdredge, Edwin (J'39) (BKS), 488 Robertsville Rd., Oak Ridge, Tenn.
Holdredge, Ernest Cardwell (J'42) (SEFKNR), Instr. in Mech. Engrg., Univ. of Tenn., Knoxville 16; *for mail*, Crestwood Rd., Fountain City, Knoxville 18, Tenn.
Holecek, John Jerry (J'39) (OBCM), Plumbing, Htg., Contrg., J. J. Holecek Co., 42 W. 65th St., New York, N.Y.
Holford, Harry Earl ('22; '35), Pres., Secy., Binghamton Fdy. & Mch. Co., Inc., N. Floral Ave.; *for mail*, 8 Cedar St., Binghamton, N.Y.
Holgate, Fred B. ('40) (BCK), Mech. Engr., Matl. Lab., N.Y. Naval Shipyard, Brooklyn; *for mail*, Cedar Shore Dr., Massapequa, N.Y.
Holland, A. Dinsmore ('34; '35) (SFKH), Prof. Mech. Engrg., Ga. Sch. of Tech., 221 North Ave.; *for mail*, 866 Greenwood Ave., N.E., Atlanta, Ga.
Holland, C. Kenneth (J'36) (ESH), Asst. Mech. Engr., Ebasco International Corp., 2 Rector St., New York 6, N.Y.; *for mail*, 15 Vista Ave., Elizabeth 3, N.J.
Holland, Cyrus Jonathan ('39; '41) (RBC), Pres., Holland Co., 332 S. Michigan Ave., Chicago 4, Ill.
Holland, Raymond (J'44) (CGM), Supt., Pollock Paper & Box Co., 4202 Jefferson St.; *for mail*, 1518 Pearson, Houston 3, Tex.
Holland, Ubert C. ('24; '35) (AGM), Prof., College of Engrg., Rutgers Univ., New Brunswick, N.J.
Hollander, Aladar ('19) (HBSKAP), Assoc. Prof., Calif. Inst. of Tech., 1201 E. California St., Pasadena 5; *for mail*, 2385 Hill Dr., Los Angeles 41, Calif.
Hollander, Emanuel ('01; '06), Cons. Engr., 153 Hillside Ave., Mt. Vernon, N.Y.
Holle, Frederick D. (J'40) (AJMNWY), Design Dept., Austin-West. Co., Aurora, Ill.
Hollensbe, H. E. ('38) (SEFCDZ), Editor, *Industry & Power*, Maujer Publ. Co., 420 Main St., St. Joseph, Mich.
Holler, Harold George (J'43) (ESFKA), Ensign, U.S.N.R., 307 E. Wilbur Ave., Milwaukee 7, Wis.
Holleran, Dominic John ('43), Plant Engr., Natl. Lock Washer Co., 40 Hermon St., Newark; *for mail*, 151 Headley Terrace, Irvington, N.J.
Hollerith, Chas. ('39) (A), V.P., Charge Engrg., Hayes Industries, Inc., Wildwood & Fern Ave., Jackson, Mich.
Hollerith, Herman, Jr. ('19; '28) (A), Matls. Engr., Glenn L. Martin Co., Middle River, Baltimore 3; *for mail*, 5 Englewood Rd., Baltimore 10, Md.
Hollerith, Richard (J'25), 307 Shrewsbury Rd., Riverton, N.J.
Holley, James J. (J'41) (EFS), 807 Pemart Ave., Peekskill, N.Y.
Holliday, Frank Richard (J'43) (ABCR), Chrysler Corp., 12800 Oakland Ave., Detroit; *for mail*, 56 Colorado, Detroit 3, Mich.
Hollingsworth, Samuel ('99; '07), Retired; Scotch Plains, N.J.
Hollins, George G. ('21; F'44) (SFEDKL), Ch. Mech. Engr., J. G. White Engrg. Corp., 80 Broad St., New York 4, N.Y.

Hollis, Earl A. (J'24) (CJM), Buyer, Bell Tel. Labs., 463 West St., New York; *for mail,* 3448—88th St., Jackson Heights, L.I., N.Y.
Hollis, James Walter, Jr. (J'38), 2nd Lt., 18th Weather Sqd., Det. 113, A.P.O. 633, c/o Postmaster, New York, N.Y.
Hollis, Oliver Nelson ('20) (FS), Engr., Power Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston, Mass.
Hollis, R. Frank ('35; '35) (DKS), Mech. Engr., Alton Box Board Co.; *for mail,* 2510 Bloomfield St., Alton, Ill.
Hollister, Ezra (J'45) (HABMZ), Hyd. Devel. Engr., Pacific Div., Bendix Aviation Corp., 11600 Sherman Way, North Hollywood, Calif.
Hollister, James F. (J'46) (AHM), Automatic Pilot Lab., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 133 Harcourt Ave., Bergenfield, N.J.
Hollister, Solomon Cady ('34) (B), Dean, College of Engrg., Cornell Univ., Ithaca, N.Y.
Holloway, Frank M. (J'40) (FJM), Partner, Charge of Sales & Design, Holloway Engrg. Co., 1119 E. 4th St., Austin, Tex.
Holloway, Kenneth J. (J'41) (FAESMC), Design Engr., Stand. Oil Co., Sugar Creek, Mo.
Holloway, Robert Allen (J'44) (OCDMNL), Lt. (j.g.), U.S.N., Engrg. Officer, U.S.S. *Cymophane,* 496 Montauk Ave., New London, Conn.; *for mail,* 605 Los Altos Ave., Long Beach 4, Calif.
Hollowell, Glenn Alonzo ('42) (SEFH), Prod. Supt., W. Tex. Utilities Co., Abilene, Tex.
Hollowell, James S. ('30) (DEL), V.P., Charge Engrg., Natural Products Refining Co., 902 Garfield Ave., Jersey City 5, N.J.
Hollweg, Charles Henry (J'39) (CMZJ), Lt. (j.g.), S (O), U.S.N.R., Prod. Officer, Naval Ord. Plant, Macon, Ga.
Holm, Sigurd Sverre ('24; '30; '35) (SKHJ), Engr., M. W. Kellogg Co., 225 Broadway, New York 7; *for mail,* 25 Park Circle, Great Neck, N.Y.
Holmberg, Charles G. ('28) (BJMY), Engr., Tool & Gage Design, West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 560 N. Chestnut St., Westfield, N.J.
Holmberg, Joseph Charles ('32; '35) (DBSHNC), Mech. Engr., Charge Conveyor Design, Swanson & Torsell, State Bank Bldg., *for mail,* 2615—14th Ave., Moline, Ill.
Holme, Thomas Timings (J'40) (CMZL), Assoc. Prof. Mech. Engrg., Lehigh Univ., Bethlehem, Pa.
Holmes, A. Clifford (J'42) (SKBENO), Asst. Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 815 S. Shea Ave., Milwaukee 4, Wis.
Holmes, Alester G., Jr. ('39) (EB), Head, Mech. Engrg. Dept., Miss. State College, State College, Miss.
Holmes, Clayton W. ('32; '33; '35) (SBHEKN), Assoc. Prof. Engrg., Haverford College, Haverford, Pa.
Holmes, George Robert ('22; '25; '35) (CJN), Pres., Treas., McLagon Fdy. Co., 100 Audubon St., New Haven 10, Conn.
Holmes, James Thomas ('26) (OAL), Partner, Holmes & Narver, 639 S. Spring St., Los Angeles 14; *for mail,* 64 Fremont Pl., Los Angeles 5, Calif.
Holmes, Joseph Addison ('32; '35) (SRLMCD), Asst. V.P., Natl. Aluminate Corp., 6216 W. 66th Pl., Chicago 38, Ill.
Holmes, Lester B. (J'39) (BHK), Lt., 1735 Riggs Place, N.W., Washington, D.C.
Holmes, Richard Beck ('39) (CDJ), Gen. Mgr., Link-Belt Co., 2045 W. Hunting Park Ave., Philadelphia 40; *for mail,* Henry Ave. & Coulter, Philadelphia 29, Pa.
Holmes, Robert Wallace (J'40) (CMZ), Prod. Scheduling Official, Ryerson Lab., Univ. of Chicago, 58th & Greenwood, Chicago; *for mail,* 6709 Cregier Ave., Chicago 49, Ill.
Holmes, William C. ('21; '27) (SFCO), Asst. Mgr., Sta. Constr. & Shops Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Holmquist, George Francis ('44) (MNJBDH), Mech. Engr., Stanley Works, New Britain; *for mail,* 674 Farmington Ave., Bristol, Conn.
Holms, Arthur Glatt (J'41) (BOCJAD), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland; *for mail,* 149 E. Center St., Berea, Ohio.
Holmstrom, Andrew Birger ('42) (CDMZYL), V.P. & Gen. Mgr., Abrasive Div., Norton Co., 1 New Bond St., Worcester 6, Mass.
Holowenko, Alfred Richard (J'45) (NBLSJA), Asst. Prof. Mech. Engrg., Purdue Univ., Lafayette, Ind.
Holpert, Merrill (J'45) (MJNBEO), 1661 E. 4th St., Brooklyn 30, N.Y.
Holsworth, Robert Charles ('42) (HKO), Owner, Holsworth Equip. Co., Box 1981, Corpus Christi, Tex.
Holt, Arthur D. ('42) (LKB), Devel. & Research Engr., Jeffrey Mfg. Co., 956-99 N. 4th St., Columbus 16; *for mail,* 1151 E. Broad St., Columbus 5, Ohio.
Holt, Charles Warren, Jr. (J'42) (AHJ), Ensign, U.S.N.R., LST-543, F.P.O., San Francisco, Calif.; *for mail,* 14 Central St., Hudson, N.H.
Holt, Clifford Curtis (J'37) (JME), Oil Equip. Engr., Butler Mfg. Co., 13th & Eastern Ave., Kansas City 3, Mo.
Holt, James ('33) (SKOF), Assoc. Prof., Mass. Inst. of Tech., Cambridge 39, Mass.
Holt, Kiate M. ('21) (CNS), Exec. Engr., Turbine Eng. Div., Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.
Holtane, Theodore K. (J'40) (BEM), Mech. Engr., Shop Foreman, Lago Oil & Transport Co., Aruba, N.W.I.
Holtby, Fulton (J'34) (JLMO), Asst. Prof. Mech. Engrg., 100 Mech. Engrg. Dept., Univ. of Minn., Minneapolis 14, Minn.
Holter, Alfred ('45) (CDL), Brevik, Norway.
Holton, Adolphus, Jr. (J'36) (LWDMGY), Mech. Designer, Raytheon Mfg. Co., Willow St., Waltham; *for mail,* 108 Cottage St., Norwood, Mass.
Holton, Paul H. ('39) (MJC), Field Engr., Carboloy Co., Inc. (of Detroit, Mich.), 613 Hardt Bldg., Philadelphia 22; *for mail,* 881 Wynnewood Rd., Philadelphia 31, Pa.
Holton, Philip J., Jr. ('29; '35) (CHS), Ch. Engr., Providence Water Supply Bd., 161 Fountain St., Providence 3, R.I.
Holtorp, Carl Heinrich ('42) (LMDKH), Mech. Engr., H. Kohnstamm & Co., 537 Columbia St., Brooklyn 31; *for mail,* 3025 Godwin Terrace, Bronx 63, N.Y.
Holtz, John Cromwell (J'34) (CFES), Explosive Engr., High Explosives Research Div., U.S. Bur. of Mines, 4800 Forbes St., Pittsburgh 13, Pa.
Holtzclaw, Henry J. ('33) (CDGMNY), Chief, Research & Devel. Engrg., Bur. of Engraving & Ptg., 14th & C Sts., S.W., Washington 25, D.C.
Holtzman, William P. ('37) (BJR), Lt. Col., U.S.A., 210 W. Patton St., Paxton, Ill.
Holtzworth, Howard Eckert (J'42), Barry St., Fairmont, W. Va.
Holway, William Rea ('22; '35) (HOS), Sr. Partner, W. R. Holway & Associates, 302 E. 18th St., Tulsa 5, Okla.
Holz, Peter Phillip (J'42) (OKDM), Asst. Fdy. Engr., Harrison-Corry Co., Box 175, Knoxville 1, Tenn.
Holzbaur, Fredrick John ('20; '25; '28) (CZKFA), M.M., Stand. Oil Co., Sugar Creek, Mo.
Holzer, Hiram Adsit ('21) (CJMNYF), Pres., Gen. Mgr., United Iron Works Co., 102 N. Locust, Pittsburg, Kan.
Homewood, William Thomas ('17; '21) (CDEFKS), Engrg. Dept., E. I. du Pont de Nemours & Co., Du Pont Bldg., Wilmington 98, Del.
Homsher, R. Lee (J'36) (BLM), 827 E. Madison St., Lancaster, Pa.
Honc, Harold John (J'45) (CDK), Elliott Co., Jeannette, Pa.; *for mail,* 2825 Grand Concourse, Bronx 58, N.Y.
Honecker, Norman Christian ('26; '46) (KCLMNA), Sales Engr., A. E. Borden Co., 142 High St., Boston 10; *for mail,* 159 Oliver Rd., Waban 68, Mass.
Honold, Robert Paul (J'39) (KSEFHL), 1656 E. Garfield, Seattle 2, Wash.
Honywill, Albert William ('11; '16; '23) (CMS), Treas., Diamond Power Specialty Corp., Box 288, Detroit 31, Mich.
Honza, Donald William (J'43) (CMDLBN), U.S.N.; *for mail,* 2919 N. 27th St., Kansas City 2, Kan.
Hood, Ben Benight ('30) (CFLMDJ), Engr., Reconstruction Finance Corp., 811 Vermont Ave., N.W., Washington 25, D.C.; *for mail,* Route 1, Falls Church, Va.
Hood, John M. (J'41) (CJD), Ensign, U.S.N., LCI (x) 761, F.P.O., San Francisco, Calif.; *for mail,* c/o B. B. Hood, Route 1, Falls Church, Va.
Hook, C. Howard ('22) (AJKN), 1414 Simona Dr., Pittsburgh 1, Pa.
Hook, George Randall (J'37) (ABEC), Sr. Aerodynamicist, Glenn L. Martin Co., Middle River, Baltimore Co.; *for mail,* 4000 Hillen Rd., Baltimore 18, Md.
Hook, Ira Thomas ('20) (JBCYK), Research Engr., Am. Brass Co., Ansonia; *for mail,* 494 Norton Pkwy., New Haven 11, Conn.
Hook, James William ('12; '14; '35) (CEFKS), Chmn. of Bd., United Illum. Co., 80 Temple St., New Haven 6, Conn.
Hooker, Howard Wolcott (J'42) (SKF), Sr. Engr., Philco Corp., Tioga & C St., Philadelphia 34; *for mail,* 5812 Green St., Philadelphia 44, Pa.
Hooker, Theodore F. ('24; '32; '35) (CL), Engr., Eastman Kodak Co., Kodak Park, Rochester 4, N.Y.
Hooper, Harry Allison (J'42) (ENMCZB), Engr., Design, United Auto Spring Co., 1612 Webster, New York 57; *for mail,* 76 Wendt Ave., Larchmount, N.Y.
Hooper, Irvin Platt (J'41) (EKS), Capt., U.S.M.C.R., Hdq. Co., 5th Serv. Bn., 5th Mar. Div., F.P.O., San Francisco, Calif.
Hooper, Leslie James ('29; '36) (HBASF), *Junior Award,* '37; Prof. Hyd. Engrg., Worcester Poly. Inst., 12 Boynton, Worcester 2; *for mail,* Highland Ave., Holden, Mass.
Hooper, Robt. P. (J'27) (FKS), Div. Engr., Tech. Serv. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York, N.Y.; *for mail,* 331 Church St., Boonton, N.J.
Hooper, William U. (J'39) (OSF), Engr., Mech. Designer, Egli & Gompf Inc., 425 St. Paul Pl., Baltimore 2; *for mail,* 6219 Lincoln Ave., Baltimore 9, Md.
Hoopes, Maurice ('01) (CSHW), Pres., Finch, Pruyn & Co., Inc., Box 350, Glens Falls, N.Y.
Hoopes, Penrose R. ('20; '26) (NMOBZL), Cons. Mech. Engr., 12 S. 12th St., Philadelphia 7; *for mail,* 435 W. Upsal St., Philadelphia 19, Pa.
Hoots, Paul Frost ('38) (SFCE), Asst. to Pres., New Orleans Pub. Serv., Inc., 317 Baronne St., New Orleans 12, La.
Hoover, Harmer Grant ('44) (ERS), V.P., Curtis Gas Eng. Corp., 2 Rector St., New York 6; *for mail,* 1645 W. 5th St., Brooklyn 23, N.Y.
Hoover, Hon. Herbert (H'25), Palo Alto, Calif.
Hoover, Ray Campbell, Jr. (J'43) (EMS), Seaman, 2nd Class, 97 Cedar St., Concord, N.C.
Hope, Herbert Ward, Jr. (J'45), Lt. (j.g.), Ch. Engr., U.S.N., 112 Atwater St., New Haven 13, Conn.
Hope, Robert DeVere ('30) (COFKTS), Raymond Commerce Bldg., Newark, N.J.
Hope, Walter Radcliffe ('15; '22) (BCDKLT), Retired; 700 W. 20th St., Wilmington 223, Del.
Hoper, Clarence H. ('39) (CASHKO), City Mgr., Alliance, Neb.
Hopewell, Geoffrey Handley ('44) (KSFCAP), Gen. Mgr., Foster Wheeler Ltd., Aldwych House, London W.C. 2, England.
Hopf, Harry Arthur ('18; '30) (C), Sr. Mem., H. A. Hopf & Co., Hopf Inst. Bldg., Ossining, N.Y.
Hopkins, Arthur G. ('33; '35), 5803—69th Lane, Maspeth, L.I., N.Y.
Hopkins, Charles Lee (J'45) (WTACSO), Asst. Application Engr., Westinghouse Elec. Corp., 9 Bowman St.; *for mail,* 56 Chapel St., Augusta, Me.
Hopkins, Hazen Ray (J'31) (CLDFK), Mech. Engr., Birdsong Storage Co., Suffolk, Va.
Hopkins, James David, Jr. (J'45) (COS), U.S.N., 4609 Langdrum Lane, Chevy Chase 15, Md.
Hopkins, John Ray (J'38) (AHJ), Inspr., Ord. Matl., War Dept., Philadelphia Ord. Dist., Mitten Bldg., Philadelphia, Pa.; *for mail,* 20 Salter Pl., Maplewood, N.J.
Hopkins, Kent Miles (J'45) Jr. Engr., A.C. Spark Plug Div., Gen. Motors Corp., Flint; *for mail,* 308 Y.M.C.A. 1, Flint 2, Mich.
Hopkins, Robert Kingsley ('25; '31) (JKHAEZ), Dir., Metal. Research & Devel., M. W. Kellogg Co., Danforth Ave., Jersey City 3, N.J.; *for mail,* 15 St. Austins Pl., West Brighton 10, S.I., N.Y.
Hopkins, William Edgar ('32; '35; '35) (SOFCBK), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 11 Willard St., Newton 58, Mass.
Hoppe, Alfred G. ('39) (BJR), Asst. Mech. Engr., Milwaukee Shops, Chicago, Milwaukee, St. Paul & Pacific R.R., Milwaukee, Wis.
Hoppe, Ewald, Jr. (J'43) (BLM), Tool Engr., Cent. Scien. Co., 1700 Irving Park Rd.; *for mail,* 3720 N. Paulina St., Chicago 13, Ill.
Hoppe, George Edward, Jr. ('40) (CKJ), Pres., Old Dominion Iron & Steel Corp., Belle Isle, Richmond 3, Va.
Hopper, Joseph Owen (J'44), Shop Supt., United Geophysical Co., 1016 Union Ave., Bakersfield, Calif.
Hopper, Philip S. (J'40) (ABJ), Asst. Project Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford; *for mail,* Box 222, Sta. A, 599 N. Main St., Manchester, Conn.
Hopper, Thomas Washburn (J'33) (SLODMC), Asst. Engrg. Mgr., Day & Zimmermann, Inc., 620 Packard Bldg., Philadelphia 2, Pa.
Hopping, Ernest LeRoy ('19) (SFHEJK), Mech. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Hopping, Russell L. (J'40) (BAE), Asst. to Design Engr., Glenn L. Martin Co., Middle River, Md.; *for mail,* 1520 N. Longfellow St., Arlington, Va.
Hopson, Wm. H. ('24; '35), Constr. Supt., W. T. Grant Co., 1441 Broadway, New York; *for mail,* 17 Hedgeway Court, Hempstead, L.I., N.Y.
Hopton, Walter Edwin ('91; '96) (FKLOST), Hopton Co., 321 Denison Bldg., Syracuse 2, N.Y.
Hopwood, John M. ('22), Pres., Hagan Corp., 304 Ross St.; *for mail,* Box 1346, Pittsburgh, Pa.
Hoque, Jack Emmet (J'43) (SNGE), Lt., Asst. Engrg. Officer, U.S.N.; *for mail,* 220 S. 12th St., San Jose 12, Calif.
Horan, Francis Timothy (J'45) (BEM), Asst. Research Engr., Elec. Boat Co., Groton, Conn.
Horan, John Joseph (J'39) (MCNJYA), Asst. Ch., Mech. Subdiv., Torpedo Div., Naval Ord. Lab., Washington 25; *for mail,* 2901 Erie St., S.E., Washington 20, D.C.
Horgan, William Stephen (J'38) (AFC), Test Engr., Tech. Operas. Group, Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford; *for mail,* 475 Adams St., Manchester, Conn.
Horger, Oscar John ('26; '37) (RBAJ), Ch. Engr., Ry. Div., Timken Roller Bearing Co., Canton 6, Ohio.
Hori, Tats (J'41) (NMAYOC), Mech. Engr., Internatl. Electronics Labs., 630—5th Ave., New York; *for mail,* 42-15 Judge St., Elmhurst, L.I., N.Y.
Horigan, John H. ('21; '35), Secy., Union Twist Drill Co.; *for mail,* 103 Cheney St., Athol, Mass.
Horlander, Leo Anthony, Jr. (J'45) (CSHKZL), Mech. Engr., Engrg. Dept., Brown-Forman Distillers Corp., 1908 Howard St., Louisville 10, Ky.
Horlebein, Edwin W. ('39) (J), Pres., Gibson & Kirk Co., Warner & Bayard St., Baltimore 30, Md.

Hormell, Duane C. ('22; '35) (SFKELO), Mech. Engr., Pub. Utility Engrg. & Serv. Corp., 231 S. La Salle St., Chicago 4, Ill.
Horn, Freeman ('31), Intelligence Officer, British Aluminium Co. Ltd., Salisbury House, London Wall, London E.C. 2, England.
Horn, Martin Richard (J'41) (CKLZ), Sales Engr., Gay Engrg. Corp., 2730 E. 11th St., Los Angeles 23; *for mail,* 4237 San Anseline Ave., Long Beach 8, Calif.
Horn, Norman E. ('16; '25) (CTAG), 15 Brayton Rd., Scarsdale, N.Y.
Horn, Robert Charles (J'42) (ABH), Test Engr., Consltd. Steel Shipbldg., Wilmington; *for mail,* 151 W. Forest Ave., Arcadia, Calif.
Horn, Robert J. ('22; '26; '35) (FESKCM), Gas Engrg. & Opera. (Gas), Cent. Hudson Gas & Elec. Corp., 50 Market St., Poughkeepsie, N.Y.
Horn, Walter (J'45) (BSHAKC), Asst. Devel. Test Engr., L. J. Wing Mfg. Co., 9 Victoria St., Newark 5; *for mail,* 24 Fairchfld Pl., Hillside, N.J.
Hornbaker, William Stanley (J'44) (BNKMHR), Mech. Engr., Clinton Engr. Works, Tenn. Eastman Corp., Box 590, Knoxville; *for mail,* Box 1253, Oak Ridge, Tenn.
Hornberger, Fred C. (J'36) (ZNHL), Designer, Moore Products Co., H & Lycoming, Philadelphia 24; *for mail,* 317 Sanford Rd., Upper Darby, Pa.
Hornberger, William Nick (J'43) (BMS), Steam Serv. Engr., Westinghouse Elec. Corp., Rm. 725, 1 Montgomery St., San Francisco; *for mail,* 6235 Buena Vista, Oakland 11, Calif.
Hornbostel, Lloyd ('42), 1638 Emerson St., Beloit, Wis.
Horne, Albert Nolan ('30) (KEP), Tex. Empire Pipe Line Co., Philtower Bldg., Tulsa, Okla.
Horne, Arthur Welch (J'45) (YOJDS), Mech. Project Engr., Bakelite Corp., River Rd., Bound Brook; *for mail,* 288 North Ave., Fanwood, N.J.
Horne, George Augustus ('19) (CK), Cons. Engr., Merchants Refrig. Co., 17 Varick St., New York, N.Y.
Horne, John ('25; '31; '35) (CDHJNO), Ch. Plant Engr., Aluminum Co. of Am., Alcoa; *for mail,* 1318 Young Ave., Maryville, Tenn.
Horne, Jos. A. ('13), Chmn. of Bd., Yale & Towne Mfg. Co., 405 Lexington Ave., New York, N.Y.
Horne, Lionel V. ('39) (P), Pressure Equip. Engr., Union Oil Co. of Calif., 617 W. 7th St., Los Angeles 14, Calif.
Horner, John S. (J'41) (CKW), Comdr., U.S.N.R., Com. Trans. Div. 101, F.P.O., San Francisco, Calif.
Hornidge, Richard Douglas (J'44) (AKFSC), Exper. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford, Conn.
Hornschuch, Hanns (J'30) (HBJ), Devel. Engr., Ingersoll-Rand Co., Phillipsburg, N.J.; *for mail,* 174 Pennsylvania Ave., Easton, Pa.
Horsley, Sherman (J'45) (DHM), Gen. Foreman, Maint., Oldsmobile Div., Gen. Motors Corp., 6813 E. 37th St., Kansas City; *for mail,* 3109 Sterling Ave., Independence, Mo.
Horst, Cecil Albert ('24; '33; '35) (CMODJ), Dist. Mgr., McCaskey Register Co., 265 Church St., New Haven 10; *for mail,* 675 Orange St., New Haven 11, Conn.
Horstkotte, E. H. ('37), Engr., Works Lab., Gen. Elec. Co., E. Lake Rd.; *for mail,* 1336 W. 9th St., Erie, Pa.
Horstmann, Frederick Bayliss ('30) (RS), Retired; 1029 W. 35th St., Chicago 9, Ill.
Horton, Albert J. ('22) (BGM), Pat. Atty., Cons. Engr., R. Hoe & Co., Inc., 910 E. 138th St., New York; *for mail,* 103 Grandview Ave., White Plains, N.Y.
Horton, Ralph D. (J'27), Maint. Bonded Fabric; *for mail,* Andover, N.J.
Horvath, Guy E. (J'38) (CDM), Asst. Supt., Packing Div., Charge Engrg., U.S. Asbestos Div., Raybestos-Manhattan, Inc., Manheim, Pa.
Horvath, Joseph Paul (J'39) (NHB), Designer, Vickers, Inc., 1400 Oakman Blvd., Detroit 6; *for mail,* 18295 Ferguson, Detroit 19, Mich.
Hosbein, Louis H. ('17; '22), 630 Washington Ave., Glencoe, Ill.
Hosford, Harry William, Jr. (J'44) (CEF), 2946 E. Overlook Rd., Cleveland Heights 18, Ohio.
Hosford, William F. (J'34) (KBRECS), Asst. to Supt. of Yards, Pullman Co., 220 S. State St., Chicago 4, Ill.
Hosford, William Fuller ('15), V.P., West. Elec. Co., Inc., 195 Broadway, New York 7, N.Y.
Hoshall, Harry Benton ('40) (GMJO), Asst. Prof. of Mech. Engrg., Univ. of Md., College Park, Md.
Hosking, James Richard (J'42), Installation Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford; *for mail,* 743 Prospect Ave., West Hartford 5, Conn.
Hoskins, Charles Howard (J'43) (BHS), Design Devel. Engr., Steam Turbine Div., De Laval Steam Turbine Co., 853 Nottinghamway, Trenton 2; *for mail,* 1501 S. Clinton Ave., Trenton 10, N.J.
Hoskins, Clement B. (J'42) (CML), Lt. (j.g.), U.S.N.R., Ord. Specialist, Bldg. 369, Norfolk Naval Shipyard, Portsmouth, Va.; *for mail,* 317 Bryn Mawr Ave., Cynwyd, Pa.
Hoskins, Richard Ireland, Jr. (J'44) (SKNE), Lt. (j.g.), U.S.N., Engrg. Officer, U.S.S. *Freestone,* F.P.O., San Francisco, Calif.; *for mail,* 356 N. 12th St., Baton Rouge 8, La.
Hosmer, Asa ('20; '35), So. Field Mgr., Factory Ins. Assn. of Hartford, Conn., 1218 Johnston Bldg., Charlotte, N.C.
Hosmer, Sidney ('02), Retired; Marion, Mass.
Hossack, Archibald Babcock ('14; '29) (CMO), Exec. V.P., Am. Appraisal Co., 1 Cedar St., New York 5, N.Y.
Hotchkiss, C. H. B. ('23; '26; '35), V.P., Ameresco, Inc., 50 Church St., New York, N.Y.; *for mail,* 241 Murray Ave., Ridgewood, N.J.
Hottel, H. C. ('35) (BFK), Prof., Dept. of Chem. Engrg., Mass. Inst. of Tech., Cambridge, Mass.
Hotze, Ernest George (J'43) (EC), Clark Bros. Co., Inc., 1212 Commerce Bldg., Houston 2, Tex.
Hou, Te-Pang ('22; '35) (JLYZ), Suite 1918, Universal Trading Corp., 630—5th Ave., New York 20, N.Y.
Houck, Frank W. (J'41) (FKS), Capt., Antiaircraft Arty. Sch., Dept. of Guns, Radar (M) Sec., Bldg. 1496, Camp Davis, N.C.; *for mail,* c/o W. J. Houck, R.F.D. 1, Irving, Ill.
Houghton, Carl Roy ('20) (BNZ), Ch. Engr., Roots-Connersville Blower Corp., 900 Mount St., Connersville, Ind.
Houghton, Clyde Arthur ('35; '35) (OS), Consultant, Reading P.O., South Woodstock, Vt.
Houghton, Harry S. ('18; '25) (YJLKC), Ch. Process Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Houghton, Horace Carter (J'29) (CM), Asst. to Mgr., Indus. Relations, Shipbldg. Div., Bethlehem Steel Co., Howard St., Quincy; *for mail,* 31 Lincoln Heights, Braintree 84, Mass.
Houghton, Richard Arnold, Jr. (J'42) (ABE), Structures Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* Cold Spring Harbor, L.I., N.Y.
Houghton, Robert Drew (J'41) (AFM), Capt., Operations Officer, Army Air Forces; *for mail,* 4 Berkeley Court, Wellesley Hills 82, Mass.
Houghton, William Maxwell ('25; '34) (NZ), Research Engr., United Shoe Mchy. Corp., Balch St., Beverly; *for mail,* 7 Outlook Court, Marblehead, Mass.
Houk, Warren Edward (J'40) (ZSN), Research Engr., Askania Regulator Co., 1603 S. Michigan Ave., Chicago 16, Ill.
Hourigan, Kenneth Francis (J'39) (CSBDE), Engr., Bldg. & Power Plant Valuation, Am. Can Co., 230 Park Ave., New York 17, N.Y.
House, Charles Pearce, Jr. (J'43) (FKMS), Lt. (j.g.), U.S.N.R., U.S.S. *Jubilant,* AM-255, F.P.O., New York, N.Y.; *for mail,* 201 N. Victoria Ave., Cleveland, Miss.
House, William Rozell (J'43), Ensign, U.S.N.R., 1711 S. Jefferson, Bay City, Mich.
Housel, Elwood Austin (J'41) (AMJCNS), Lab. Analyst A, Lockheed Aircraft Corp., 1705 Victory Pl., Burbank; *for mail,* 1761 North Canyon Dr., Hollywood 28, Calif.
Houser, Arthur M. ('12) (CJMS), Cons. Engr., Crane Co., 836 S. Michigan Ave., Chicago 5; *for mail,* Stevens Hotel, Michigan Ave., Chicago, Ill.
Houska, Robert William (J'44) (CDM), 21-57—42nd St., Astoria 5, N.Y.
Housley, Thomas P. ('21; '23; '35) (BL), Mech. Engr. Barrett Div., Allied Chem. & Dye Co., Frankford, Philadelphia 37, Pa.
Houston, Albert J. R. ('21; '34) (CHESM), *Student Award,* '21; Pres., Midland Constructors, Inc., 211 W. Wacker Dr., Chicago 6, Ill.
Houston, George Harrison ('13) (C), Sr. Partner, George H. Houston & Co., 52 Wall St., New York, N.Y.
Houston, Livingston Waddell ('19; '35) (CHM), Pres., Rensselaer Poly. Inst., 110—8th St.; *for mail,* 45 Maple Ave., Troy, N.Y.
Houston, Robert Ewing (J'37) (MGE), Mech. Engr., Fed. Tel. & Radio Labs., 67 Broad St., New York 4, N.Y.
Hovey, Otis Wadsworth ('44) (BJR), Devel. Engr., Alloys Devel. Co., 1102 Park Bldg., Pittsburgh 22; *for mail,* 328 S. Winebiddle St., Pittsburgh 24, Pa.
Hovey, Walter Freeman ('38) (BES), Engr., Sanderson & Porter, 52 William St., New York 5, N.Y.
Howard, Alan ('40) (BFS), Turbine Engrg. Dept., Gen. Elec. Co., Schenectady, N.Y.
Howard, Bernard (J'42) (OLHSDK), Mech. Engr., Mance Corp., 400 W. Broadway, New York; *for mail,* 2078 Wallace Ave., Bronx 60, N.Y.
Howard, Clifton Perry ('24) (CMDBJ), Supt. of Mfg., Rockwood Sprinkler Co., 38 Harlow St.; *for mail,* 18 Davidson Rd., Worcester 5, Mass.
Howard, Douglas Legate (J'44) (KCGOSH), Jr. Engr., Carrier Corp., 300 Ivy St., N.E., Atlanta, Ga.
Howard, Edgar Scott (J'35) (CMDLNG), Dir. of Trng., Minn. Mining & Mfg. Co., 900 Fauquire St., St. Paul 6, Minn.
Howard, Edward Athelstane (A'45), Pres., Aero. Engr., Edward A. Howard & Co., Engrs., Los Angeles; Gen. Mgr., Vernon Pattern, Vernon; *for mail,* 2173 N. Roosevelt Ave., Altadena, Calif.
Howard, Ernest E. ('13; F'45) (BNO), Partner, Howard, Needles, Tammen & Bergendoff, 921 Walnut St., Kansas City 6, Mo.
Howard, James Harris (J'38) (CMS), J. H. Howard & Co., 1009 Scanlan Bldg., Houston 2; *for mail,* 1455 Godwin St., Houston, Tex.
Howard, John Eager (J'23), Capt., Air Corps, U.S.A.; formerly Dist. Mgr., B. F. Sturtevant Co., 10 E. Mulberry St., Baltimore, Md.
Howard, John Francis ('41) (CM), Supvr., Salary & Wage Evaluation, Sperry Gyroscope Co., Inc., Great Neck, L.I., N.Y.
Howard, Karl Sharp ('19) (RDMASN), Asst. V.P., Gen. Steel Casting Corp., Eddystone, Pa.
Howard, Otis N. ('26; '29) (CES), Asst. Gen. Supt., Okla. Gas & Elec. Co., 321 N. Harvey, Oklahoma City, Okla.
Howard, Roy Emerson, Jr. (J'44) (NJKMF), Mech. Engr., Houston Branch, Tex. Div., Dow Chem. Co., M & M Bldg., Houston; *for mail,* 8314 Findlay St., Houston 12, Tex.
Howard, Thornton Wayne ('20; '35) (SCOKJB), Supt., Turbine & Mar. Installations, Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Howard, William Minnick (J'42) (KMDC), 24 Fuller Ave., Chatham, N.J.
Howarth, Elbert S. (J'42) (JNBMCA), Chief, Metal Working Div., Aluminum Research Labs., Box 772, New Kensington, Pa.
Howarth, H. A. Stevens ('08) (ONBMJC), *Melville Medallist,* '36; Propr., Firm of H. A. S. Howarth, Bearing Engr., 567 Pedley Rd., Philadelphia 28, Pa.
Howarth, Jacob M. ('13; '21; '35) (FHS), Maj., U.S.A., 1303 Glenlake Ave., Chicago 40, Ill.
Howarth, Walter O. (J'41) (ACM), 3138 Glendale Ave., Pittsburgh 27, Pa.
Howe, Hon. Clarence D. (H'41), Minister of Munitions & Supply, Canadian Govt., Ottawa, Ont., Can.
Howe, Everett D. ('25; '37) (KSEH), Assoc. Prof. Mech. Engrg., Asst. Dean of Engrg., Univ. of Calif., Berkeley 4, Calif.
Howe, Everett Wheeler (J'37) (YNCSOD), Plant Engr., U.S. Rubber Co., 355 Valley, Providence, R.I.
Howe, Harlan Lewis (J'44) (AKR), Jr. Engr., Test Work, Carrier Corp., 300 S. Geddes St., Syracuse; *for mail,* 412 Greenwood Pl., Syracuse 10, N.Y.
Howe, Jack Ludlam ('15; '22; '35) (R), Sales Rep., Otis Elevator Co., 1375 E. 6th St., Cleveland 14, Ohio.
Howe, James Alfred ('27) (CMJZ), Spec. Instrument Engr., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 594 Moonachie Ave., Wood-Ridge, N.J.
Howe, James Forrest ('15) (BDJ), Cons. Engr., 323 June St., Worcester 2, Mass.
Howe, James Taburn (J'43) (JNMDAF), Mech. Engr., Clinton Engr. Works, Tenn. Eastman Corp.; *for mail,* 107 E. Pasadena Rd., Oak Ridge, Tenn.
Howe, Paul Henry (J'37) (KCMFBD), Mech. Engr., Supvr. of Shipbldg., U.S.N., Kaiser Co., Inc., Vancouver, Wash.; *for mail,* 1606 N.E. 58th Ave., Portland 13, Ore.
Howe, Thomas Benton (J'45), Asst. Matls. Engr., Sperry Gyroscope Co., Inc., 55 Johnson St., Brooklyn 1; *for mail,* 900 St. Marks Ave., Brooklyn 13, N.Y.
Howe, William Walter (J'43), Supvr., Mar. Engr., United Engrg. Co., Ltd., 298 Steuart St., San Francisco; *for mail,* 107 Redding Way, San Rafael, Calif.
Howe, Winthrop Keith ('42) (RAJYNW), Cons. Engr., Gen. Ry. Signal Co., 801 West Ave., Rochester 1, N.Y.
Howell, Arthur K. ('12; '21) (EFS), Pres., A. K. Howell Co., 915 Olive St., St. Louis 1; *for mail,* 6336 Pershing Ave., St. Louis 5, Mo.
Howell, Francis Kitchell ('26) (SKDLF), Partner, Frank Howell Co., 412 Am. Bldg., Richmond 1, Va.
Howell, Glen Hiatt ('44) (OSNHKE), Owner, Howell Design & Engrg. Co., 1440 Maccabees Bldg., Detroit 2, Mich.
Howell, Harry Monroe (J'40) (CDM), Spec. Assignments, Control Mfg. Div., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 18 Chestnut St., Schenectady 7, N.Y.
Howell, Henry Wilson, Jr. (J'26) (CRAFBH), V.P., Cushman & Wakefield, Inc., 281 Madison Ave., New York 17; *for mail,* 4 W. 43rd St., New York 18, N.Y.
Howell, Jesse Monroe (A'41) (CJM), Works Mgr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Howell, John Donohoe ('33; '45) (NBO), Design Engr., York Research Corp., 63 Park Row, New York 7, N.Y.
Howell, Roger S. ('21; '35), Prof. Mech. Engrg., Dir., Evening Sch. of Applied Sci., Ga. Sch. of Tech., Atlanta, Ga.
Howland, Lewis A. ('10) (ACEFOS), Retired; 2288 Mott Ave., Box 357, Far Rockaway, L.I., N.Y.
Howorth, Robert Ward (J'41) (OCYHLS), Sales Engr., Allis-Chalmers Mfg. Co., 607 Chamber of Commerce Bldg., Cincinnati 2, Ohio.
Howse, Godfrey Lyle (J'38) (EFP), Owner, Firm of G. L. Howse, 712 Peoples Bank Bldg., Tyler, Tex.
Howson, Louis Richard ('29) (HOSEF), Partner, Alvord, Burdick & Howson, 20 N. Wacker Dr., Chicago 6, Ill.
Hoxie, George L. ('93; '04), 3312 Manhattan Ave., Manhattan Beach, Calif.
Hoxie, Vivian Walter ('15) (SFK), Dir., C. C. Moore & Co., Engrs., 450—1st St., San Francisco 5; *for mail,* 84 Commonwealth Ave., San Francisco 18, Calif.
Hoyle, Robert James, Jr. (J'41) (ZCSOLA), Mech. Engr., Stands. Lab., Taylor Instrument Cos., Ames St., Rochester; *for mail,* 55 Prince St., Rochester 7, N.Y.

Hoyt, Albert James ('16; '26) (CM), Mgr. of Operas., Cleveland Dist., Am. Steel & Wire Co., 614 Superior Ave., N.W., Cleveland 13; *for mail*, 2914 Torrington Rd., Shaker Heights 22, Ohio.
Hoyt, Charles P. (J'29) (CBEDJ), Asst. Dept. Head, Maint. & Constr., Stand. Oil Co. of La.; *for mail*, Box 864, Baton Rouge, La.
Hoyt, Raymond Dudler ('13), Pres., United Contrg. Co., 311 Stock Exch. Bldg., Portland 4, Ore.
Hoyt, Simes Thurston ('23; '35) (DHS), Cons. Engr., Castle & Cooke, Ltd., Merchant St., Honolulu 2, T.H.
Hoyt, Wm. R. ('29), Gen. Mgr., Yale & Towne Mfg. Co., 230 Henry St., Stamford, Conn.
Hrankowski, Stanley Victor (J'45) (NBAJOY), Design Engr., Strauss Fasteners, Inc., 154 W. 14th St., New York, N.Y.; *for mail*, 347 Hunter Ave., Plainfield, N.J.
Hrones, John Anthony ('37; '43) (ZNBHMA), Assoc. Prof., Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Hubbard, Cecil Robert ('27) (CHYM), V.P., Charge Prod., Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
Hubbard, Clyde W. ('31; '39), Lt., U.S.N.R., David Taylor Model Basin, Bur. of Ships, Navy Dept., Washington, D.C.
Hubbard, Eben Russell ('44) (CDT), R.D. 1, Millstone Rd., Wilton, Conn.
Hubbard, George Wallace ('13) (OF), Firm of Geo. W. Hubbard, 630 Ry. Exch., Chicago 4, Ill.
Hubbard, Karl Hammond ('36) (LBNCY), Ch. Engr., Charge Engrg. & Research, Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Hubbell, Charles Wesley ('13) (AJWCMS), Tech. Writer, Editor, Internatl. Correspondence Schs., 1000 Wyoming Ave., Scranton 9; *for mail*, 419 Mulberry St., Scranton 3, Pa.
Hubbell, George Wright (J'21) (CMWL), Babcock & Wilcox Co.; *for mail*, 224 E. Tuscarawas Ave., Barberton, Ohio.
Hubbell, John Edmund ('23) (FKSZ), Pat. Lawyer, 8 W. 40th St., New York 18, N.Y.
Hubbs, David Raymond (J'44) (ABC), Engrg. Div., Chrysler Corp., Highland Park; *for mail*, 144 Winona, Highland Park 3, Mich.
Hube, Arthur Burns (J'42) (EFSW), Test Engr., Ranger Aircraft Engines Div., Fairchild Eng. & Airplane Corp., Farmingdale, L.I.; *for mail*, 93-19 Hollis Court Blvd., Queens Village 8, N.Y.
Hubeny, Frank George (J'38), 756 Erudo St., Linden, N.J.
Huber, Alfred L. (J'40) (JNMBKL), Ch. Engr., Estimator, Jersey City Welding & Mch. Works, 400 Grand St., Jersey City 4; *for mail*, 153 West End Ave., Florham Park, Madison, N.J.
Huber, D. Grant (J'42) (HBYAGS), Demonstrator, Mech. Engrg., Univ. of Toronto; *for mail*, Apt. 608, 432 Jarvis St., Toronto, Ont., Can.
Huber, Ernest Godfrey ('42), Designer, Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee; *for mail*, 3551 S. Taylor Ave., Milwaukee 7, Wis.
Huber, George Joseph, Jr. (J'37) (SCRF), Prod. Asst. Supt., Ind. & Mich. Elec. Co., Mishawaka, Ind.
Huber, George L. ('30; '35) (SJBC), V.P., W. K. Mitchell & Co., Inc., 2940 Ellsworth St., Philadelphia 46, Pa.
Huber, Henry Jacques (J'43) (JBKS), Turbine Engr., Gen. Elec. Co., 1 River Rd.; *for mail*, 730 Union St., Schenectady 5, N.Y.
Hubert, Douglas G. (J'36) (SEFKMN), Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 107 Tibbetts Rd., Yonkers 5, N.Y.
Hubschmitt, Richard Wendell (J'39) (MKDLHY), Mch. Designer, West. Elec. Co., Inc., 403 Hudson St., New York 14, N.Y.
Huck, William F. ('30; '35) (GBNMCO), Huck Co., 420 Lexington Ave., New York 17; *for mail*, 81 Greenway Terrace, Forest Hills, L.I., N.Y.
Huckert, Jesse William ('29) (BMS), Asst. Editor, *Product Engineering*, McGraw-Hill Publ. Co., 330 W. 42nd St., New York; *for mail*, 225-05—130th Ave., Laurelton 13, N.Y.
Huckle, Myron Samuel ('30; '32) (OEJDCL), Pres. & Owner, M & H Industries, 1315 N. Ellison, Oklahoma City 6, Okla.
Huddle, Harley Edward (J'43) (MLDS), Ch. Engr., Howard Engrg. & Mfg. Co., 2235 Buck St., Cincinnati, Ohio; *for mail*, 1070 Lorain Court, Newport, Ky.
Huddleston, Stanley Crane (J'44) (HKACSB), Mech. Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 14737 Lorain Ave., Cleveland 11, Ohio.
Hudec, Edward Joseph Roy (J'43) (MJLHCZ), Ensign, U.S.N.R., 12116—29th Class, N.T.S. (I), Ft. Schuyler, New York 61, N.Y.
Hudson, Donald E. (J'38) (BZJOA), Asst. Prof. Mech. Engrg., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.
Hudson, Edward David (J'42) (DKBS), Ch. Engr., Oil Seed Products Co., 803 Bank of Am. Bldg., Fresno 1; *for mail*, 1321 N. Calaveras St., Fresno 4, Calif.
Hudson, Edward L. ('24; '35), 6 Carmody Terrace, Painesville, Ohio.
Hudson, Hendrik Reynolds (J'34) (AEJCMY), *Student Award*, '34; Instr., Mch. Shop, Tech. High Sch.; *for mail*, 782 Techwood Dr., N.W., Atlanta, Ga.
Hudson, Ray Morehouse ('30) (CLMTWY), Indus. Exec., New England Council, Statler Bldg., Boston 16; *for mail*, 20 Appleby Rd., Wellesley 81, Mass.
Hudson, Wellborn Royston, Jr. (J'41) (CAM), Staff Engr., Indus. Engrg., Pan Am. Airways, Inc., Box 3311, Miami 31; *for mail*, 4120 Lybyer Ave., Miami 33, Fla.
Hudson, Wilbur ('41) (DS), 5541 Everett Ave., Chicago 37, Ill.
Hudson, William S. ('28), Park Gate Farm, Tenterden, Kent, England.
Huebner, William Carl (A'29) (AGLZ), Dir., Huebner Labs., 305 E. 46th St., New York 17, N.Y.
Huet, André Philippe Jean ('45) (AKRS), Pres., Dir. Gen., Compagnie des Surchauffeurs, 29 rue de Berri, Paris 8ᵉ, France.
Huettl, Wm. A. ('31; '35) (BMS), Draftsman, Atlas Valve Co., 282 South St., Newark; *for mail*, 618 Floral Ave., Elizabeth, N.J.
Huey, Claude L. ('41) (EFS), Sales Engr., Babcock & Wilcox Co., 1203 Chandler Bldg., Atlanta, Ga.
Huey, John Spencer ('34) (JLOSWY), Cons. Engr., 915 Maritime Bldg., New Orleans 12, La.
Huff, George Franklin ('26) (FHS), Propr., Engrg. Products Co., 831 Grant Bldg., Pittsburgh 19; *for mail*, 6040 Bryant St., Pittsburgh 6, Pa.
Huff, Norman M. (J'40) (CJM), Prod. & Test Engr., Harper Wyman Co., 8562 Vincennes Ave.; *for mail*, 9314 S. Green St., Chicago, Ill.
Huffman, C. A. ('42), 1302—25th Ave., Longview, Wash.
Huffman, C. Warren ('36) (AEHMNO), Cons. Engr., Box 457, Big Bear Lake, Calif.
Huffman, Robert Lyal (J'41) (ZJMN), Devel. Engr., Automatic Elec. Co., 1033 W. Van Buren St., Chicago 7; *for mail*, 1354 Argyle St., Chicago 40, Ill.
Huffman, Samuel A. (J'27) (ABNJCM), Partner, Micro Model Co., Box 442, Pasadena 18, Calif.
Hufford, James D. (J'43) (ACM), 852 Talbot St., Jacksonville 5, Fla.
Huge, Ernest C., Jr. ('30; '41), Research & Serv. Engr., Babcock & Wilcox Co., Barberton; *for mail*, 283 Castle Blvd., Akron 3, Ohio.
Hugenbruch, Ernest Raymond (J'41) (CMAHSB), 913 Main St., Paterson 3, N.J.
Hugger, Richard Hudson (J'24) (CNMYWT), Gen. Mgr., Cordo Chem. Corp., 34 Smith St.; *for mail*, 37 George Ave., Norwalk, Conn.
Hughes, Arthur Douglas ('32; '42) (SEFRKZ), Research Engr., Gas Turbine Devel. Div., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 2073 S. 76th St., West Allis 14, Wis.
Hughes, Brooks (J'41), Linde Air Products Co.; *for mail*, 1100—50th St. N., Birmingham, Ala.
Hughes, Burton Shelley ('08), Pres., Zaremba Co., 506 Crosby Bldg.; *for mail*, Buffalo Athletic Club, Buffalo 2, N.Y.
Hughes, Edward R. ('37) (OYJK), Schwarz, Hughes & McCarthy, 225 Broadway, New York 7, N.Y.
Hughes, Elmer Lisle ('44) (SDEFAY), Owner, Hughes Mchy. Co., 4034 Broadway, Kansas City 2, Mo.
Hughes, Frederick George ('13) (CM), Gen. Mgr., New Departure Div., Gen. Motors Corp.; *for mail*, 116 Belridge Rd., Bristol, Conn.
Hughes, George Griffin ('45), Dean of Engrg., Southwest. La. Inst., College of Engrg., Southwest. La. Inst. Sta.; *for mail*, 211 W. College Ave., Lafayette, La.
Hughes, Harley Roland (J'38) (LMCA), Plant Supt., Colgate-Palmolive-Peet Co., Jeffersonville, Ind.; *for mail*, Anchorage, Ky.
Hughes, Henry E. ('16) (CEFHS), 546 Stuyvesant Ave., Rutherford, N.J.
Hughes, Hunter Russel, Jr. (J'41) (ACES), Southwest. Editor, *Southern Power and Industry*, W.R.C. Smith Publ. Co., 1341 Liberty Bank Bldg., Dallas 1, Tex.
Hughes, J. Stanley ('42) (CKL), 97 Chatham Ave., Buffalo, N.Y.
Hughes, Paul Ambrose (J'41) (MJCASE), Lt. (j.g.), U.S.N.R.; 901 Bradford Ave., Nashville 4, Tenn.
Hughes, Raymond LeRoy, Jr. (J'39) (OKSCEF), 2815 Guilford Ave., Baltimore 18, Md.
Hughes, Robert Emmet (J'44), 2805 Pond Pl., New York 58, N.Y.
Hughes, Robert G. ('08), Franklin Lake Rd., Oakland, N.J.
Hughins, Gordon Roy ('39) (CEFS), Ch. Mech. Engr., St. Louis Admin. Center, 4300 Goodfellow, St. Louis 20, Mo.; *for mail*, Box 47, Wood River, Ill.
Hugil, Wilfred Charles, Jr. (J'36) (ABHC), Exec. Engr., Exper. Towing Tank, Stevens Inst. of Tech., Hoboken; *for mail*, 291 Millburn Ave., Millburn, N.J.
Hulbert, Robert Eugene (J'44) (NMABYC), Ensign, E (L), U.S.N.R., Pat. Engr., Box 1663, Santa Fe, New Mex.
Hulbert, William George ('25) (BCJRD), V.P., Taylor Wharton Iron & Steel Co., Box 229, Easton, Pa.
Hull, Burton E. ('16) (CE), Pres., Mgr., Tex. Pipe Line Co., Box 2332, Houston, Tex.
Hull, Donald Robert ('44) (TNLB), Engrg. Research Supvr., E. I. du Pont de Nemours & Co., Wilmington; *for mail*, 1 Delaware Ave., Newport, Del.
Hull, Edwin Hodges (J'24) (BYZAS), Research Lab., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 226 Ballston Ave., Scotia, N.Y.
Hull, James Henry, Jr. (J'40) (NER), Jr. Mech. Engr., Atlas Powder Co., Wilmington; *for mail*, 2505 W. 18th St., Wilmington 172, Del.
Hull, Jesse Lyle (J'36) (BEF), Capt., U.S.N., Comdr. Sub. Sqd. 36, F.P.O., San Francisco, Calif.
Hull, John Benjamin ('42) (MR), Dist. Engr., Westinghouse Air Brake Co., 1101 Matson Bldg., San Francisco 5, Calif.
Hull, Richard Duhart (J'43) (NCS), Lt. (j.g.), U.S.N.R., LST-223, F.P.O., New York, N.Y.; *for mail*, 137 High St., Montclair, N.J.
Hull, Thomas N., Jr. (J'42) (BNKJAC), Engr., Gas Turbine Engrg. Div., Gen. Elec. Co.; *for mail*, 225 Jackson Ave., Schenectady, N.Y.
Hulse, George Egbert ('20; F'38) (BJKNR), *Manager*, '40-'43; Ch. Engr., Safety Car Htg. & Ltg. Co., Inc., Box 904, New Haven 4, Conn.
Hulsizer, Robert Leon ('21; '37) (CDFKLS), Supvr., Power & Steam, N.Y. & Pa. Co., Inc., Lock Haven, Pa.
Hulson, John William ('41) (R), Pres., Gen. Mgr., Hulson Co., 332 S. Michigan Ave., Chicago 4, Ill.
Hultan, K. A. ('26; '35), Box 5331, New Orleans, La.
Hultberg, Arvid John (J'43) (CDJMN), Counselor, 2nd Serv. Comd., 30th Co., 1262nd Serv. Comd. Unit, Ft. Dix, N.J.
Humbert, William Frank ('43) (TOMC), Plant Engr., Mfg. Div., Marshall Field & Co., Spray, N.C.
Humble, James Ward (J'38) (ESF), Asst. to Ch. Engr., Combustion Equip. Div., Todd Shipyard Corp., 601 W. 26th St., New York 1; *for mail*, 1430 W. 4th St., Brooklyn 4, N.Y.
Humiston, Hiram Aiken (A'46) (ELRS), Lub. Engr., Stand. Oil Co., 1708 Midland Bldg., Cleveland 15, Ohio.
Hummel, Joseph O. P. ('30; '36) (CMJDLG), Assoc. Prof. Indus. Engrg., The Pa. State College; *for mail*, 805 N. Holmes St., State College, Pa.
Hummel, Ragnar A. ('22) (CDFS), Pres., Lone Star Cement Corp., 342 Madison Ave., New York 17, N.Y.
Hummer, Ralph Gorman ('42) (MJCB), Defiance Automatic Screw Co., Ft. Wayne Rd., Defiance, Ohio.
Humphrey, Donald Dell (J'44) (NOMZFK), Jr. Mech. Engr., Applied Physics Lab., Univ. of Wash., Seattle 5; *for mail*, Clinton, Wash.
Humphrey, Howard M. (J'41) (EHP), 1st Lt., U.S.M.R.O., VMTB-232, Mar. Aircraft Gp. II, Fleet Mar. Aircraft Wing, Flight Ech., Mar. Air Corps, F.P.O., San Francisco, Calif.
Humphrey, Paul E. (J'37), Giltner, Neb.
Humphrey, (Noah) William ('40) (HWA), Assoc. Mech. Engr., Wash. Water Power Co., 825 W. Trent St.; *for mail*, W. 203—27th Ave., Spokane, Wash.
Humphreys, Cyril G. ('32; '35) (FKS), Apt. 7G, 14 Metropolitan Oval, New York 62, N.Y.
Hundley, Frank George (J'31) (CHMJ), Engr., Neptune Meter Co., 192 Jackson Ave., Long Island City; *for mail*, 217-05—49th Ave., Bayside, N.Y.
Hunfalvy, Harold Albert (J'43) (NMJ), Mech. Drafting Instr., Henry Abbott Tech. Sch., Main St.; *for mail*, 15 Starr Ave., Danbury, Conn.
Hungate, Louis Hill, Jr. ('42) (FEC), Maj., QM. Corps, 5114 Goodwin, Dallas 6, Tex.
Hungerford, Warren H. ('19; '25) (BJM), Pres., Evanston Stamping Co., Evanston, Ill.
Hunsaker, Jerome C. ('33; H'42) (BAHJ), *Vice-President*, '39-'41; Prof., Charge Mech. Engrg. Dept., Mass. Inst. of Tech., Cambridge, Mass.
Hunt, Clarkson T. ('26; '35) (R), M.M., Pa. R.R. Co., Philadelphia; *for mail*, 222 E. Orange St., Lancaster, Pa.
Hunt, Edward Eyre ('41) (C), Dir., Italian Branch, Office of Foreign Liquidation Commr., U.S. Dept. of State; *for mail*, 943 Massachusetts Ave., N.W., Washington, D.C.
Hunt, Franklin Bucknell ('24; '34) (SFK), Ch. Engr., Compressed Gas Div., Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Hunt, George ('43), John Fensom Co., 603 Atlantic Life Bldg., Richmond 19; *for mail*, Box 2253, Richmond 17, Va.
Hunt, Gorman Earl (J'42) (SKBA), Field Engr., Allis-Chalmers Mfg. Co., West Allis; *for mail*, 802 N. 27th St., Milwaukee 8, Wis.
Hunt, Herve Henry (J'43) (LCOSZW), Lt. (j.g.), U.S.N.R., Electronic Prod. Officer, Office of Asst. Indus. Mgr., U.S.N., Ferry Bldg., San Francisco, Calif.
Hunt, Homer H. (J'41) (JMS), Ensign E-V(S), U.S.N.R., 2226—21st Ave. S., Birmingham, Ala.
Hunt, Horace Sinclair ('19) (SHEFO), Pres., Gen. Mgr., Fargo Engrg. Co., 120 W. Michigan Ave., Jackson, Mich.
Hunt, Howard E. (J'41) (HKW), Engr., Control Div., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 11 N. Church St., Schenectady 5, N.Y.
Hunt, James Edward ('21; '35) (SHC), Supt., Generating Stas., Philadelphia Elec. Co., 9th & Sansom Sts., Philadelphia 5, Pa.
Hunt, James Francis ('38; '42) (HKS), Cory & Joslin, Inc., 50 Hawthorne St., San Francisco 5, Calif.

Hunt, James Webster (J'45) (SEBR), Asst. Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 2146 S. 7th St., Milwaukee 14, Wis.

Hunt, John Martin ('44), Distr. Mgr., Sales Engr., Aerovent Fan Co., 1629 Quarrier St., Charleston, W.Va.

Hunt, Melvin W. (J'38) (BLS), Supt., Maint. & Serv., Dow Chem. Co., Box 629, Torrance; *for mail,* 1211 W. 160th St., Gardena, Calif.

Hunt, Myron C. (J'44) (DJK), Asst. Prod. Engr., Perfect Circle Co., Hagerstown, Ind.

Hunt, Nathan C. ('34), Pres., C. B. Hunt & Son, 1913 E. State St., Salem, Ohio.

Hunt, Robert ('23) (ER), Berkley Mch. Works & Fdy. Co., Inc., Berkley, Norfolk, Va.

Hunt, Robert Everett (J'46), Lt. (j.g.), U.S.N.R., 25 Wilson Rd., Stoneham, Mass.

Hunt, Samuel Parker ('25) (CFHJLS), Cons. Engr., 852 Elm St., Manchester, N.H.

Hunt, Victor F. (J'41) (BJM), Time Study Engr., Aluminum Co. of Am., 3311 Dunn Rd.; *for mail,* 2365 E. Grand Blvd., Detroit, Mich.

Hunt, William Floyd ('94; A'03), Life Associate; c/o Pearson Hunt, Morgan Hall, Soldier's Field, Boston, Mass.

Hunt, William Frederick ('44; '45) (CHJMNZ), Ch. Engr., Extrusion Div., Reynolds Metals Co., 4th & Montana Sts., Louisville, Ky.

Hunt, William Roland (J'42) (ANG), Engr., Flight Test Analysis, Plant 2, Boeing Aircraft Co., Seattle; *for mail,* 3013—62nd Ave., S.W., Seattle 6, Wash.

Hunter, Anthony ('44; '46) (BJO), Mech. Engr., Charge Specification Sec., Donald R. Warren Co., 500 S. Figueroa St., Los Angeles; *for mail,* 1610 N. Normandie Ave., Los Angeles 27, Calif.

Hunter, Arthur T. (J'23), 1616 Walnut St., Philadelphia, Pa.

Hunter, Charles Frederick ('45) (CFS), Dist. Mgr., No. Ind. Pub. Serv. Co., 300 S. Main; *for mail,* 510 Vistula St., Elkhart, Ind.

Hunter, Charles William, Jr. (J'41) (CHK), Lt., Army Air Forces; *for mail,* 205 Hilton Ave., Maplewood, N.J.

Hunter, Clarence James (A'36) (SRJK), Tech. Dir., Dampney Co. of Am., 1243 River St., Hyde Park, Boston 36, Mass.

Hunter, Frank Raymond, Jr. (J'41) (SHCALK), Sales Engr., Allis-Chalmers Mfg. Co., 1410 Waldheim Bldg., Kansas City 6; *for mail,* 120 N. Bales Ave., Kansas City 1, Mo.

Hunter, Franklin John (J'45) (HRS), Engrg. Estimator, Worthington Pump & Mchy. Corp., 400 W. Madison Ave., Chicago, Ill.

Hunter, Geo. E. ('90), Retired; Box 553 Hendersonville, N.C.

Hunter, J. R. ('18; '35), Maint. Analysis, Goodyear Tire & Rubber Co., 1144 E. Market St., Akron, Ohio.

Hunter, Jas. D. ('94), Pres., Jas. Hunter Mch. Co., North Adams, Mass.

Hunter, James Edward, Jr. (J'44), 36 Case Ave., Cranston 10, R.I.

Hunter, James Francis ('99; '09) (EO), Cons. Engr., Gas Utilities, 17 Longue Vue Ave., New Rochelle, N.Y.

Hunter, John (J'46) (HSW), 255 Cedar Rd., River Edge Manor, N.J.

Hunter, John ('09; F'39), *Manager,* '13-'16, *Vice-President,* '17-'19; Retired; *for mail,* 2346 Fairbanks Ave., Winter Park, Fla.

Hunter, John A., Jr. (J'33) (CJMDEL), Market Analyst, U.S. Steel Corp. of Del., 436—7th Ave., Pittsburgh 30, Pa.

Hunter, John Alexander ('09) (SFJKEM), Retired; 123 Miller Ave., Maryville, Tenn.

Hunter John Arthur (J'44) (SFEBCK), Steam Serv. Engr., Westinghouse Elec. Corp., 10 High St., Boston 10; *for mail,* 10 Glenwood St., Worcester 3, Mass.

Hunter, John S. (J'41), Army Air Forces, 700th Base Unit, Sec. C., Wright Field, Dayton, Ohio.

Hunter, Louis Nathaniel ('36) (FJKCL), V.P., Research, Natl. Radiator Co., 221 Central Ave., Johnstown, Pa.

Hunter, William Laurin ('40) (SCMHEJ), Ch. Engr., No. Equip. Co., 1945 Grove Dr.; *for mail,* 1253 W. 10th St., Erie, Pa.

Hunting, Ronald Walter ('39), Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5; *for mail,* 738—16th Ave., Prospect Park, Pa.

Huntington, Fred M. (J'38) (CDL), Staff Engr., Booz, Allen & Hamilton, 285 Madison Ave., New York 17, N.Y.

Huntington, Robert H. (J'43) (CJM), Abrasive Engr., Carborundum Co., 1150 W. 3rd St., Cleveland 13; *for mail,* 3130 Albion Rd., Cleveland 20, Ohio.

Huntly, Phil C. ('38) (BJNORY), Prof. & Dir. of Civ. Engrg., Ill. Inst. & Armour Research Foundation, 3300 Federal St., Chicago 16, Ill.

Hunton, Fred Wilson (J'41) (JBK), Mech. Engr., Design, Golden State Co., Ltd., 425 Battery St., San Francisco; *for mail,* 2110 Hearst Ave., Berkeley 4, Calif.

Huntoon, Charles H. (J'33) (CMN), Maj., Ord. Dept., U.S.A., Contracting Officer, Detroit Ord. Dist., Natl. Bank Bldg., Detroit; *for mail,* 15880 Woodingham Dr., Detroit 21, Mich.

Hurley, George Henry ('45), Maj., U.S.A.; *for mail,* 1084 Richmond Terrace, Staten Island, N.Y.

Hurley, Raymond Bryson (J'41) (JM), U.S. Army; 8 Winston Pl., Schenectady, N.Y.

Hurley, Wesley Veach (J'43) (EKS), Suite 3, 122 Washington St., Marblehead, Mass.

Hursh, Robert William (J'37) (AYENBT), Tire Devel. Engr., B. F. Goodrich Co., 500 S. Main St., Akron; *for mail,* Box 2081, Goodrich St. Sta., Akron 11, Ohio.

Hurst, John Frank ('22) (E), Supt., Steam Prod., Louisville Gas & Elec. Co., 311 W. Chestnut St., Louisville; *for mail,* 1288 Everett Ave., Louisville 4, Ky.

Hurt, James M. (J'43) (SFEKHC), Mech. Engr., Dallas Power & Light Co., 1506 Commerce, Dallas 1, Tex.

Hurt, Reuben M. ('34; '41) (CHEBM), Asst. Div. Supt., Carter Oil Co., Box 1151, Seminole, Okla.

Hurt, Robert Francis (J'42) (CJMNA), Mgr., Appliance Div., Simpson Steel Co., Box 203, Yernon, Los Angeles 11; *for mail,* 4318 Bel Air Dr., La Canada, Calif.

Hurt, William Coates, Jr. ('29; '39) (CMNB), Capt., Corps of Engrs., Asst. Engrg. Officer, Engr. Sec., Hdq. Base X, A.P.O. 358, c/o Postmaster, San Francisco, Calif.; *for mail,* 43 Arlington St., Pittsfield, Mass.

Hurvitz, Harold (J'44) (EJS), 5514 S. Drexel Ave., Chicago 37, Ill.

Hushen, Fred (J'40) (HSA), 366 E. 26th St., Brooklyn, N.Y.

Hushen, Thomas Michael, Jr. (J'42), 1121 Parkwood Blvd., Schenectady 8, N.Y.

Huss, Harry O. ('34; '45) (ABHNJK), Devel. Engr., Chem. Warfare Serv., War Dept., Edgewood Arsenal, Md.

Hussander, George Carl, Jr. (J'42) (AZ), Lt. (j.g.), U.S.N.R., Rocket Trng. Unit, Box A, Naval Air Sta., El Centro, Calif.; *for mail,* 7337 Ibsen St., Chicago 31, Ill.

Hussey, Thomas O. ('41) (CDMN), Maint. Engr., Div. Engrs. Office, Pub. Bldgs. Admin., Fed. Works Agency, 731 Custom House, New York 4; *for mail,* 179-02—135th Ave., Springfield Gardens 13, N.Y.

Hussey, William E. ('04), Retired; 331 Scudder Ave., Northport, N.Y.

Hustead, Thomas Eastman (J'41) (MJNC), Sales Engr., Hyatt Bearings Div., Gen. Motors Corp., 702 Grant Bldg., Pittsburgh 19; *for mail,* 1230 Hulton Rd., Oakmont, Pa.

Huston, Alfred B. (J'40) (JMS), 119 McKinley St., Bennington, Vt.

Huston, Charles Lukens (A'87), V.P., Lukens Steel Co., S. 1st Ave.; *for mail,* 64 S. 1st Ave., Coatesville, Pa.

Huston, Edwin H. (J'41) (A), Lt., Army Air Forces, 237 E. High St., Lisbon, Ohio.

Huston, Frederick Pelot ('39) (RJFEKS), Charge R.R. Devel., Devel. & Research Div., Internatl. Nickel Co., Inc., 67 Wall St., New York 5, N.Y.; *for mail,* 281 South Ave., Fanwood, N.J.

Hustvedt, Erling H. (J'41), Lt., U.S.N.R., U.S.S. *McCalla,* DD-488, F.P.O., San Francisco, Calif.

Hutchcraft, David Brent (J'39) (E), 1507 E. 20th St., Tulsa, Okla.

Hutchcraft, David K. ('18; '35) (ABCEHJ), Sales Mgr., Clark Bros. Co., Inc., 125 W. 1st St., Tulsa, Okla.

Hutchings, Albert Randall ('46) (CJM), Prod. Mgr., B. F. Perkins & Son, Inc., Holyoke; *for mail,* 39 Sheridan Rd., Wellesley Hills, Mass.

Hutchings, Warren (J'41) (BJM), 4315 W. Van Buren St., Chicago 24, Ill.

Hutchins, William Edward ('39; '45) (CMD), Dist. Engr., Liberty Mutual Ins. Co., 643 Liberty Ave., Pittsburgh 22, Pa.; *for mail,* 4 Highland Ave., Montclair, N.J.

Hutchinson, Arthur (A'37) (EFR), Smoke Abatement Engr., City of Cleveland, 601 Lakeside Ave., Cleveland 14, Ohio.

Hutchinson, Arthur H. ('99), 44 Park Lane, N.E., Atlanta, Ga.

Hutchinson, David Wark (J'45), Westinghouse Elec. Corp., Essington 13; *for mail,* 822—14th Ave., Prospect Park, Pa.

Hutchinson, Davis (J'42), Instr., Ore. State College; *for mail,* 128 N. 11th St., Corvallis, Ore.

Hutchinson, Donald Marshall (J'39) (TDKO), Sales Engr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., Cresmont & Haddon Ave., Camden; *for mail,* 210 Mt. Vernon Ave., Haddonfield, N.J.

Hutchinson, Ely Champion ('12; F'36) (HSCEAL), *Manager,* '28-'31, *Vice-President,* '33-'35; Chief, Prod. Research & Devel. Div., U.S. Dept. of Commerce, Washington 25; *for mail,* Brighton Hotel, 2123 California St., N.W., Washington 8, D.C.

Hutchinson, James Bruce (J'43) (MCJBHY), Mech. Engr., No. Elec. Co., Ltd., P.O. Drawer 369; *for mail,* 769 Mount Royal East, Montreal 34, Que., Can.

Hutchinson, John Alfred ('16; '21) (CZSDM), Ch. Engr., R. Wallace & Sons Mfg. Co., Quinnipiac St., Wallingford; *for mail,* 11 Lambert Ave., Meriden, Conn.

Hutchison, Fred P. (J'18), 125 Main St., Orange, N.J.

Huthsteiner, Robert Eugen ('34; '35) (CEF), V.P., Controller, Cummins Eng. Co., Inc., 5th & Wilson St.; *for mail,* Box 512, Columbus, Ind.

Hutt, Arthur R. (J'35) (ZMABH), Engr., Sperry Gyroscope Co., Marcus Ave., Great Neck; *for mail,* Box 533, 133 Manhattan Ave., Westbury, L.I., N.Y.

Huttinger, William Reynolds (J'06) (ERS), V.P., Ch. Engr., Elec. Power Equip. Co., 412 N. 18th St., Philadelphia; *for mail,* 86 E. Greenwood Ave., Lansdowne, Pa.

Hutton, Frank Edward ('44), Paper Mill Specialist, Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* R.F.D. 1, Wyckoff Ave., Wyckoff, N.J.

Hutton, Junius Oscar (J'41), Consltd. Vultee Aircraft Corp.; *for mail,* 338 Elm St., Reno, Nev.

Hutton, S. E. ('12) (CSHBDM), Asst. to Supvg. Engr., Bur. of Reclamation, Dept. of Int., Coulee Dam, Wash.

Huvane, James F. ('24; '25; '35) (ES), Mgr., Compressor Div., Chicago Pneumatic Tool Co., 6 E. 44th St., New York 17, N.Y.

Huxford, George T. ('43), c/o Waymoth Corp., Pawtucket, R.I.

Huyser, Francis C. ('34; '46) (BCNMZJ), Ch. Engr., Riehle Testing Mchs. Div., Am. Mch. & Metals, Inc.; *for mail,* 2321—7th St., East Moline, Ill.

Hyatt, Robert Shideler ('40) (SFK), Steam & Power Engr., Procter & Gamble Co., Mar. Bldg., Ivorydale; *for mail,* 323 Durrell Ave., Wyoming, Ohio.

Hyde, Edward Bolton, Jr .('46) (KOS), Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 170-11—35th Ave., Flushing, N.Y.

Hyde, Edward M. (J'40) (ABKNMT), Spec. Tool & Mch. Design Engr., Proctor & Schwartz, Inc., 7th & Tabor Rd., Philadelphia 20, Pa.

Hyde, Eric Francis ('42) (BFHKSZ), Hyde & Bobbio, 405 Trans. Bldg., Detroit 26; *for mail,* 708 Oakland Ave., Birmingham, Mich.

Hyde, George Gaynor ('39) (JMO), Cons. Engr., Rm. 1503, 450—7th Ave., New York 1, N.Y.

Hyde, Glenn Cook ('16; '35) (CS), Pres., La. Pub. Utilities Co., Inc. & Gulf Pub. Serv. Co., Inc., 1510 Mercantile Bank Bldg., Dallas 1, Tex.

Hyde, Harlan W. (J'36) (CDLSZ), Asst. Plant Engr., Lever Bros. Co., 5300 Holabird Ave., Baltimore 3, Md.

Hyde, James Patten ('28; '35) (EBNR), Engr., Diesel Eng. Div., Ingersoll-Rand Co.; *for mail,* 665 Hillcrest Blvd., Phillipsburg, N.J.

Hyland, William Leo ('41) (AHO), Sr. Engr., Fay, Spofford & Thorndike, 11 Beacon St., Boston, Mass.

Hyler, Loiell LeRoi ('42; '45) (BHOAJ), Ch. Engr., Tractor Equip. Div., La Plant-Choate Co., Inc., 1st Ave.; *for mail,* 2325 Meadowbrook Dr., S.E., Cedar Rapids, Iowa.

Hymans, Fred'k ('12) (BH), Cons. Engr., Otis Elev. Co., 260—11th Ave., New York 1, N.Y.

Hynes, Lee Powers ('29) (LKSY), Ch. Engr., Hynes Elec. Htg. Co., West & Clinton Sts., Camden, N.J.

I

Ibold, Peter Arthur ('41) (BJNSCH), Design Engr., Farrel-Birmingham, Inc., Ansonia; *for mail,* 71 Alameda Pl., Bridgeport 4, Conn.

Iddles, Alfred ('13; '17; '22; F'36) (SFCRKB), *Manager,* '34-'37, *Vice-President,* '38-'40; V.P., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.

Iddles, Gordon (J'37) (FSKOCD), Capt., Corps of Engrs., U.S.A., 203 Orchard Way, Wayne, Pa.

Idenden, Francis Stevenson (J'43) (COHE), Mech. Engr., Caribbean Petroleum Co., Apartado 19, Maracaibo, Venezuela, S.A.

Iglehart, R. Lannert (J'36) (CBLKNS), Asst. Ch. Engr., Shell Devel. Co., 4560 Horton St., Emeryville 8, Calif.

Igleheart, George Priest ('18; '22; '35) (CMJNW), V.P., Engrg. Prod., Stapling Mchs. Co., Rockaway, N.J.

Igoe, Bert John (J'42), Test Engr., Ford Instrument Co., 32-36—47th Ave., Long Island City; *for mail,* 141-70—85th Rd., Briarwood, L.I., N.Y.

Ihasz, James Michael (J'39) (HLEF), Sr. Engrg. Writer, Ford Instrument Co., 32-36—47th Ave., Long Island City; *for mail,* 209 Verbena Ave., Floral Park, L.I., N.Y.

Ilaria, Alfred (J'44) (ONBZAS), Engr., Strobel & Salzman, 500—5th Ave., New York; *for mail,* Apt. 6E, 214 Riverside Dr., New York 25, N.Y.

Iles, Frank Berneigh (J'45), Ch. Quarterman Machinist X36, U.S. Naval Drydocks, Terminal Island; *for mail,* 309 E. 46th, Long Beach, Calif.

Ilg, Henry Lucas (J'39) (ACL), Capt., Army Air Forces, Wright Field, Dayton; *for mail,* 116 Patterson Village Dr., Dayton 9, Ohio.

Iliff, Wm. L. ('17; '25), 246 Valley Rd., Montclair, N.J.

Illes, Stephen R. (J'41) (CFSZ), Sales Engr., J. L. Vergilio Co., 3030 Euclid Ave., Cleveland 15, Ohio.

Illfelder, Edgar Leopold (J'41) (CMD), 222 W. 23rd St., New York 11, N.Y.

Illmer, Louis ('13) (EAO), Pat. Atty., Professional Research Engr., 24 N. Church St., Cortland, N.Y.

Imbembo, Emil A. (J'32) (JMLB), Sr. Metallurgist, Matl. Lab., N.Y. Naval Shipyard, Brooklyn; *for mail,* 2641 Marion Ave., New York 58, N.Y.

Imbeninato, Paul Albert (J'43) (AHJ), Test Engr., Eclipse Aviation Corp., Teterboro; *for mail,* 88 Dickerson St., Newark 4, N.J.

Imbornoni, Albert John (J'45), Test Engr., Worthington Pump & Mchy. Corp., Harrison, N.J.; *for mail,* 2460 Nostrand Ave., Brooklyn 10, N.Y.

Immel, John Richard (J'42) (YABJS), Mech. Engr., U.S. Rubber Co., 1230—6th Ave., New York 20, N.Y.; *for mail,* 7 Pease Ave., Verona, N.J.
Immele, Leonard Bernard (J'41), Aviation Cadet, Aviation Cadet Det., Class 42-7, Bks. T-565, Chanute Field, Ill.; *for mail,* Pilot Grove, Mo.
Impagliazzo, A. Michael (J'36) (KSB), Engr., Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.
Ims, Edward Charles (J'31) (CA), Head Timekeeper, Gen. Elec. Co., East Lake Rd., Erie; *for mail,* R.D. 1, Townline Rd., Fairview, Pa.
Imse, Philip Joseph (J'39) (CJB), Works Mgr.'s Dept., Chain Belt Co., 1600 W. Bruce St., Milwaukee 4, Wis.
Infanger, Arthur William (J'42) (EBHRYZ), Vibration Engr., Am. Loco. Co., 100 Orchard St., Auburn; *for mail,* Pine Springs, Cayuga, N.Y.
Ingalls, C. H. ('19) 1214 Union Trust Bldg., Providence, R.I.
Ingalls, David ('43) (AJMC), Ch. Engr., Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
Ingalls, R. I., Jr. (J'30), Jr. Engr., Birmingham Tank Co., Birmingham, Ala.
Ingersoll, Raymond Crary (J'40) (HKABE), 350 Cabrini Blvd., New York 33, N.Y.
Ingersoll, Robert Joseph ('44) (COBGJY), Engr., U.S. Engr. Office, Box 1070, Nashville 1; *for mail,* 1706 Temple Ave., Nashville 4, Tenn.
Ingle, Henry Wilson ('20; '35) (MND), Container Engr., Hartford-Empire Co., Homestead Ave., Hartford 1; *for mail,* 35 Giddings Ave., Windsor, Conn.
Inglee, Clinton ('20) (HYA), Pres., Gen. Mgr., Natl. Water Main Cleaning Co., 30 Church St., New York 7, N.Y.
Inglee, Clinton F. (J'41) (AESC), Project Engr., Ranger Aircraft Engines Div., Fairchild Eng. & Airplane Corp., Farmingdale; *for mail,* 119-A Union Ave., Amityville, N.Y.
Ingles, John Spencer (J'36) (RFKZ), Hulson Co., 332 S. Michigan Ave., Chicago 4; *for mail,* 1844 Sycamore Rd., Homewood, Ill.
Inglis, Robert Napier ('17), Retired; 617 Lawrence Ave., Westfield, N.J.
Inglish, Harold C. (J'35) (ABHWZN), Struc. Gp. Engr., Lockheed Aircraft Corp.; *for mail,* 201 W. Ash Ave., Burbank, Calif.
Ingold, John Frederick, Jr. (J'42) (YLSD), Indus. Engr., E. I. du Pont de Nemours & Co., 85 Mill Plain Rd., Fairfield, Conn.
Ingraham, Arthur K. ('39) (AHS), Capt., Corps of Engrs., Overseas Constr. Comd., A.P.O. 18642, c/o Postmaster, San Francisco, Calif.
Ingram, George, Jr. (J'43) (CMSO), Staff Engr., Stevenson, Jordan & Harrison, Inc., 19 W. 44th St., New York 18; *for mail,* 1113 Madison Ave., New York 28, N.Y.
Ingram, Joseph L. ('42), Lt., U.S.C.G., 406 P.O. Bldg., Tampa, Fla.
Ingram, William Torbert ('36; '44) (HNZBY), Exper. Engr., Chrysler Corp., 12800 Oakland St.; *for mail,* 145 Colorado St., Detroit 3, Mich.
Inman-Emery, J. Inman ('30), Air Ministry, British Govt., Adastral House, London S.W. 7; *for mail,* 81 Lichfield Court, Sheen Rd., Richmond, Surrey, England.
Intemann, Hermann Kolle (J'31) (CYTMDL), Asst. Gen. Sales Mgr., Bakelite Corp., 30 E. 42nd St., New York 17, N.Y.
Inwright, John Arnold ('19; '26; '35) (SCF), Supt., Burlington Generating Sta., Pub. Serv. Elec. & Gas Co., W. Pearl St., Burlington; *for mail,* 325 Chester Ave., Moorestown, N.J.
Iorillo, Domenick Joseph (J'36) (BHKSNO), Mech. Engr., Fletcher-Thompson, Inc., 211 State St., Bridgeport; *for mail,* 1166 Ogden St. Ext., Bridgeport 8, Conn.
Ip, Ching-U (J'43) (BCDJNZ), Indus. Engr., Giffels & Vallet, Inc., 1000 Marquette Bldg., Detroit, Mich.
Ipsen, Peter Grover (J'44) (SHN), Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1420 Keyes Ave., Schenectady 8, N.Y.
Ireland, Mark Lorin ('02; '13) (ZJMLCY), Exper. Engr., A.C. Spark Plug Div., Gen. Motors Corp., 1300 N. Dort Highway, Flint 2; *for mail,* 1704 Vernon Ave., Flint 6, Mich.
Ireland, Mark Lorin, Jr. ('28; '42) (SBJFKH), Engrg. Dept., Tech. Div., Newport News Shipbldg. & Dry Dock Co., 4101 Washington Ave., Newport News, Va.
Ireland, Robert W. (J'41) (BLM), 27 Baker Ave., Beverly, Mass.
Ireland, William F. A. (J'32) (JYNM), Engr., Tool Design, Columbian Rope Co., Auburn, N.Y.
Irey, George Woodbury ('30) (SFHKEP), Supt. of Plants & Prod., Empire Dist. Elec. Co., 602 Joplin St.; *for mail,* 515 Jackson, Joplin, Mo.
Irey, Glenn Reben (J'39) (SHEFNM), Mech. Engr., Signal Corps Engrg. Labs., Bradley Beach; *for mail,* 15 Hillsdale, Long Branch, N.J.
Irey, Laurence D. (J'42) (SZFEK), Test. Engr., Duquesne Works, Carnegie-Ill. Steel Corp.; *for mail,* 290 Utah Ave., Duquesne, Pa.
Irion, William ('26; '35) (SFEKAB), Ch. Engr., Geo. LaMonte & Son, 299 Kingsland Rd.; *for mail,* 686 Bloomfield Ave., Nutley 10, N.J.
Irish, Sumner Bartholf (J'43) (BLC), Ensign, U.S.N.R., 56th N.C.B., F.P.O., San Francisco, Calif.; *for mail,* Sigert Ave., Lexington, Ky.
Irmer, Charles Bernard ('19; '25; '35) (ZNJY BM), Engrg. Staff, Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44; *for mail,* Easton Rd. & Independence Ave., Roslyn, Pa.
Irons, Davison Ewing ('46) (CLM), Mech. Engr., Thompson Rd. Plant, Gen. Elec. Co., Syracuse; *for mail,* Piercefield Dr., Solvay, N.Y.
Irons, Milton Howard ('43), Research Engr., Gen. Fire Extinguisher Co., 260 W. Exchange St., Providence; *for mail,* 35 Allerton Ave., East Providence, R.I.
Irons, Oliver E., Jr. (J'40) (CJS), Design Draftsman, Moore Dry Dock Co., Foot of Adeline St., Oakland; *for mail,* Box 533, Route 2, Walnut Creek, Calif.
Irvine, James P. (J'40) (AKW), 95 W. Middle Turnpike, Manchester, Conn.
Irvine, John Alexander (J'43) (ACEFMS), Lt., U.S. Maritime Serv., Instr., N.Y. Officers Upgrade Sch., 39 Broadway, New York 6, N.Y.; *for mail,* 384 Woods Rd., Teaneck, N.J.
Irvine, John Woodward ('41) (NBDEOS), Asst. Plant Engr., Newport News Shipbldg. & Dry Dock Co., 4100 Washington Ave., Newport News, Va.
Irving, Henry Francis ('44) (LYHABN), Process Engr., Baker Perkins, Inc., 1000 Hess St., Saginaw, Mich.
Irving, Thomas Patrick ('42) (JMR), Advis. Mech. Com., 1308 Midland Bldg., Cleveland; *for mail,* 1667 Lincoln Ave., Lakewood 7, Ohio.
Irwin, Daniel B. ('42) (B), Madisonville, La.
Irwin, John Schwenk (J'41) (ZYKLCB), Mech. Engr., Warner Tool Design Corp., 522—5th Ave., New York 18; *for mail,* 120 Briggs Ave., Yonkers 2, N.Y.
Irwin, Joseph William ('40) (CMJNOS), Lt. Comdr., U.S.N.R.; *for mail,* 431 Thomas Ave., Riverton, N.J.
Irwin, Kilshaw McHenry ('17; '23; '30) (SFE), *Vice-President,* '39-'41, '41-'42; Mgr., Engrg. Dept., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Irwin, Paul Lawrence (J'22) (BRJNOZ), Engr. of Tests, Baldwin Loco. Works, Philadelphia 42; *for mail,* 100 Morton Ave., Ridley Park, Pa.
Irwin, Russell Douglass ('46) (BHZ), Flowmeter Engr., Brown Instrument Co., Wayne & Roberts Sts., Philadelphia 44; *for mail,* 512 Oreland Mill Rd., Oreland, Pa.
Irwin, Vincent H. ('21; '35) (EFS), Power Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail,* Devon, Pa.
Irwin, William (J'39) (SFC), Capt., Ord. Dept., Medical Field Serv. Sch., Carlisle Barracks, Pa.
Isabella, Bernard Joseph (J'43) (CAMRNJ), *Charles T. Main Award,* '42; Thompson Products, Inc., 23555 Euclid Ave., Euclid; *for mail,* 1681 Wood Rd., Cleveland Heights 21, Ohio.
Ischinger, Alfred Ernst ('27; '31) (BCEGJL), Pat. Atty., 803 Medical Arts Bldg., 230 N. 5th St., Reading, Pa.
Isely, Francis Dort (J'42) (BASEZF), Pvt., Army Air Forces, 4000th Base Unit, Sqd. I, Wright Field, Dayton, Ohio.
Isenberg, Martens H. ('18; '21; '35), V.P., Charge Mfr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York; *for mail,* 16 Strickland Pl., Manhasset, N.Y.
Isenberg, Paul Franklin (J'45) (SFMN), Steam Power Engr., D. M. Bare Paper Co.; *for mail,* 427 Walnut St., Roaring Spring, Pa.
Isham, Clarence Atwood (J'42) (FSE), Engr., N.Y. City Bd. of Education, New York; *for mail,* 63 Metropolitan Oval, Parkchester, Bronx 62, N.Y.
Isham, Homer L. (J'41) (KL), Engr., Shell Oil Co., Inc., Box 728, Wilmington; *for mail,* 9116 8th Ave., Inglewood, Calif.
Ishimura, Lyuho S. ('17; '21; '35) (C), Exec. Secy., Nippon Denchi Kabushiki Kaisha, Shinmachi-Imadegawa, Kamikyoku; *for mail,* Jodoji-Minamidacho, 154, Sakyoku, Kyoto, Japan.
Isidin, Benedict John ('31) (BENARJ), Asst. to V.P., Charge of Engrg., White Motor Co., 842 E. 79th St., Cleveland 1; *for mail,* 4065 Ellison Rd., South Euclid 21, Ohio.
Isles, Frederick Wilson ('21; '25) (LKFEDZ), Engr., Chem. Products Plant, Bayway Refinery, Stand. Oil Co. of N.J., Brunswick Ave., Linden; *for mail,* 10 Arlington Rd., Cranford, N.J.
Israelson, Arlo Floyd (J'44) (BGM), Mech. Engr., James M. Montgomery, Rm. 1107, 306 W. 3rd St., Los Angeles; *for mail,* 10005 Orange Ave., South Gate, Calif.
Ivers, Robert Carleton (J'43) (ASE), Lt., 28th Bomb. Sqd., 19th Bomb. Gp., A.P.O. 334, c/o Postmaster, San Francisco, Calif.
Iversen, Harold Walter (J'42) (HAKB), Asst. Prof., Dept. of Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.
Ives, Alver H., Jr. (J'41) (ABL), Draftsman, Glenn L. Martin Co., Baltimore; *for mail,* Haddon Ave., Govans Sta., Woodbrook, Baltimore 12, Md.
Ives, Charles Quincy ('40) (NLDYZW), Ives Lab., 132 Oak St.; *for mail,* Box 151, Reading, Mass.
Ives, Clifford Blair ('45) (ZLKHC), Ch. Engr., Kieley & Mueller, Inc., 2033—43rd St., North Bergen, N.J.
Ives, George Stoughton (J'40), Lt., U.S N.R., F.P.O., San Francisco, Calif.; *for mail,* 22 Windell Pl., Glenbrook, Conn.
Ivey, Carl Edward (J'43) (ERJWAC), Engr., Otis Engrg. Corp., 6612 Denton Dr., Dallas 1, Tex.
Ivins, Clinton Fox, Jr. (J'44) (MCF), Lt. (j.g.), U.S.N.R., U.S.S. *Hydrus,* AKA-28, F.P.O., New York, N.Y.; *for mail,* 937 Hillside Ave., Plainfield, N.J.

J

Jack, Carl Rebada ('36) (CBOMNR), 1713 W. 8th Ave., Eugene, Ore.
Jack, George ('26; '35) (DKL), V.P., J. M. Lehmann Co., Inc., 550 New York Ave., Lyndhurst, N.J.
Jack, William Albert (J'44) (BAMNKJ), Project Engr., Calif. Div., West. Gear Works, Box 192, Lynwood; *for mail,* 16956-B Brighton Ave., Gardena, Calif.
Jacka, Paul George (J'27) (CKLMHJ), Pres., Goslin-Birmingham Mfg. Co., Inc., 3700—10th Ave. N., Birmingham, Ala.
Jackel, Melvin Ely (J'43), Ensign; 3965 Sedgwick Ave., New York 63, N.Y.
Jacklin, H. M. ('17; '25) (EAOBZ), Consultant, Automotive & Mech. Engrg., 1436 W. 27th St., Indianapolis 8, Ind.
Jackling, Daniel Cowan (F'41) (CDJMRS), Firm of D. C. Jackling, Personal Investment Business, 1800 Hobart Bldg., San Francisco 4, Calif.
Jacklitch, John James, Jr. (J'40) (ABHK), Instr., Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6; *for mail,* 415 Dover Center, Bay Village, Ohio.
Jacks, Ivan Titus ('44) (CHSOEK), V.P., Gen. Mgr., Anonyne Hellenique Des Eauz Des Villes D'Athenes, Piree et Enviv., 4 Kolokotroni St., Athens, Greece.
Jacks, Wilbert Jacob (J'42), Ensign, U.S.N.R.; 714 Westgate Ave., University City, Mo.
Jackson, Albert Andrew ('21; '26; '35) (EO), Exchange Hotel, Franklin, Pa.
Jackson, Arthur C. ('03; '10) (JCADM), Owner, Jackson Associates, 152 N. 15th St., Philadelphia 2, Pa.
Jackson, Charles Arthur ('11) (HSO), Cons. Engr., 194 Godwin Ave., Ridgewood, N.J.
Jackson, Charles H. (J'40) (CJM), Asst. Radio Engr., Signal Corps, War Dept., Munitions Bldg., Washington, D.C.; *for mail,* 31 N. Granada, Arlington, Va.
Jackson, Dugald C. ('90; F'38; H'45) (OSRC), Prof. Emeritus Elec. Engrg., Mass. Inst. Tech., Cambridge 39; *for mail,* 5 Mercer Circle, Cambridge 38, Mass.
Jackson, Dugald C., Jr. ('23; '28) (CMJ), Col., 5 Mercer Circle, Cambridge 38, Mass.
Jackson, E. Edmund ('38), Mech. Inventory Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Jackson, Frederick Raymond ('17; '24; '28) (SH BFK), Mech. Engr., Duke Power Co., Power Bldg., Charlotte 1, N.C.
Jackson, Geo. P. ('21; '35), Ch. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 33-42—161st St., Flushing, N.Y.
Jackson, H. Olin ('23; '24) (TSMCD), Treas., Gen. Mgr., Great Falls Bleachery & Dye Works, Inc., Main St.; *for mail,* 39 Page St., Somersworth, N.H.
Jackson, Harry C. (J'38) (ACJNOT), Mech. Engr., Morton Mch. Works, 1718—3rd Ave., Columbus, Ga.
Jackson, Henry Wallace ('13; '26) (CJMDZ), Pres., Bearings Co. of Am., Harrisburg Ave.; *for mail,* 154 School Lane, Lancaster, Pa.
Jackson, James Alfred ('22; '35) (ACD), Engrg. Consultant, Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Jackson, James B. ('31; '35) (LOJ), Estimator, Chem. Constr. Corp., 350—5th Ave., New York 1; *for mail,* 34-24—82nd St., Jackson Heights, L.I., N.Y.
Jackson, Jesse Atwater (J'28) (CMDNWH), Ch. Engr., Smith Shipyards, Inc., Box 1751, Pensacola, Fla.
Jackson, John Kenneth ('39) (BCL), Ch. Engr., Corning Glass Works, 8th & McLean, Charleroi, Pa.
Jackson, John Price ('03; F'39) (CES), Engr., Indus. Adviser, J. C. McMurtrie, R.F.D. 1, Pittston, Pa.
Jackson, John Ralph ('15; '22) (RSF), Engr. of Tests, Mo. Pacific R.R., 3001 Chouteau Ave., St. Louis 3; *for mail,* 4392 Maryland Ave., St. Louis 8, Mo.
Jackson, John W. (J'40) (JBNAH), Assoc. Prof. Mech. Engrg., Univ. of Md., College Park, Md.
Jackson, Lawrence B. Wood ('38) (OCBKN), Cons. Engr., 301 Rule Bldg., Amarillo, Tex.
Jackson, Lawrence Bailey ('18) (ERF), Dir. of Engrg., Diesel Div., Am. Loco. Co., Schenectady 5, N.Y.
Jackson, Lewis R. ('33; '35) (FKS), Sales & Engrg., Boiler Dept., Henry Vogt Co.; *for mail,* 5612 Southland Blvd., Louisville, Ky.
Jackson, Marcus ('44) (AJR), Union of South Africa Govt. Supply Mission, 907—15th St., N.W., Washington 5, D.C.
Jackson, Robert Clemence (J'43) (ABFNTO), Project Engr., Ranger Aircraft Engs. Div., Fairchild Eng. & Airplane Corp., Farmingdale; *for mail,* Box 648, Huntington, L.I., N.Y.

Jackson, Robert L. (J'40) (JS), Turbine Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 226 Jackson Ave., Schenectady 4, N.Y.
Jackson, Tom Cleland (J'45) (AEBKFM), Mech. & Aero. Engr., Army Air Forces, Aero. Dept., Calif. Inst. of Tech., Pasadena, Calif; *for mail*, Spalding Ave., Lebanon, Ky.
Jackson, Thomas E. (J'37), Instr. in Mech. Engrg., Mech. Engrg. Dept., Lehigh Univ., Bethlehem, Pa.
Jackson, Thomas Woodrow (J'41), 1760 Walnut St., Berkeley 4, Calif.
Jackson, Warren Cowan (J'40) (JKS), Engr., Fed. Power Comm., 412 Neil P. Anderson Bldg., Ft. Worth 2; *for mail*, 2804 Westridge, Ft. Worth 7, Tex.
Jackson, Warren Earle (J'45) (JCBEFK), Ensign, U.S.N., U.S.S. *Albany*, Newport, R.I.; *for mail*, 75 E. Main St., Portland, Conn.
Jackson, William George ('42) (CDJKLS), Designer, Nichols Engrg. & Research Corp., 60 Wall Tower, New York 5, N.Y.; *for mail*, 63 Van Reypen St., Jersey City 6, N.J.
Jacob, Brent Cooke ('22) (BAHDYK), Engr., Indus. Brownhoist Corp., 135 Washington Ave.; *for mail*, 205 N. Mountain St., Bay City, Mich.
Jacobi, Edward N. ('19; '35) (BNMJKZ), Ch. Engr., Briggs & Stratton Corp., Milwaukee 1; *for mail*, 2128 E. Lafayette Pl., Milwaukee 2, Wis.
Jacobs, Don Henry (J'42), 1126 E. Moreland, Phoenix, Ariz.
Jacobs, Jay Allen ('27; '35) (DLOSFC), Dept. Head, Spec. Assignments, Cent. Engrg. Div., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City 2, N.J.
Jacobs, John J., Jr. (J'38) (SFE), Application Engr., Westinghouse Elec. Internatl. Co., 40 Wall St., New York 5, N.Y.
Jacobs, Leonard (J'43) (BJMNYZ), Mech. Design Engr., Radio Corp. of Am., Camden, N.J.; *for mail*, 5347 N. Camac St., Philadelphia 41, Pa.
Jacobs, Milton Russel (J'43) (AMJ), Project Engr., Thompson Products, Inc., 6402 Cedar Ave.; *for mail*, 1966 E. 71st St., Cleveland 3, Ohio.
Jacobs, Robert Morton (J'44) (AES), 1925 Bergen St., Brooklyn 33, N.Y.
Jacobs, Samuel Sydney ('23) (NCMJBD), Asst. to Mgr., Equip. Div., Am. Can Co., 230 Park Ave., New York 17, N.Y.
Jacobs, Ward S. ('97; '04) (KFAZND), Cons. Engr., Walton Co., Hartford 3; *for mail*, 70 Terry Rd., Hartford 5, Conn.
Jacobsen, Lydik S. ('42) (BAS), Exec. Head, Mech. Engrg. Dept., Sch. of Engrg., Stamford Univ., Stamford University, Calif.
Jacobson, Carl Arthur ('37) (ERAB), Cons. Engr., Loco. & Southwark Div., Baldwin Loco. Works, Eddystone; *for mail*, 112 Golf View Rd., Ardmore, Pa.
Jacobson, Chas. N. (J'39), Plant Mgr., Dake Engine Co., Grand Haven, Mich.
Jacobson, Conrad C. ('17), Ch. Engr., John Robertson Co., 133 Water St., Brooklyn, N.Y.; *for mail*, 27 Douglas Rd., Glen Ridge, N.J.
Jacobson, Eugene William ('34; '35) (EBNLJZ), Ch. Design Engr., Gulf Research & Devel. Co., Box 2038, Pittsburgh 30, Pa.
Jacobson, Frank Franz ('16; '26) (BCJKRZ), Pres., Seamlex Co., Inc., 27-27 Jackson Ave., Long Island City 1, N.Y.
Jacobson, Harvey (J'44) (BJM), Supt., Manne-Knowlton Insulation Co., 416 W. 13th St., New York 14, N.Y.; *for mail*, 7 Foster St., Newark 5, N.J.
Jacobson, Richard Fisher (J'45) (JFS), Staff Sergeant, U.S.A., 3rd Co., Trng. Gp., Armd. Sch., Ft. Knox, Ky.; *for mail*, 61 Stone Ave., Ossining, N.Y.
Jacobson, Saul Baum (J'34) (OSYT), Draftsman, E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington; *for mail*, Rm. 606, 11th & Washington Sts., Wilmington 9, Del.
Jacobus, David D. ('34) (ZJY), Staff Mem. Radiation Lab., Mass. Inst. of Tech.; *for mail*, 18 Farwell Pl., Cambridge, Mass.
Jacobus, David Schenck ('89; H'34), *Manager*, '00-'03, *Vice-President*, '03-'05, *President*, '16; Retired; 93 Harrison Ave., Montclair, N.J.
Jacobus, Robert Fulton ('07; '12) (OLYSMD), Partner, Francisco & Jacobus, 511—5th Ave., New York 17, N.Y.
Jacobus, William Wesley (J'45) (ACMDJ), Project Engr. Co-ordinator, Wright Aero. Corp., 132 Beckwith Ave., Paterson; *for mail*, 228 Midland Ave., Montclair, N.J.
Jacoby, Henry Edward ('16) (KL), 205 E. 42nd St., New York 17, N.Y.
Jacoby, Nicholas Peter (J'38) (FSZCLK), Field Engr., Hagan Corp., 225 S. 15th St., Philadelphia; *for mail*, Butler Pike, Three Tuns, Ambler, Pa.
Jacoby, Willis (J'41) (AF), 2601 N. Franklin St., Wilmington, Del.
Jaeger, Jerald Francis (J'45), Plastics Div., Continental Can Co., 725 N. 6th St., Cambridge, Ohio.
Jaffe, Bernard (J'39) (ALW), 1411 Benson St., Bronx, N.Y.
Jaffe, William Julian (J'41) (CDL), Instr., Math. Dept., Newark College of Engrg., 367 High St., Newark 2; *for mail*, 1030 Anderson Ave., Palisade, N.J.
Jagdmann, Edwin Fred (J'36) (NLDMCH), Devel. Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail*, 24 W. Red Bank Ave., Woodbury, N.J.
Jahncke, Donald Edward (J'39) (CDMZ), Plant Layout Engr., Plymouth Div., Chrysler Corp., 6334 Lynch Rd., Detroit 31; *for mail*, 1541 Hollywood Rd., Grosse Pointe Woods 30, Mich.
Jahncke, Paul Frederic ('44), Pres., Jahncke Serv., Inc, 814 Howard; *for mail*, 7 Audubon Pl., New Orleans, La.
Jahnke, Noble Walter (J'42) (CMNJ), Lt., U.S.N.R., Asst. Matl. Supt., Naval Shipyard, Boston; *for mail*, 56 Charlesgate, East Boston 15, Mass.
Jahnke, Ralph Harry (J'42) (ACD), 3446 S. Elmwood Ave., Berwyn, Ill.
Jaklitsch, Joseph John (J'41) (ZCM), Editorial Dept., A.S.M.E., 29 W. 39th St., New York 18; *for mail*, 42-14—64th St., Woodside, L.I., N.Y.
Jaklitsch, Louis Joseph (J'39) (MJCA), Process Engr., Sperry Gyroscope Co., Inc., 40 Flatbush Ave. Ext., Brooklyn; *for mail*, 70-16—65th Pl., Glendale 27, N.Y.
Jakob, Max ('37) (KBSOAZ), Research Prof. of Mech. Engrg., Ill. Inst. of Tech. & Armour Research Foundation, 3300 Federal St., Chicago 16; *for mail*, 5412 East View Park, Chicago 15, Ill.
Jakobsson, Gustav Herman ('07), Pat. Atty. & Translator, 228 Spruce Ave., Takoma Park, Washington 12, D.C.
James, Carroll D. (J'45), Ensign, 644 W. 7th St., Plainfield, N.J.
James, David T. (J'39), *Student Award*, '39; R.F.D. 1, Frederick, Md.
James, Hardy Monroe ('42) (J), Box 1296, Monroe, La.
James, Ivor G. (J'37), c/o Engrg. Dept., Sunderland Tech. College, Sunderland, Durham, England.
James, Jack Kenneth (J'43), Box 825, Albuquerque, New Mex.
James, John R. ('29) (SODFHJ), Sr. Engr., Engrg. Div., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
James, Lourimer Abner (J'45) (DCA), Sales Engr., Rudel Mack Co., 137 Wellington St. W., Toronto, Ont., Can.
James, Patrick Henry (J'38) (AZ), Lab. Chief, Robert H. Ray, Inc., 2535 Dunstan; *for mail*, 3614 Georgetown, Houston 5, Tex.
James, William Judson (J'42) (NCOAL), Ch. Engr., Positel Mfg. Co., Berkeley; *for mail*, 2500 Durant Ave., Berkeley 4, Calif.
James, William Stubbs ('43) (BFNKYJ), Dir., Automotive Research, Ford Motor Co., Dearborn, Mich.
Jameson, Albert Stapleton (J'41) (SKLZFB), Research & Devel. Engr., Elliott Co., Jeannette, Pa.
Jameson, Stanley L. ('31; '44) (OCLMK), Engr., Gen. Elec. Co., Schenectady; *for mail*, 402 Ballston Rd., Schenectady 2, N.Y.
Jamison, Charles Chambers, III (J'45), Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail*, 1320 Lincoln Ave., Prospect Park, Pa.
Jamison, George Stillwell ('18; '34) (CAH), Secy., Engr., Glens Falls Ins. Co., 191 Glen St., Glens Falls, N.Y.
Jamron, Richard (J'43) (NBMCAO), Engr., Eastman Kodak Co., Kodak Park, Rochester; *for mail*, 44 Sodus St., Rochester 9, N.Y.
Janas, Leo John ('37; '45) (CMO), Indus. Engr., Republic Steel Corp., 116th & Burley Sts., Chicago 17; *for mail*, 6911 S. Kimbark Ave., Chicago 37, Ill.
Janco, Nathan ('32; '45) (JCANMR), Pres., Centrifugal Casting Mch. Co., Box 947, Tulsa 1, Okla.
Janda, James Frank ('28) (DONYMB), Ch. Sales Engr., Barber-Greene Co.; *for mail*, 122 S. Harrison Ave., Aurora, Ill.
Janda, Rudolph William ('44) (BJL), Ch. Mech. Engr., Ceco Steel Products Corp., 5701 W. 26th St., Cicero; *for mail*, 6938 Riverside Dr., Berwyn, Ill.
Jandrisevits, Peter (J'33) (KLS), Mech. Engr., Merck & Co., Inc., Rahway; *for mail*, 155 Highland Ave., Metuchen, N.J.
Janes, Andrew Mead (J'45) (NJOEBC), Lt., Corps of Engrs., U.S.A., Engr. Officer Rehabilitation Program, 9834th Tech. Serv. Unit, Det. 4, Army Serv. Forces Trng. Center, Ft. Lewis, Wash.; *for mail*, 76 Christie Ave., River Edge, N.J.
Janicek, Joseph J. (J'39) (CMWAL), Container Corp. of Am., 404 E. North Water St., Chicago; *for mail*, 8014 S. Sangamon St., Chicago 20, Ill.
Janitsch, Anthony Duchesnay (J'42) (DCL), Engr., Stephens-Adamson Mfg. Co. of Can., Ltd., Belleville, Ont., Can.
Jannett, Anthony Verniqud, III (J'32) (CDLTY), Capt., Sanitary Corps, U.S.A., Medical Dept., Lawson Gen. Hospital, Atlanta, Ga.; *for mail*, 1533—41st St., Belview Heights, Birmingham 8, Ala.
Janousek, Joseph (J'26) (WMC), Dept. Ch. of Oper., West. Elec. Co., Inc., 6600 Metropolitan Ave., Middle Village; *for mail*, 46-02—25th Ave., Long Island City 3, N.Y.
Janson, Nils Frederick (J'46) (BCE), 532—5th Ave., Watervliet, N.Y.
Janssen, Earl (J'36), Box 4, Vernon Branch, Los Angeles, Calif.
Jansson, John H. ('24; '35) (SBOEZ), Supvg. Engr., Turbine Engrg. Dept., Gen. Elec. Co., 744 Broad St., Newark 2; *for mail*, 692 Larch Ave., Teaneck, N.J.
Jansson, Martin E. ('31; '35), Comdr., Civil Engr. Corps., U.S.N., Civil Engr. Corps Officers Sch., Davisville, R.I.
Jansson, Oscar E. (J'42) (WCY), Pres., Erik Jansson Co., 901-23 E. Girard Ave., Philadelphia 25, Pa.
Jantac, Arthur R. (J'41) (AER), Eng. Tester, Electro-Motive Corp., La Grange; *for mail*, 2240 S. 60th Court, Cicero, Ill.
Jantsch, Emil ('42) (SHK), Draftsman, Youngstown Dist., Carnegie-Ill. Steel Corp., 912 Salt Spring Rd.; *for mail*, 123 S. Lakeview Ave., Youngstown 9, Ohio.
Japikse, Bertrand ('30; '41) (HBAYWD), Asst. Mgr., Hyd. Dept., Birdsboro Steel Fdy. & Mch. Co., Birdsboro; *for mail*, 3400 Romig Ave., Reiffton, Pa.
Japp, Albert L. ('38) (CHK), Center Lane, Delmar, N.Y.
Jappe, Kurt Wade ('34) (CLM), *Treasurer*, '44 to date; Retired; 32 W. 40th St., New York 18, N.Y.
Jaqua, George Remig (J'38) (HSZAEL), Power Plant & Hyd. Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail*, 543 Park Ave., Paterson 4, N.J.
Jarcho, Ralph ('28; '35; '37) (CAEMLS), Pres., Sutter Constr. Co., Inc., 285 Madison Ave., New York; *for mail*, 99-59—65th Rd., Forest Hills, N.Y.
Jarchow, Werner Christian ('45), Mech. Supt., Gen. Steel Castings Corp., Granite City, Ill.; *for mail*, 336 Lookout Dr., Baden Sta., St. Louis 15, Mo.
Jardh, Wilhelm ('20; '35) (AH), Retired; 1047 W. 64th St., Los Angeles 44, Calif.
Jardine, Frank ('22) (ABCEFH), Mgr., Devel. Div., Cleveland Branch, Aluminum Co. of Am., 2210 Harvard Ave., Cleveland 5, Ohio.
Jardine, Thomas S. ('38) (EKMZDN), Plant Engr., United Rexall Drug Co. Ltd., 68 Broadview Ave., Toronto 8; *for mail*, 15 Newmarket Ave., Toronto 13, Ont., Can.
Jarema, John D. ('26; '41) (CAM), Project Engr., Firestone Tire & Rubber Co., Akron; *for mail*, 728 Chestnut Blvd., Cuyahoga Falls, Ohio.
Jarnagin, James Floyd ('24; '42) (CMDJY), Budget Dept., Dodge Plant, Chrysler Corp., 7900 Joseph Campau Ave., Detroit 10; *for mail*, 14475 Lappin, Detroit 5, Mich.
Jaros, Alfred L., Jr. ('22) (AES), Partner, Jaros, Baum & Bolles, 415 Lexington Ave., New York, N.Y.
Jarrell, Gordon James (J'39) (MCL), Plant Engr., Willard Storage Battery Co. of Can. Ltd., 269 Campbell Ave., Toronto 9, Ont., Can.
Jarvis, A. C. G. (J'46), Instrument Control Engr., Que. North Shore Paper Co., Ltd.; *for mail*, Baie Comeau, Saguenay Co., Que., Can.
Jaschka, John H. ('16), Dist. Sales Mgr., Natl. Malleable & Steel Castings Co., 411 N. 7th St., St. Louis 1, Mo.
Jasis, Peter Gregory (J'42) (BCMNSY), Engr., Am. Steel Fdys., Canal & Dock St., East Chicago; *for mail*, 7206 Grand Ave., Hammond, Ind.
Jaski, Frank Edward ('41) (HSBCO), Application Engr.; Hyd. Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 156 N. 87th St., Wauwatosa 13, Wis.
Jasper, Norman Hans (J'41) (JMY), 6210 Madawaska Rd., Washington 16, D.C.
Jasper, Thomas McLean ('26) (BJP), Gen. Am. Trans. Corp., 135 S. La Salle St., Chicago 90, Ill.
Jauss, August C. ('42) (BLM), 301 W. Providence Rd., Aldan, Delaware Co., Pa.
Jaxon, Leonard Henry (J'45) (CDO), Richardson Scale Co., Clifton, N.J.
Jeffcock, Howard Walter ('40; '45) (HMJ), Tool Engr., Morris Mch. Works, Baldwinsville, N.Y.
Jeffcott, Robt. C. ('12) Retired; Sunset Hill, Patagonia, Ariz.
Jefferds, Joseph Crosby, Jr. (J'40) (BC), Maj., Ord. Dept., Comd. & Gen. Staff Sch., Ft. Leavenworth, Kan.
Jefferies, Fred'k L. ('00; '05), Retired; 330 Blackstone Ave., La Grange, Ill.
Jeffers, Frederick James (J'39) (SEFHK), Testman, Westport Sta., Consltd. Gas, Elec. Light & Power Co., Baltimore 2; *for mail*, 225 Hawthorne Rd., Baltimore, Md.
Jeffords, Tom ('29; '35), Supt., Power Prod., Pub. Ltg. Comm., City of Detroit, 5425 W. Jefferson, Detroit 9, Mich.
Jeffrey, Robert Hutchins ('30) (DMBFKJ), Chmn. of Bd., Jeffrey Mfg. Co., E. 1st Ave., Columbus 16, Ohio.
Jeffries, Ernest ('23; '32) (BCJM), Asst. Ch. Engr., Aluminum Co. of Am., Cressona; *for mail*, 1946 Howard Ave., Pottsville, Pa.
Jeffries, William Raine (J'42) (MNJAYL), Electronic Technician's Mate, 2nd Class, U.S.N., Instr., Naval Trng. Sch., Del Monte, Calif.
Jeheber, Rodrique Albert ('27; '35) (HJEBDY), Research & Devel. Engr., United Engrg. & Fdy. Co., 1st Natl. Bank Bldg., Pittsburgh 22; *for mail*, 621 S. Lang Ave., Pittsburgh 8, Pa.
Jehle, Ferdinand ('13; '18; '26) (KBNSHE), Dir. of Engrg., Hoffman Specialty Co., 1001 York St., Indianapolis 7, Ind.

Jelinek, Donald Edwin (J'43) (AKSHER), Research Analyst, No. Am. Aviation, Inc., Inglewood; for mail, 2444 Crenshaw Blvd., Los Angeles 16, Calif.
Jelinek, Julius Joseph (J'39) (BJHC), Research Assoc., Univ. of Calif., Berkeley; for mail, 554—30th St., San Francisco, Calif.
Jelleme, William Oscar ('45), Dir. of Devel., Pacific Mills, 1 W. Central Ave., East Newark; for mail, Box 362, Short Hills, N.J.
Jelter, Earl (J'40) (AM), Prod. Engr., Lockheed Aircraft Corp., Burbank; for mail, 308 N. Central, Glendale 3, Calif.
Jenkins, Donald Ray (J'43) (AKOS), Pacific Coast Steam Engr., Westinghouse Elec. Corp., 420 S. San Pedro St., Los Angeles 13, Calif.
Jenkins, Francis Grey ('42) (CFJ), Surplus Property Buyer, Pur. Dept., Kodak Pk. Works, Eastman Kodak Co., Rochester 4, N.Y.
Jenkins, Harold Brewster ('13; '22; '35) (OSEAL), Ch. Mech. Engr., John W. Harris Associates, Inc., 30 Rockefeller Plaza, New York; for mail, 50 Rushmore Ave., Douglaston, L.I., N.Y.
Jenkins, James Gatewood (J'43) (SBEFL), Bartlett, N.H.
Jenkins, James Leon (J'43) (SRMA), Tool Designer, Hawk-Eye Works, Eastman Kodak Co., 1447 St. Paul St., Rochester 5, N.Y.
Jenkins, John Thomas, Jr. (J'43) (ACM), James Robertson Hotel, Nashville 3, Tenn.
Jenkins, Peter ('21) 110 Fowler Ave., Yonkers, N.Y.
Jenkins, Robert Wilkinson (J'44) (YNBEAH), Owner, Wilkinson Jenkins Engrg. & Sales Co., Box 506, Charleston; for mail, Yonges Island, S.C.
Jenkins, Schuyler VanRensselaer ('34; '38) (LZBNYK), Design Engr., E. I. du Pont de Nemours & Co.; for mail, 4018 DeVeaux St., Niagara Falls, N.Y.
Jenkins, Svend (J'30), 201 Dean St., Brooklyn, N.Y.
Jenks, Frank (J'41), Magnesium Reduction Co., Luckey, Ohio.
Jenks, Glen F. ('09; '14) (JYBALM), Col., U.S.A., Retired; 232 Avondale Ave., Los Angeles 24, Calif.
Jenks, Stephen Moore ('23; '34) (CJR), Gen. Supt., Carnegie-Ill. Steel Corp., 1 Broadway, Gary, Ind.
Jenney, Russell Herbert (J'45) (HPZ), Supt. Gas Measurement, Ky. W.Va., Gas Co.; for mail, 2600 Country Club Court, Ashland, Ky.
Jennings, Burgess H. ('26; '35; '35) (SEBK), Prof. & Chmn., Dept. of Mech. Engrg., Tech. Inst., Northwestern Univ., Evanston, Ill.
Jennings, Donn Ogden (J'41) (CBHS), Plant Engr., Container Corp. of Am., 404 E. North Water St., Chicago 11, Ill.
Jennings, Edward John (J'42) (HBM), Lt., U.S.N.R., Office of Asst. Indus. Mgr., U.S.N., Ferry Bldg., San Francisco, Calif.
Jennings, Francis H. (J'43) (SOEFZD), Designer, Gibbs & Hill, Inc., Pa. Sta., New York 1; for mail, 78 Mackey Ave., Port Washington, N.Y.
Jennings, Frederick Anderson (J'37) (EAH), Project Engr., Chrysler Corp., 12800 Oakland Ave., Highland Park; for mail, 1653 Taunton Rd., Birmingham, Mich.
Jennings, Irving C. ('08; '15; '18), Pres., Nash Engrg. Co., Wilson Rd., South Norwalk, Conn.
Jennings, James Morton, Jr. (J'44) (MKCES), Lt., U.S.N., Naval Air Sta., c/o Aerology, Bldg. 45, Pensacola, Fla.; for mail, 3155 State St., New Orleans 15, La.
Jennings, James Thomas, Jr. (J'41) (CJMBNO), Product Engr., Butler Mfg. Co., 13th & Eastern Ave., Kansas City 3, Mo.
Jennings, Lawrence David ('44) (JNMBC), Mech. Engr., Westinghouse Elec. Corp., 469 Sharpsville Ave.; for mail, 1009 George St., Sharon, Pa.
Jennings, Uel P. (J'37) (BSH), Mech. Devel. Engr., E. I. du Pont de Nemours & Co., Gibbstown; for mail, Box 494, Route 1, Sewell, N.J.
Jennings, William Arthur ('42) (SFE), Supt., Rowland Tompkins & Son, 420 Lexington Ave., New York 17, N.Y.
Jens, Arthur Henry ('35; '41) (COML), Ch. Engr., Springfield Fire & Mar. Ins. Co., 222 W. Adams St., Chicago 6, Ill.
Jensen, Charles Black ('44) (LSN), Mech. Engr., Armour & Co., Union Stock Yards, Chicago 9, Ill.
Jensen, Henry Hartwig (J'39) (DCMJ), Engr., Staff & Plant Engrg. Dept., Babcock & Wilcox, Stirling Ave.; for mail, 228 E. Ford Ave., Barberton, Ohio.
Jensen, Irving Clifford ('45) (SHZOBE), Mech. Engr., Pub. Works Dept., Naval Shipyard, Mare Island; for mail, Box 422, Route 1, Walnut Creek, Calif.
Jensen, J. O. ('33), 6345 University Ave., Chicago 37, Ill.
Jensen, James A. (18; '23; '35) (KM), Pres., Quaker City Iron Works, Aramingo Ave. & Ontario St., Philadelphia; for mail, 330 Quarry Lane, Haverford, Pa.
Jensen, James Allen (J'45) (EMP), Ensign, U.S.N., Colony Hotel, Ocean Dr., Miami Beach, Miami, Fla.
Jensen, Jens Scott ('20; '27) (CFS), Allube Corp., 928 Allen Ave., Glendale 1, Calif.
Jensen, Joseph W. (J'44) (NCBMZY), Asst. Ch. Engr., Victor Animatograph Corp., 527 W. 4th St.; for mail, 1022 W. 16th St., Davenport, Iowa.
Jensen, Marion Allison ('26) (CDEMSY), Gen. Mgr., Prod. & Engrg. Div., Vocam Batteries, Inc., Box 1124, Tampa 1; for mail, 4007 San Juan, Tampa 6, Fla.
Jensen, Ove (J'41) (KNFJ), Jr. Mech. Engr., F. L. Smidth & Co., 11 W. 42nd St., New York; for mail, 220-17—97th Ave., Queens Village 9, L.I., N.Y.
Jensen, Sigurd R. ('31) (EFHLOS), Cons. Engr., 53 W. Jackson Blvd., Chicago 4, Ill.
Jensen, Walter Edward (M'45) (NHO), Design Engr., Charge Mch. Design, Ill. Watch Case Co., 853 Dundee Ave., Elgin, Ill.
Jensen, William H. ('40) (SC), Ch. Engr., Calumet Sta., Commonwealth Edison Co., 3200 E. 100th St., Chicago 17, Ill.
Jenseth, Harold Conrad (J'41) (NBCG), Mech. Engr., Gen. Elec. Co., 1 River Rd., Schenectady; for mail, Box 1215, Scotia 2, N.Y.
Jeppson, Geo. N. ('13), Treas., V.P., Norton Co., 1 New Bond St., Worcester, Mass.
Jerauld, Covel T. (J'41), Lt., c/o Air Trans. Comd., Control Office, Grenier Field, Manchester, N.H.
Jergens, Andrew N. ('37) (BCJ), Pres., Andrew Jergens Co., 2535 Spring Grove Ave., Cincinnati 22, Ohio.
Jerger, Joseph J. ('37; '39) (ABHKCS), Ch. Aerodynamics Engr., Lark Div., Fairchild Eng. & Airplane Co., 184-10 Jamaica Ave., Jamaica; for mail, 8 Dover Pl., Hempstead, L.I., N.Y.
Jerkovich, Michael Thomas (J'37) (CMJLDB), Pvt., 1st Class, U.S.M.C.R., Signal Co., Hdq. Bn., 6th Mar. Div., F.P.O, San Francisco, Calif.
Jermy, Laurence Edmund ('27; '35) (NBYMJC), Editor, *Machine Design*, Penton Bldg., Cleveland 13, Ohio.
Jernberg, Evert Herman (J'40) (BSM), 3447 Polk, N.E., Minneapolis 13, Minn.
Jernstrom, Karl Walter (J'37) (CO), Mar. Engr., with George G. Sharp, Naval Arch., 30 Church St., New York, N.Y.; for mail, 51 Oakridge Rd., Verona, N.J.
Jervis, Thomas John ('39) (BAEFJ), Gen. Mgr., F. I. Isotta Fraschini, Via Monterosa, Milan, Italy; for mail, 142 Rynda Rd., South Orange, N.J.
Jesatko, John M. (J'40) (BKS), Asst. Engr., Locke Insulator Corp., Charles & Cromwell Sts., Baltimore 3; for mail, 825 N. Collington Ave., Baltimore 5, Md.
Jessen, Preben ('42) (CDMO), Owner, Preben Jessen Co., Chrysler Bldg., New York 17, N.Y.; for mail, 330—76th Rear House, North Bergen, N.J.
Jessup, Albert Hall ('41) (DH), Ch. Designing Engr., Leeds, Hill, Barnard & Jewett, 1000 Edison Bldg., Los Angeles 13; for mail, 224 N. Segovia Ave., San Gabriel, Calif.
Jett, Carter Coleman ('02; '26) (BG), Prof. Mch. Design, Univ. of Ky., Lexington 29, Ky.
Jett, George C. ('25), Retired; 2629 N. Summit Ave., Milwaukee 11, Wis.
Jett, Lowell F. (J'39), Comdr., U.S.N.R., U.S.S. *Los Angeles*, F.P.O., San Francisco, Calif.
Jetter, William ('22) (DHJ), Field Engr., SKF Industries, Inc., 1242 W. Washington Blvd., Chicago 7; for mail, 930 Bonnie Brae, River Forest, Ill.
Jewett, Arthur Crawford ('09) (MCG), U.S. Office of Education, 26th & Water Sts., N.W., Washington 25; for mail, 532—20th St., N.W., Washington 6, D.C.
Jewett, Francis B. ('25; '35), 1165 Hyde Park Ave., Boston 36, Mass.
Jewett, Frank B., Jr. (J'38), 54 Longfellow Rd., Wellesley Hills, Mass.
Jewett, George Lester ('34; 43) (SKBH), Engr., Cent. Tech. Dept., Shipbldg. Div., Bethlehem Steel Co., Quincy 69; for mail, 29 Lincoln St., Hingham, Mass.
Jewson, Herbert Frederick, Jr. (J'38) (EFZKJ), Gas Engr., Philadelphia Elec. Co., West Conshohocken; for mail, 665 E. Rector St., Philadelphia 28, Pa.
Jezierski, Walter Frank (J'46) (KOS), Lt., U.S.N.; 89-06—138th St., Jamaica, N.Y.
Jimerson, Francis Arthur ('20; '35) (JN), Ch. Engr., Pneumatic Tool Div., Ingersoll-Rand Co., 101 N. Main St., Athens, Pa.
Joa, Curt G. ('35; '35) (BCJMWT), Pres., Gen. Mgr., Curt G. Joa, Inc., Sheboygan Falls, Wis.
Jobes, Harry W. (J'41) (ABR), Capt., Army Air Forces, Matériel Center, Rochester Area Office, Rochester 7; for mail, 111-45 Farmers Ave., Hollis, N.Y.
Joblove, Louis Julius (J'44) (NAME), Pvt., 1st Class, U.S.M.C., Mar. Serv. Sqd, Mar. Aircraft Gp. 25, F.P.O., San Francisco, Calif.; for mail, 1829—63rd St., Brooklyn 4, N.Y.
Jobst, Conrad ('42) (CJKNZ), Ch. Engr., Owens Brush Co., 901 Buckingham; for mail, 2356 Torrey Hill Dr., Toledo, Ohio.
Joeckel, Stanley V. (J'41) (BCJ), Devel. Dept., Wallace & Tiernan Co., Inc., 11 Mill St., Belleville, N.J.
Joerger, C. Albert ('22; '35) (SFE), Prof. Mech. Engrg., Univ. of Cincinnati, Cincinnati 21, Ohio.
Joers, Joseph Edward (J'44) (NEBOCK), Lt., U.S.N.R., Navy Dept., Washington; for mail, 536 Peabody St., N.W., Washington 11, D.C.
Johannes, Edward Gent (J'43) (NZMC), Prod. Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Johannsen, John Frederick ('42) (CDMO), Mgr., Export Div., Hyster Co., 2902 N.E. Clackamas St., Portland 8; for mail, 2916 N.E. 21st Ave., Portland 12, Ore.
Johansen, Henry (J'37), 3711 Bonsall Ave., Drexel Hill, Pa.
Johler, Robert Andrew (J'42) (LOJMC), Project Engr., E. I. du Pont de Nemours & Co., 3500 Grays Ferry, Philadelphia 46; for mail, 5124 Cedar Ave., Philadelphia 43, Pa.
John, Alexander, Jr. (J'37) (CMNJ), Asst. Gen. Mgr., Thomas C. Wilson, Inc., 2111—44th Ave., Long Island City 1, N.Y.; for mail, 523—1st St., Palisades Park, N.J.
John, Carl Frank ('45) (SFCEKO), Asst. to V.P., Charge of Power, Wis. Elec. Power Co., 231 W. Michigan St., Milwaukee 1; for mail, 3284 N. Summit Ave., Milwaukee 11, Wis.
John, Edw. T. ('21; '30) (CJW), Supt., J. R. Clark Co., 2nd Ave. N., & Aldrich St.; for mail, 5050 Garfield Ave. S., Minneapolis, Minn.
Johns, Cyrus N. ('27) (CDJ), V.P., Am. Chain & Cable Co., Inc., 230 Park Ave., New York, N.Y.
Johns, Henry Bennett ('43) (CZBM), Research Engr., Stevens Inst. of Tech., Hoboken, N.J.; for mail, 34 Parkview Pl., Baldwin, N.Y.
Johns, Robert Wythes (J'44) (ACMYD), Departmental Asst., Indus. Engrg. Sec., Pan-American World Airways, Box 3311, Miami 31, Fla.
Johns, Willard Lloyd (J'39) (L), Indus. Engr., E. I. du Pont de Nemours & Co.; for mail, Box 73, Parlin, N.J.
Johnsen, Bjornulf ('22; '35) (NMABYL), Goss & Johnsen, 71 Murray St., New York 7, N.Y.
Johnson, A. E. Fredrick ('44) (CMNDJA), V.P., Charge of Prod., Chicago Metal Hose Corp., 1315 S. 3rd Ave., Maywood; for mail, 134 S. Martha St., Lombard, Ill.
Johnson, Albert Myron ('22) (CM), Pres., Barnes Drill Co., 814 Chestnut St.; for mail, 1922 Clinton St., Rockford, Ill.
Johnson, Albert Pemberton (J'32) (C), 1625 Perry St., Columbus, Ohio.
Johnson, Allen J. ('35) (EFKSR), Mech. Engr., Anthracite Inst., 101 Park Ave., New York 17, N.Y.; for mail, 344 Congress Ave., Lansdowne, Pa.
Johnson, Allen Parke, Jr. (J'39) (CDNM), V.P., Charge of Engrg. & Prod., Chase Fdy. & Mfg. Co., 2300 Parsons Ave., Columbus 7, Ohio.
Johnson, Andrew Ingwer (J'46), 645 Locust Pl., Clinton, Iowa.
Johnson, Arthur Edward ('90; '92) (AD), Retired; 2949 Mills Ave., N.E., Washington 18, D.C.
Johnson, Arthur Frederick ('24) (EMHJD), Prof. Mech. Engrg., Sch. of Engrg., George Washington Univ., Washington, D.C.; for mail, 1208 Oakcrest Rd., Arlington, Va.
Johnson, Ashmore Clark (J'16) (CLKJ), V.P., Downingtown Iron Works, Inc., Wallace Ave., Downingtown, Pa.
Johnson, B. E. (J'19), Family Theatre, Rockford, Ill.
Johnson, Boyd Milo (J'22) (FKLS), Chief, Engrg. Dept., Carborundum Co., Perth Amboy, N.J.
Johnson, Bradford Joseph ('40; '45) (CDMJO), Mgr., Salvage & Reclamation Div., Reynolds Metals Co., 3rd & Grace St., Richmond 19, Va.
Johnson, Bradley S. ('09; '13), Sales Engr., W. H. Miner, Inc., 667 The Rookery, Chicago, Ill.
Johnson, Carl Edmund ('40; '45) (JBFCMR), Product Engr., Scaife Co., Oakmont; for mail, Box 242, R.D. 2, Verona, Pa.
Johnson, Charles Lewis (J'30) (CMYHLG), Treas. Shop Mgr., Charleston Elec. Supply Co., 914 Kanawha Blvd. E., Charleston 29, W.Va.
Johnson, Charles Thomas (J'45), York Corp.; for mail, Y.M.C.A., York, Pa.
Johnson, Charles Wesley (J'30) (CMNB), Partner, Mgr., Amarillo Welding & Mch. Works, 217 N. Polk St., Amarillo, Tex.
Johnson, Charles William (J'46), 2926 N. 77th Court, Chicago 35, Ill.
Johnson, Clarence Gardiner ('18) (O), Cons. Engr., 39 S. La Salle St., Chicago; for mail, 202 S. Madison Ave., La Grange, Ill.
Johnson, Clinton Victor ('42) (MBNALC), Mgr., Engrg. & Sales, Johnson Gage Co., 534 Cottage Grove Rd., Bloomfield, Conn.
Johnson, Daniel Calvin (J'44) (AH), Design Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; for mail, 403 N. Monroe St., Media, Pa.
Johnson, Darrel Vernon (J'40) (AENFBJ), 1st Pilot, Capt., Internatl. Div., Transcontinental & West. Air, Inc., 10 Richards Rd., Kansas City, Mo.; for mail, 515 N. Oak St., Falls Church, Va.
Johnson, Donald Seiz (J'43) (MACZBN), Pvt., 1st Class, A.U.S., Hdq. & Hdq. Det., 1325th Serv. Comd. Unit, Sta. Complement, Indiantown Gap Military Reservation, Indiantown Gap; for mail, 44—6th Ave., Collegeville, Pa.
Johnson, Douglas O. ('45) (BEHOZJ), Partner, Johnson-Fagg Engrg. Co., 313 Thompson Bldg., Tulsa 1; for mail, 1630 E. 35th St., Tulsa 5, Okla.
Johnson, E. Sigurd (J'43) (CMWADY), Cons. Indus. Engr., 211 Westminster Ave., Syracuse 10, N.Y.
Johnson, Edward Andrew (J'40) (ZLCYSH), Sales Engr., Proportioneers, Inc., 9 Codding St., Providence 1; for mail, 86 Princeton Ave., Providence 7, R.I.

Johnson, Einar John (J'45) (RSW), 7143 S. Vernon Ave., Chicago 19, Ill.
Johnson, Elmer Gardner (ABKCN), Chief, Spec. Devel. Unit, Wind Tunnel Branch, Army Air Forces, Wright Field, Dayton; *for mail,* 54 N. Grand Ave., Osborn, Ohio.
Johnson, Eric Goodwin ('21) (JMY), Head, Mch. Dept., Sch. of Indus. Arts, W. State St., Trenton 8, N.J.
Johnson, Ernest Clayton ('38) (ORSCKF), Exec. V.P., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.
Johnson, Floyd Frederick (A'45) (JKSFE), Asst. Boiler Safety Engr., Div. of Indus. Safety, Dept. of Indus. Relations, State of Calif., 309 Balboa Bldg., San Francisco 5; *for mail,* 1805 Francisco St., Berkeley 3, Calif.
Johnson, Francis E. ('16), V.P., M. W. Kellogg Co., 225 Broadway, New York, N.Y.
Johnson, Fred Ivar (J'43) (CMN), Indus. Engr., No. Furniture Co.; *for mail,* 1328 Lincoln Ave., Sheboygan, Wis.
Johnson, Fred Richard (J'45) (BCW), 47 Gladys St. (Tiorunda Sec.), Buffalo 23, N.Y.
Johnson, Fred'k M. (J'38) (BJM), Sales Engr., W. F. & John Barnes Co., 301 Water St.; *for mail,* 613 Summit St., Rockford, Ill.
Johnson, Fred'k P. (J'38), 28 Woodbine Ave., New Rochelle, N.Y.
Johnson, George Albert (J'44) (CJKLOS), Sr. Mech. Engr., Smith-Hinchman & Gryll, Inc., 800 Marquette Bldg., Detroit; *for mail,* 7639 Kendall Ave., Dearborn, Mich.
Johnson, George Arthur ('15) (RCM), Exec. V.P., W. H. Miner, Inc., 667—209 S. La Salle St., Chicago 4, Ill.
Johnson, George Dugan ('44) (SHOC), Hyd. Engr., S. Morgan Smith Co., York, Pa.
Johnson, George Tewksbury ('14; '22) (RFCJ SD), Pres., Gen. Mgr., Buckeye Steel Castings Co., 2211 Parsons Ave., Columbus 7, Ohio.
Johnson, Hamilton ('27), Prof. Mech. Engrg., La. State Univ., Baton Rouge, La.
Johnson, Harley A. ('34) (CMR), Gen. Mgr. for Trustees, Chicago Rapid Transit Co., 72 W. Adams St., Chicago; *for mail,* 322 S. Oak Park Ave., Oak Park, Ill.
Johnson, Harold Alexander (J'44) (SKHEFA), Asst. Prof. Mech. Engrg., Univ. of Calif., Berkeley 4; *for mail,* 5140 Proctor Ave., Oakland 11, Calif.
Johnson, Harold K. ('44) (BNMC), Asst. Tool Engr., Chapman Valve Mfg. Co., Pinevale St., Indian Orchard; *for mail,* 33 Wellesley St., Springfield 9, Mass.
Johnson, Harold Thurston (J'43) (ABCENJ), Stress Analyst, Consltd. Vultee Aircraft Corp., Ft. Worth; *for mail,* 1732 Ashland St., Ft. Worth 7, Tex.
Johnson, Herbert C. (J'41), 821 W. 16th St., Sioux Falls, S.D.
Johnson, Herman Henry ('26; '33; '35) (SFC), Asst. Engr., Mech. Engrg. Dept., Boiler Div., Consltd. Edison Co. of New York, Inc., 4 Irving Pl., New York 3; *for mail,* 730 Riverside Dr., New York 31, N.Y.
Johnson, Horace Adam ('38) (SCFNO), Asst. V.P., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.; *for mail,* 123 Howland Ave., West Englewood, N.J.
Johnson, Hugo Ellwood ('45) (EOS), Sales Engr., Edward Valve & Mfg. Co., Inc., 1200 W. 145th St., East Chicago, Ind.; *for mail,* 6816 Oglesby Ave., Chicago 49, Ill.
Johnson, Ira O., Jr. (J'40) (ZAMKBN), Engr., Fulton Sylphon Co., Kingston Pike, Knoxville, Tenn.
Johnson, Ivar Conrad (J'44) (AJW), 125—4th Ave., Cochrane, Ont., Can.
Johnson, J. Adolph ('16) (AEM), Factory Mgr., Bade-Cummins Mfg. Co., 3401 Jewel St.; *for mail,* 4517 Western Pkwy., Louisville 12, Ky.
Johnson, J. Marshall ('39) (SHFCKE), Asst. Prod. Engr., Tenn. Valley Authority, Power Bldg., Chattanooga; *for mail,* 1108 Hanover, Chattanooga 5, Tenn.
Johnson, James Albert (J'44) (BCMDNY), Prod. Engr., Elgin Natl. Watch Co., 107 National St.; *for mail,* 361 Elm St., Elgin 1, Ill.
Johnson, James Gaston (J'44) (AOR), 2nd Lt., U.S.M.C.R., Co. D, Replacement Bn., Transit Center, Mar. Aircraft, Fleet Marine Force, Pacific, F.P.O., San Francisco, Calif.; *for mail,* Box 84, R.D. 1, Rocky Mount, N.C.
Johnson, James W. ('17; '25), Pres., Johnson, Inc., 95 Liberty St., New York, N.Y.
Johnson, Jerome Allen ('45) (DH), Fire Protection Engr., Factory Ins. Assn., 555 Asylum St., Hartford, Conn.; *for mail,* 290 W. 11th St., New York, N.Y.
Johnson, John Algot (J'43) (AEKGBC), Exec. Engr., Bendix Helicopter Inc., 50 Rockefeller Plaza, New York, N.Y.; *for mail,* 2 Cove View Dr., Stamford, Conn.
Johnson, John Eskil (A'43), Asst. Dir., Pub. Relations, Gen. Motors Corp., 2044 W. Grand Blvd., Detroit 2, Mich.
Johnson, John Morrison (J'45) (BES), 6 E. Newfield Way, Cynwyd, Montgomery Co., Pa.
Johnson, Joseph Benjamin ('20) (SFELKH), Ch. Mech. Engr., Pub. Serv. Dept., City of Burbank, 174 W. Magnolia Blvd., Burbank, Calif.
Johnson, Joseph Blaine ('18; '24; '28) (CNM AJ), V.P., Gen. Mgr., Bryant Chucking Grinder Co., 257 Clinton St., Springfield, Vt.
Johnson, Joseph Edward (A'45) (FG), Dist. Mgr., Peabody Coal Co., 17 Battery Pl.; *for mail,* 230 Riverside Dr., New York, N.Y.
Johnson, Joseph Emery, Jr. (J'44) (HJEBNO), Sales Engr., Tex. Automatic Sprinkler Co., 2330 Summer St., Dallas; *for mail,* 3428 Stanford St., Dallas 5, Tex.
Johnson, Karl Einar (J'42) (SMCAK), Mgr., Charge Sales & Engrg., Rm. 200, J. F. Pritchard & Co., 205 W. Wacker Dr., Chicago 6, Ill.
Johnson, Kurth Hilding (J'41) (CDGLMZ), Supt. Emulsion Mfg., Photo Prod. Dept., E. I. du Pont de Nemours & Co.; *for mail,* du Pont Club, Parlin, N.J.
Johnson, Lawrence Victor (J'44) (AE), Pvt., 1st Class, Co. A, 1298th Engr. Combat Bn., A.P.O. 929, c/o Postmaster, San Francisco, Calif.
Johnson, Leon Henry ('39) (CMJL), Ch. Engr., Struthers-Wells Corp.; *for mail,* 500—5th Ave., Warren, Pa.
Johnson, Lester William (J'41) (HE), Ensign, O-V(S), U.S.N.R., Naval Torpedo Sta., Newport, R.I.; *for mail,* 597 Belvidere Ave., Phillipsburg, N.J.
Johnson, Lloyd M. ('33) (CS), Comm. of Sts. & Electricity, City of Chicago, 710 City Hall, Chicago, Ill.
Johnson, Lowell Curtis (J'42) (CMAF), Box 254, North Bend, Neb.
Johnson, Martin M. ('22; '38) (BKS), Hotel St. George, Brooklyn, N.Y.
Johnson, Nathan Clarke ('42), 260 Glennwood Rd., Englewood, N.J.
Johnson, Owen Champlin (J'42) (ACG), Tool Designer, Northrop Aircraft, Inc., Hawthorne; *for mail,* 3305-D W. 83rd St., Inglewood, Calif.
Johnson, Paul A., Jr. (J'46), Dake Eng. Co., Grand Haven, Mich.
Johnson, Paul Emmer (J'44) (AES), Specifications Engr., Carnegie-Ill. Steel Corp., N. Broadway; *for mail,* 550 Jackson St., Gary, Ind.
Johnson, Philip A. (A'30), Treas., Hampton Co., Easthampton, Mass.; *for mail,* Norwichtown, Conn.
Johnson, Ralph Paine ('39) (RKFBEJ), Ch. Engr., Baldwin Loco. Works, Philadelphia 42, Pa.
Johnson, Rawleigh Mac ('44) (SNOEZ), Engr., Charge Engrg. Test Dept., Ingersoll-Rand Co., Phillipsburg, N.J.; *for mail,* 381 Shawnee Dr., Easton, Pa.
Johnson, Raymond Carldon ('44) (BDFKS), V.P., Charge of Research, Anthracite Inst., Primos, Del. Co., Pa.
Johnson, Richard Paul ('45) (CDL), 539 Sanford Ave., Newark, N.J.
Johnson, Robert Howard (J'44) (ABHEGS), Engr., Research Lab., Gen. Elec. Co., 1 River Rd.; *for mail,* 8 Washington Ave., Schenectady 5, N.Y.
Johnson, Robt. L. ('39), Robt. L. Johnson Co., 328 Monadnock Bldg., San Francisco; *for mail,* 22 Sonia St., Oakland, Calif.
Johnson, Robert Lawrence (J'42) (EAKJBD), Asst. Mech. Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Municipal Airport, Cleveland; *for mail,* 1360 West Clifton, Lakewood, Ohio.
Johnson, Robert Marro (J'42) (JBM), 14343 Forrer, Detroit, Mich.
Johnson, Robert Vernon (J'41) (MNCJHA), 9346 S. Leavitt St., Chicago 20, Ill.
Johnson, Roy Edwin ('25; '28; '34) (BCL), V.P., Arenco Mch. Co., Inc., 25 W. 43rd St., New York, N.Y.
Johnson, Roy Ernest (J'40) (KLS), Plant Engr., Albemarle Paper Mfg. Co., Richmond; *for mail,* 3905 Decatur St., Richmond 24, Va.
Johnson, Roy F. ('41) (RBJWAC), Research Engr., Pullman Standard Car Mfg. Co., 79 E. Adams St., Chicago 3, Ill.
Johnson, Ruben Emmons ('32) (EKS), Head Mar. Engr., Code 260, Design Branch, Puget Sound Naval Shipyard, Bremerton, Wash.
Johnson, Russell Arthur (J'41) (OCMA), Engr., Chicago Branch, Frigidaire Sales Corp., 661 N. La Salle St., Chicago 10; *for mail,* 2950 Des Plaines Ave., Riverside, Ill.
Johnson, Samuel E., Jr. (J'38), Ensign, U.S.N.R.; 603 Dreams Landing, Annapolis, Md.
Johnson, Sidney, Jr. (J'46), Engr., Charge of Spec. Projects, Barre Wool Combing Co., Ltd., South Barre, Mass.
Johnson, Sigurd Olaf (J'42) (KSAEHB), Mech. Engr., Buffalo Forge Co., 490 Broadway, Buffalo; *for mail,* 267 Elmwood Ave., Buffalo 9, N.Y.
Johnson, Sigurd T. ('44) (BCDKSZ), Plant Engr., Am. Smelting & Refining Co., Garfield, Utah.
Johnson, Theodore Somers (J'38) (AE), Socony-Vacuum Oil Co., Inc., 62, Sharia Ibrahim Pasha, Cairo, Egypt.
Johnson, Theodore Woolsey ('00; '13) (GABNS), Capt., U.S.N., Retired; 11 Acton Pl., Annapolis, Md.
Johnson, Theodore Woolsey, Jr. (J'30) (AJO HM), Lt. Comdr., U.S.N.R., Air Maint. Officer, U.S.S. *Franklin,* c/o Postmaster, New York, N.Y.; *for mail,* 102 Severn Ave., Wardour, Annapolis, Md.
Johnson, Thomas L. (J'42) (COMAY), Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 963 Cleveland Ave., Schenectady 6, N.Y.
Johnson, Victor Emanuel ('44) (SFHK), Application Engr., Westinghouse Elec. Corp., 538 N. Broadway, Milwaukee 2; *for mail,* 4101 N. Woodburn St., Milwaukee 11, Wis.
Johnson, Vincent A. (J'37) (MYZJ), Design Engr., Maico Co., 21 N. 3rd St., Minneapolis 1; *for mail,* 3246 Buchanan St., N.E., Minneapolis 13, Minn.
Johnson, Waldon Tulien (J'42) (AJE), Ensign, Naval Ord. Lab. Exper. Facility, Hiwassee Dam, N.C.
Johnson, Wallace A. (J'41) (CMLDEA), Factory Engr., East Cleveland Lamp Works, Gen. Elec. Co., Nela Park, Cleveland 12, Ohio.
Johnson, Walter A. ('35; '45) (NMOBJR), Assoc. Prof. Mch. Design, Cornell Univ., 303 E. Sibley, Ithaca, N.Y.
Johnson, Warren (A'04), Chmn. of Bd., Todd-Johnson Dry Docks, Inc., Box 239, New Orleans, La.
Johnson, Wayne G. (J'36) (CO), Sr. Staff Engr., Fry, Lawson & Co., 135 S. La Salle St., Chicago 3, Ill.
Johnson, Wilfrid E. ('30; '34; '35) (ACBMRE), *Pi Tau Sigma Medallist,* '38; Mgr. Engrg. Div., Gen. Elec. Co., 5 Lawrence St., Blomfield; *for mail,* 48 Baker Rd., Livingston, N.J.
Johnson, William Frank (J'42), Sales Engr., Harper-Wyman Co., 8562 Vincennes, Chicago 20, Ill.; *for mail,* 326 Taft St., Gary, Ind.
Johnson, William Frederick (J'41), 2nd Lt., O.S.B.C. 25, Field Arty. Sch., Ft. Sill, Okla.
Johnson, William Henry ('20; '35) (EFK), Asst. Indus. Engr., Mich. Consltd. Gas Co., 415 Clifford St., Detroit 26; *for mail,* 13340 Coyle Ave., Detroit 27, Mich.
Johnson, Wistar W. ('32) (FKS), Engr., Gen. Elec. Co.; *for mail,* 12 Concord St., Lynn, Mass.
Johnston, Elmer (J'40) (ACG), Maj., 2nd Replacement Regt., Army Ground Forces, Replacement Depot 2, Ft. Ord, Calif.
Johnston, Carlton Halley ('44) (HL), Hyd. Engr., Bushnell Controls & Equip. Co., 117 W. 9th St., Los Angeles 15; *for mail,* 705 Winthrop Rd., San Marino 9, Calif.
Johnston, Clarence Nettleton ('40) (HK), Assoc. Irrigation Engr., Asst. Prof. of Engrg., Irrigation Div., College of Agric., Univ. of Calif.; *for mail,* 42 College Park, Davis, Calif.
Johnston, D. Bruce ('43) (NJBDHO), Partner, McPhee & Johnston, 90 West St., New York 6, N.Y.
Johnston, George X. ('27; '35) (EB), 402 Brown St., St. Clair, Mich.
Johnston, James Ambler ('07; '21) (O), Carneal, Johnston & Wright, 1000 Atlantic Life Bldg., Richmond 19, Va.
Johnston, Kenneth Mackenzie ('30) (CDFLS), Managing Dir., Reynolds Sons & Partners (Pty.) Ltd., Box 4561, Johannesburg, S. Africa.
Johnston, Paul K. ('22; '26), Head, Dept. of Physics & Elec., Ohio Mechanics Inst., Walnut St. & Central Pkwy., Cincinnati, Ohio.
Johnston, R. M. (J'39), Lt. (j.g.), U.S.N.R.; 192 Duke of Gloucester St., Annapolis, Md.
Johnston, Robert Smith ('21) (BJORL), 6 Alton Rd., Delarue Manor, Yardley, Pa.
Johnston, William Alexander (J'43) (BJZ), Mech. Engr., Mar. Div., Bendix Aviation Corp., 1 Hanson Pl., Brooklyn, N.Y.; *for mail,* 1304—73rd St., North Bergen, N.J.
Johnston, William Hartshorne (J'44), 1637 S. Jefferson Davis Pkwy., New Orleans, La.
Johnston, William Saunderson ('44) (ACMG), Firm of W. S. Johnston, 222 N. Hermitage Ave., Trenton 8, N.J.
Johnston, William Stanley ('23; '26) (SKF), Sanderson & Porter, 52 William St., New York 5, N.Y.
Jolly, Thomas D. ('28) (BCDMNO), V.P., Charge of Engrg. & Purchases, Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19, Pa.
Jones, A. Darby (J'41), 1002 Guilford Ave., Greensboro, N.C.
Jones, Albert I. ('92) (CDM), Factory Mgr., Pfaudler Co., 89 East Ave.; *for mail,* 369 Maplewood Ave., Rochester, N.Y.
Jones, Alfred ('18) (MHNDY), Retired; 213 N. West End Ave., Lancaster, Pa.
Jones, Allan Livingston (J'43) (NMEC), Ship Repair Officer, U.S.N., Box 71, Route 1, Orange, Calif.
Jones, Allan Theodore (J'43) (DFHKNZ), Ch. Engr., S.A. Armstrong Ltd., 115 duPont St., Toronto, Ont., Can.
Jones, Alton DuBois (J'41) (AJM), 1st Lt., Engrg. Officer, Army Air Forces, Overseas; *for mail,* 19 Electric Ave., West Somerville 44, Mass.
Jones, Bassett ('42), Retired; Nantucket Island, Mass.
Jones, Carl Lee (J'37) (OCDEAS), 508 E. Central Ave., Valdosta, Ga.
Jones, Carlton W. ('42), Manchester, Md.
Jones, Cecil Edmund ('46) (BHJ), Designing Engr., Shovel & Crane Div., Lima Loco. Works; *for mail,* 417 Mobel Ave., Lima, Ohio.
Jones, Charles Coleman ('26) (CEOZ), Supt., Engrg. Div., Philadelphia Gas Works Co., 1800 N. 9th St., Philadelphia 22; *for mail,* 226 E. Highland Ave., Philadelphia 18, Pa.
Jones, David J. ('18; '30) (DFSRHB), Mech. Asst. Ill. Cent. System, 135 E. 11th Pl., Chicago 5; *for mail,* 705 E. 88th Pl., Chicago 19, Ill.
Jones, David T. ('98; '04), Gen. Mgr., Wilbraham Green Div., Roots-Connersville Blower Corp.; *for mail;* 818 High St., Pottstown, Pa.
Jones, Donald D. (J'38) (CJL), Extruded Plastics Inc., New Canaan Ave., Norwalk, Conn.

Jones, Don R. A. (J'41) (EFKZHS), Indus. Gas Engr., So. Calif. Gas Co., 515 E. Compton Blvd.; *for mail,* 404 S. Chester Ave., Compton, Calif.
Jones, Edward L. (J'40) (BCE), c/o Day & Zimmermann, Engrs., Packard Bldg., Philadelphia, Pa.
Jones, Edward Stratton ('19) (EMSKF), Cons. Engr., 58 Sutter St., San Francisco 4, Calif.
Jones, Ernest (J'40) (BCSFHK), Engrg. Asst., Lever Bros. Ltd., 299 Eastern Ave., Toronto 8, Ont., Can.
Jones, Ernest E. ('24; '28; '35), Power & Elec. Engr., Arnold Print Works; *for mail,* 36 Beacon St., North Adams, Mass.
Jones, Fiske Reed ('27) (ACM), White St., Lunenburg; *for mail,* Box 482, Fitchburg, Mass.
Jones, Forrest Emory ('46) (ES), Lt. Col., U.S.A.; Grandview, Mo.
Jones, Frank Alfred ('35; '44) (JMRACB), Ch. Design Engr., Enterprise Wheel & Car Corp., Bristol, Va.; *for mail,* 60 Wall St., Bristol, Tenn.
Jones, Frank Stoddard (J'43) (ACJ), 55 Northfield Rd., New Rochelle, N.Y.
Jones, Frederick S. ('43) (OCSWTH), Staff Cons. Engr., Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4, N.Y.
Jones, George Merrell ('38) (CDELRS), Sales Mgr., Landes Mchy. Co., 171 W. South Temple, Salt Lake City 1; *for mail,* 319 Douglas St., Salt Lake City 2, Utah.
Jones, Harlen R. E. (J'31), Safety Engr., Royal Indemnity Co.; *for mail,* 129 S. King St., Honolulu, T.H.
Jones, Harold L. ('19; '25; '26), V.P., W.W. Farrier Co., 44 Montgomery St., Jersey City, N.J.
Jones, Henry R. (J'31) (JLM), Gen. Chem. Co.; *for mail,* 27—5th St., Pulaski, Va.
Jones, Henry Stewart ('42), Clewiston, Fla.
Jones, Homer L. (J'38) (CDJ), Lt., Corps of Engrs. U.S.A., Philippine Dept., Ft. McKinley, Manila, P.I.; *for mail,* 167 Clay St., Rochester, Pa.
Jones, Houston (J'41) (CM), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 1567 Union St., Schenectady 8, N.Y.
Jones, J. Arnold (J'41) (AEF), Mgr., Lubricants Research, Kendall Refining Co., Bradford, Pa.
Jones, James Bernard ('23; '35) (EFHS), Head, Dept. of Mech. Engrg., Va. Poly. Inst., Box 205, Blacksburg 3, Va.
Jones, James Beverly (J'44), Sch. of Mech. Engrg., Purdue Univ.; *for mail,* 1202 North St., Lafayette, Ind.
Jones, Mrs. James Beverly (J'45) (AES), 1202 North St., Lafayette, Ind.
Jones, James Dyer ('21) (CDFJLO), V.P., Charge of Engrg., H. A. Brassert & Co., 60 E. 42nd St., New York 17, N.Y.
Jones, James Robert (J'44) (SFERCA), Serv. Engr., No. Ill. Coal Corp., 310 S. Michigan Ave., Chicago 4, Ill.
Jones, John Delbert ('44) (EHON), M.M., Pipe Line Div., Gulf Refining Co., Box 661, Tulsa 2, Okla.
Jones, John Franklin (J'42) (ABKEN), Design Calculations, Chrysler Corp., Box 1919, Detroit 31; *for mail,* 1611 Cortland Ave., Detroit 6, Mich.
Jones, John Gething ('40) (COL), Assoc., Robert Heller & Associates, Inc., 1052 Union Commerce Bldg., Cleveland 14; *for mail,* 41 W. Patterson Ave., Columbus, Ohio.
Jones, John Kenneth (J'44), Lt., U.S.N.R., Naval Degaussing Sta., Naval Base, Pearl Harbor, T.H., Navy 128, F.P.O., San Francisco, Calif.; *for mail,* 1819 Dunstan Rd., Houston 5, Tex.
Jones, John Lucien ('43) (SKEFHG), Asst. Prof., Mech. Engrg., Va. Poly. Inst., Box 645, Blacksburg, Va.
Jones, Kenneth A. (J'43), 807 James Bldg., Chattanooga, Tenn.
Jones, Leslie William ('45) (NBMGJY), Product Engr., Hilliard Corp., 102 W. 4th St.; *for mail,* 176 Bower Rd., Elmira, N.Y.
Jones, Lewis ('15), McKay Mch. Co., Rayen & Foster Sts., Youngstown 2, Ohio.
Jones, Lloyd B. ('16) (SFJR), Test Engr., Test Dept., Pa. R.R., Altoona, Pa.
Jones, Mack M. ('42) (KBEH), Cooling Lab. Supvr., Airtemp Div., Chrysler Corp., 119 Leo St., Dayton; *for mail,* 58 E. Fairview Ave., Dayton 5, Ohio.
Jones, Michael D. ('42) 120 S. 16th St., Allentown, Pa.
Jones, Morris William (J'32) (J), Co-Partner, J. & M. Metal Bonders, 119 Dyer St., Providence 1, R.I.
Jones, Ralph Wilson, Jr. (J'38) (NCMHJZ), Mech. Engr., Naval Ord. Test Sta.; *for mail,* Bachelor Officers' Quarters-C, Naval Ord. Test Sta., Inyokern, Calif.
Jones, Robert M. (J'44) (BCM), Research Engr., Stevens Inst. of Tech., Hoboken, N.J.
Jones, Robert Murph, Jr. (J'45) (NMO), Mech. Engr., Designer, Air Conditioning Div., Brown & Root, Inc., Box 3, Houston 1; *for mail,* 1217 Kenwood St., Houston 6, Tex.
Jones, Russell Compton ('12; '19) (KS), Pres., Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.
Jones, Russell Eugene ('28) (JBKCDH), Ch. Engr., Birmingham Tank Co. Div., Ingalls Iron Works Co., Birmingham; *for mail,* 1136—10th Pl. S., Birmingham 5, Ala.
Jones, Rutgers Crosby (J'41) (SJ), Acting Ch. Inspr., Heine Boiler Div., Combustion Engrg. Co., Inc., 5319 Shreve Ave., St. Louis 15, Mo.
Jones, Samuel B. ('24) (WSOLKE), 652 Cane Run, Harrodsburg, Ky.
Jones, Sidney Upson (J'45) (AESRYF), Ensign, U.S.N., Engrg. Officer, U.S.S. *San Jacinto,* CVL-30, F.P.O., San Francisco, Calif.; *for mail,* 911 Catalonia Ave., Coral Gables, Fla.
Jones Spencer Ainsworth ('44) (OJDBMZ), Cons. Engr., 122 E. 42nd St., New York 17, N.Y.
Jones, Thomas A. ('44) (LDSOKC), Ch. Engr., Kingan & Co., Inc., Blackford & Maryland Sts., Indianapolis 6; *for mail,* Box 311-J, Route 16, Indianapolis, Ind.
Jones, Walter Alfred (J'43) (AZN), Project Engr., Kollsman Instrument Div., Square "D" Co., 80-08—45th Ave., Elmhurst, L.I.; *for mail,* 235 E. 82nd St., New York 28, N.Y.
Jones, Walter Bond ('25; '30; '35) (HBNGE), Asst. Engr., Bur. of Engrg., City of San Francisco, 367 City Hall, San Francisco 12, Calif.
Jones, Walter F. ('20; '35) (JKL), Branch Mgr., Carrier Corp., 405 Lexington Ave., New York 17, N.Y.
Jones, Walter Idris, Jr. (J'43) (CMJ), Sgt., Property Branch, Twin Cities Field Office, Chicago Ord. Dist., 609—2nd Ave. S., Minneapolis 2, Minn.; *for mail,* 1218 Taylor St., Joliet, Ill.
Jones, William Philip (J'45) (NMCTO), Jr. Engr., Product Devel., Chicopee Mfg. Co., Main St., Chicopee Falls; *for mail,* 271 Norfolk St., Springfield 9, Mass.
Joos, Charles E. ('41) (CKS), Mgr., Sales Apparatus Div., Cochrane Corp., 17th & Allegheny Ave., Philadelphia, Pa.
Joost, Geo. E. ('29; '35) (CDMNW), Supvg. Engr., Engrg. Records Dept., Hyster Co., 2902 N.E. Clackamas St., Box 4318, Portland 8, Ore.
Jordan, William A. ('03; '12), Salesman, Essex Wire Corp., 457 Frelinghuysen Ave., Newark 5, N.J.; *for mail,* 340 E. 57 St., New York 22, N.Y.
Jordan, William McCay (J'43), Cpl., Aircraft Mechanic, Army Air Forces, 44th Bomber Hdq. & Aircraft Sqd., Det. A, Army Air Base, Portland 19, Ore.; *for mail,* 502 Redlands St., Redlands, Calif.
Jordre, William Starling ('42) (OSDN), V.P., Oberle-Jordre Co., Inc., 613 Traction Bldg., Cincinnati, Ohio; *for mail,* Box 986, R.R. 4, Erlanger, Ky.
Jorgensen, Wm. ('22; '42) (BKS), Mar. Design & Turbine Engrg., N.Y. Shipbldg. Corp., Camden, N.J.
Jorgenson, John George (J'41) (NJA), Devel. Engr., Kearney & Trecker Corp., 6784 W. National Ave., Milwaukee 14; *for mail,* 133 E. Capital Dr., Hartland, Wis.
Jorstad, Oswald J. (J'39) (BGS), Babcock & Wilcox Co., 85 Liberty St., New York, N.Y.; *for mail,* 184—6th St., N.W., Barberton, Ohio.
Jory, Robert ('26) (SF), Sales Engr., Riley Stoker Corp., 103 Park Ave., New York 17, N.Y.
Josepho, Anatol M. ('29) (AN), Exper. Engr., 1229 Wilshire Blvd., Santa Monica, Calif.
Josephs, John Pearson (J'42) (BE), 1st Lt., Ord. Dept., U.S.A.; *for mail,* Choice St., Bel Air, Md.
Josephs, Lyman Colt, Jr. ('19; '25) (EBC), Ch. Engr., Automotive Div., Ord. Research & Devel. Center, Aberdeen Proving Ground; *for mail,* Bel Air, Harford Co., Md.
Josephs, Lyman Colt, III (J'43) (ABC), Lt., U.S.N.R., Bur. of Aero., Navy Dept., Washington 25, D.C.
Josephson, Sidney N. (J'39) (DOCESJ), Designer, Natl. Tube Co., Box 508, Gary, Ind.
Joslyn, Ray Oliver ('31) (CH), Pres., Layne West. Co., 1010 W. 39th St., Kansas City 2, Mo.
Jourdin, Willis W. ('12), 1906 Cincinnati Ave., San Antonio 1, Tex.
Joy, Frank A. (J'44) (CKS), 85 Maple St., Malden, Mass.
Joy, Jos. ('17; '21; '35), Sales Engr., Diamond Chain & Mfg. Co., Indianapolis, Ind.; *for mail,* Apt. E.B.P.H., 40 Monroe St., New York, N.Y.
Joy, Richard Pierce (J'43) (AFEC), Jr. Engr., Union Oil Co., Wilmington; *for mail,* 575 Glen Holly, Pasadena 2, Calif.
Joyce, Charles S. ('22) (ABE), Capt., U.S.N., Retired; Army & Navy Club, Washington 6, D.C.
Joyce, Harry B. ('31) (FKSW), Consultant Engr., 616 Commerce Bldg., Erie, Pa.
Joyce, Reginald ('30; '45) (WNCBM), V.P., Asst. Tech. Dir., Consumers' Research, Inc.; *for mail,* R.F.D. 1, Washington, N.J.
Joyce, Richard E. Jr. (J'43) (AGM), Sales Engr., Cuno Engrg. Corp.; *for mail,* 21 Hillcrest Terrace, Meriden, Conn.
Jozefowicz, Edward (J'43) (ABM), Propeller Div., Curtiss-Wright Corp., 880 Main St., Stamford, Conn.
Jubel, Henry Albert (J'40) (MLN), Prod. Engr., Sterling Aluminum Products Co., 2925 N. Market St., St. Louis; *for mail,* 1207 Telegraph Rd., Lemay 23, Mo.
Juchtern, Charles David ('28; '34; '35) (BFKS), Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 80 Princeton St., Garden City, N.Y.
Jud, John Joseph (J'44) (KHSZBF), Test Engr., Philco Radio & Television Corp., C & Allegheny Ave., Philadelphia; *for mail,* 3505 Lansing St., Philadelphia 36, Pa.
Judd, Charles M. (J'43) (CMJWTD), Prod. Mgr., Chas. W. Breneman Co., 2045 Reading Rd., Cincinnati 2, Ohio.
Judd, Marcus Ambrose (J'44) (CEW), Sr. Engr., Aide (Mech.), U.S. Engr. Office, Fed. Bldg., Milwaukee 1; *for mail,* 2521-A N. 38th St., Milwaukee 10, Wis.
Judd, Sebastian (J'40) (CJNR), Design & Devel. Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7; *for mail,* 319 E. Beaumont Ave., Milwaukee 11, Wis.
Jude, Henry (J'16), Spec. Rep., Manning, Maxwell & Moore, Inc., 135 E. 42nd St., New York 17, N.Y.
Judge, J. Emmet (J'44) (CMD), Indus. Engr., Wallace Clark & Co., 521—5th Ave., New York 17; *for mail,* 85 Parkview Ave., Bronxville 8, N.Y.
Judkins, Malcolm Faulkner ('32; '33; '35) (CJM), Ch. Engr., Firthite Div., Firth-Sterling Steel Co.; *for mail,* 1203 Summit St., McKeesport, Pa.
Judson, Charles James (J'44) (LDWZSO), Indus. Consultant, 818 Jackson St., Oregon City, Ore.
Judson, Robert Edward (J'44), 5555 Everett Ave., Chicago 37, Ill.
Judy, George Lee (J'42) (SNHBCJ), Asst. Maint. Supt., Consolidation Coal Co., Fairmont, W.Va.
Juelss, David ('35) (CYMNDW), Ch. Engr., Am. Pencil Co., 500 Willow Ave., Hoboken, N.J.
Juer, Robert (J'37) (MLDCKS), 328 S. Sycamore St., Petersburg, Va.
Jukoff, David (J'44) (AJK), Ensign, U.S.N., Rm. 213, Bachelor Officers' Quarters Annex, Naval Air Sta., Moffett Field, Calif.
Julien, James Herbert (J'43) (LCMZ), Major Unit Shift Foreman, Carbide & Carbon Chemicals Corp.; *for mail,* Boone Hall, Oak Ridge, Tenn.
Jull, Thomas Alfred (J'43) (HCSEM), Test Engr., John Ingus Co. Ltd., 14 Strachan Ave.; *for mail,* 59 Rosedale Heights Dr., Toronto, Ont., Can.
Junas, Ladislav John (J'44) (EORS), Motor Machinist's Mate, 1st Class, U.S.N.R., Berthing Facility, Tompkinsville, S.I.; *for mail,* 45-11 Junction Blvd., Corona, L.I., N.Y.
Jungbauer, John J. (J'39) (JYG), Mech. Engr., Engrg. Lab., Minnesota Mining & Mfg. Co., 900 Faquier Ave., St. Paul 6; *for mail,* 636 Lafond Ave., St. Paul 4, Minn.
Junker, Alex J. ('32; '44) (BMR), Ch. Engr., Devel. Div., Libbey-Owens-Ford Glass Co., 1701 E. Broadway, Toledo; *for mail,* 337 Hillside Dr., Rossford, Ohio.
Junker, William Henry ('42) (CLKBO), Works Mgr., Emery Industries, Inc., June & Long; *for mail,* 6068 Dryden Ave., Cincinnati 17, Ohio.
Junttila, Wilbert Henry (J'42) (AM), Capt., 8th Army Air Force, 352nd Fighter Sqd., 353rd Fighter Gp., A.P.O. 637, c/o Postmaster, New York, N.Y.
Jupa, Edward Charles (J'44) (BHS), Cowles & Co., 612 N. Michigan Ave., Chicago 11; *for mail,* 3506 S. Euclid Ave., Berwyn, Ill.
Juram, Albert Edward (J'44), Indus. Engr., Saw Div., Henry Disston & Sons, Inc., Tacony St., Philadelphia; *for mail,* 2476 W. 79th Ave., Philadelphia 38, Pa.
Juran, Joseph M. ('41) (CMO), *Worcester Reed Warner Medallist,* '45; Prof. & Chmn., Admin. Engrg., N.Y. Univ., University Heights, New York 53, N.Y.
Justice, Charles Theodore (J'42) (FKS), Mech. Plant Betterment, Ebasco Services Inc., 2 Rector St., New York, N.Y.
Justice, Felix Calvin ('33; '41) (CESKF), Engr., Dallas Power & Light Co., 1506 Commerce St., Dallas 1, Tex.
Justice, Lloyd W. (J'39) (CRS), Student Test Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 119 Washington St., Marblehead, Mass.
Justice, Russel Wilson, Jr. (J'42) (ABNSME), Serv. Engr., Factory A-1, Lockheed Aircraft Corp., Burbank; *for mail,* 8713—12th Ave., Inglewood, Calif.
Justice, William Clayton (J'35) (SZF), Asst. Engr., U.S. Capitol Power Plant, 1st & E Sts., S.E., Washington 3, D.C.; *for mail,* 4202—53rd Ave., Hyattsville, Md.
Justus, James E. ('35; '35) (RKEFJB), Mgr., Diesel Loco. Serv. Dept., Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago, Ill.; *for mail,* 724½ Chapin St., Beloit, Wis.

K

Kacmarcik, Vladimir Ernest (J'42) (BMHNCA), Plant & Tool Engr., Hill-Acme Co., 6400 Breakwater Ave., Cleveland 2; *for mail,* Pershing Dr., Brecksville, Ohio.
Kaczynski, Zygmund Lawrence (J'40) (BDM), Secy., Charge Prod., Sioux Mfg. Co., Detroit; *for mail,* 29145 Dufty Rd., Route 1, Birmingham, Mich.
Kaddeland, Christen Rosfjord ('29) (GCOMS), Engr., Specialties & Constr., Harris-Seybold-Potter Co., 4510 E. 71st St., Cleveland, Ohio.
Kadz, Harald Dungan ('45), Gen. Mgr., Equip. Div., Kay-Brunner Steel Products, Inc., Los Angeles; *for mail,* 656 Balboa Ave., Glendale, Calif.
Kaeding, Raymond R. (J'43) (AHZS), Lab. Tech., Woodward Governor Co., N. 2nd St. Rd.; *for mail,* Wilson Ave., R.R. 2, Rockford, Ill.
Kaemmerling, Gustav Henry ('16; '22; '35) (YABJEN), Pat. Research Engr., Lord Mfg. Co., 1635 W. 12th St.; *for mail,* 403 W. 6th St., Erie, Pa.

Kaempffert, Waldemar ('30), Sci. & Engrg. Editor, *New York Times*, 10th Floor, 229 W. 43rd St., New York, N.Y.
Kaeufler, Curt W. V. ('27) (BCM), Secy., Treas., Gen. Engrg. & Mfg. Co., 1523 S. 10th St., St. Louis 4; *for mail*, 8149 Kingsbury Blvd., Clayton 5, Mo.
Kaganowich, Morton Lawrence (J'44) (LNG DAC), Div. Engr., Ansco Div., Gen. Aniline & Film Corp., Charles St.; *for mail*, 64 Murray St., Binghamton, N.Y.
Kahlenberg, Roger William (J'25) (CEMN), V.P., Ch. Engr., Kahlenberg Bros. Co.; *for mail*, 2600 Washington St., Two Rivers, Wis.
Kahler, William Grimes (J'44) (BNOCKL), Mech. Engr., Carbide & Carbon Chemicals Corp., South Charleston 3; *for mail*, 447 Forest Circle, South Charleston, W.Va.
Kahn, Bertrand B. ('18) (FJKB), 1st V.P., Estate Stove Co., East Ave., Hamilton, Ohio.
Kahn, Herman (J'35), Consltd. Products Co., 15 Park Row, New York, N.Y.
Kahn, Jas. M. ('30; '35), Sales Engr., Internatl. Filter Co., 1260 W. Wesley Rd., Atlanta, Ga.
Kahrs, Henry George (J'45) (NCBFLZ), Gen. Mgr., Clark & George, Inc., 10 S. Cottage St., Valley Stream; *for mail*, 37 Clark Ave., Lynbrook L.I., N.Y.
Kain, Edward M. (J'36), 3585 Normandy Rd., Shaker Heights, Ohio.
Kainer, John E. (J'40) (JNOCEM), Owner, Kainer Welding Serv., 108 Fahrenthold, El Campo, Tex.
Kaiser, Elmer Albert ('45), Dist. Mgr., Detroit Stocker Co., 410 Traction Bldg., Cincinnati 2, Ohio.
Kaiser, Elmer R. ('34; '46) (FE), Asst. Dir. of Research, Bituminous Coal Research, Inc., Oliver Bldg., Pittsburgh 22, Pa.
Kaiser, Franz F. (J'40), Devel. Engr., Hillsdale Steel Products; *for mail*, 15 Charles St., Hillsdale, Mich.
Kaiser, Herbert A. (J'43) (BEKSR), Office Engr., Hartford Steam Boiler Inspc. & Ins. Co., 90 John St., New York 7; *for mail*, 62-64—80th Rd., Glendale, Brooklyn 27, N.Y.
Kaley, George Budd ('13; '23) (J), Owner, Firm of George B. Kaley, 39 Broadway, New York 6, N.Y.
Kalkhoff, Amos W. ('40; '44) (CLBYDN), Plant Engr., Flintkote Co.; *for mail*, 1325 Franklin Ave., Chicago Heights, Ill.
Kallam, Floyd Loren ('43) (KEFLSN), *Junior Award*, '22; Pres., Indus. Engrs., Inc., 819 E. 59th St., Los Angeles 1; *for mail*, 3861 E. California St., Pasadena 8, Calif.
Kallet, Eli Alan (J'45) (HBSERK), 1st Lt., Army Air Forces, Contract Termination Officer, Lear Aviation, Inc., Grand Rapids, Mich.; *for mail*, c/o Henenfeld, 945 Faile St., New York 59, N.Y.
Kallgren John Elton (J'41) (BD), Mech. & Planning Engr., U.S. Rubber Co., 1230—6th Ave., New York 20, N.Y.
Kallgren, Robert Murray (J'46) (EKS), 529 N. Firestone Blvd., Akron 1, Ohio.
Kalmbach, Charles Frederic (J'41) (RBA), Maj., Ord. Dept., U.S.A., Ft. George G. Meade, Md.; *for mail*, 269 N. Highland Ave., Lansdowne, Pa.
Kalmbach, Fred'k ('28) (BFK), Pres., Gen. Mch. Co., Inc., 4th & Furnace Sts., Emmaus, Pa.
Kalustian, Edward (J'43) (ACZMYO), Vibration Field Test Engr., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail*, 719 George St., Teaneck, N.J.
Kalvelage, Frank J., Jr. (J'42) (MBCASE), 12104 Monica Ave., Detroit, Mich.
Kambe, Tom Jack (J'43) (BHKMAC), Mech. Design Engr., Dept. 39-44, Plant A 2, Lockheed Aircraft Corp., Box 71, Burbank; *for mail*, Apt. 1, 529 E. Maple St., Glendale 5, Calif.
Kammer, Karl Pfister ('37) (CSKF), Gen. Supt., Elec. Dept., New Orleans Pub. Serv., Inc., 317 Baronne St., New Orleans 9, La.
Kammerhoff, H. H. Meno ('14) (BCMS), 405—10th Ave., Haddon Heights, N.J.
Kammerlohr, Harry William (J'43) (ABM), 1st Lt., Corps of Engrs., Co. C, 2807th Engr. Bn., A.P.O. 244, c/o Postmaster, San Francisco, Calif.; *for mail*, Kenesaw, Neb.
Kamo, Masawo ('25; H'29), Emeritus Prof., Tokyo Imperial Univ., Tokyo, Japan.
Kamps, Henry Edward (J'46), Research Dept., Elliott Co., Jeannette, Pa.
Kamptner, Gustav ('42) (BGNR), Loco. Designer, Denver & Rio Grande West. R.R. Co., 1531 Stout St., Denver 1; *for mail*, 4530 W. 46th Ave., Denver 12, Colo.
Kandiko, Joseph C., Jr. (J'44), Engr., Gen. Elec. Co.; *for mail*, 612 Union St., Schenectady, N.Y.
Kane, Alvin Bernard ('44), Ch. Engr., Mech. & Elec. Div., Mutual Boiler Ins. Co. of Boston, 60 Batterymarch St., Boston; *for mail*, 25 Wedgemere Rd., Medford, Mass.
Kane, Edmund J. ('16) (DEN), Engr., Moore Bros., 606 S. Dearborn St., Chicago; *for mail*, 123 N. Waller Ave., Chicago 44, Ill.
Kane, Eneas Dillon (J'38) (NBM), Asst. Prof. Mech. Engrg., Univ. of Calif., Berkeley; *for mail*, 1220 Neilson St., Berkeley 6, Calif.
Kane, William Russell (J'45) (BMNOHD), Plant Engrg. & Process Design, Indus. Rayon Corp., W. 98th St. & Walford Ave., Cleveland 1; *for mail*, 1361 W. 80th St., Cleveland 2, Ohio.
Kanik, Rudolph Mike ('29) (CIMK), Works Mgr., Bagley & Sewell Co., 101 Pearl St., Watertown, N.Y.
Kanter, Jerome Julius ('38) (BJ), Matls. Research Engr., Crane Co., 4100 S. Kedzie Ave., Chicago 32, Ill.
Kantor, James ('40) (BHK), Ch. Engr., Mch. Div., Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago, Ill.
Kaplan, Eugene Stanley (J'43) (NMJ), Asst. Mech. Engr., Fed. Tel. & Radio Corp., 100 Kingsland Rd., Clifton, N.J.; *for mail*, 6213—23rd Ave., Brooklyn 4, N.Y.
Kaplan, Harry (J'16), Gibbs & Hill Pa. Sta., New York 1; *for mail*, 1686 Grand Concourse, New York, N.Y.
Kaplan, Louis G. (J'42) (BEH), 1322 Sherwin Ave., Chicago 26, Ill.
Kaplan, Malcolm John (J'40) (AJSEHN), Htg. Engr., Constr. Serv., Veterans Admin., 252—7th Ave., New York; *for mail*, 476 Fulton Ave., Hempstead, L.I., N.Y.
Kappelman, Raymond Vernon (J'44) (ABL), Pvt., Sqd. D-2, A.P.O. 19780 D, c/o Postmaster, New York, N.Y.
Kaprelian, Edward Karnig (J'34) (ZBE), Optical Engr. & Pat. Atty., Ground Signal Pat. Agency, Belmar; *for mail*, 25 Locust Ave., Long Branch, N.J.
Kapstein, Seymour Charles (J'46), 143 Babcock St., Brookline 46, Mass.
Karassik, Igor John ('35; '45) (SHEDL), Application Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail*, 476 Walton Rd., Maplewood, N.J.
Karch, Richard Granger (J'45) (CDJ), Research Engr., Am. Thermos Bottle Co., Norwich, Conn.
Karelitz, Michael B. ('38) (BCEH), Mech. Engr., Gen. Precision Labs., Box 667, Pleasantville, N.Y.
Karg, William Earnest ('24; '43) (CJKS), Ch. Engr., Burlington Generating Sta., Pub. Serv. Elec. & Gas Co., Burlington, N.J.
Karge, Fritz Wilhelm ('43) (EHKO), Ch. Engr., Pipe Line Dept., Union Oil Co. of Calif., 617 W. 7th St., Los Angeles 14; *for mail*, 1980 Escarpa Dr., Los Angeles 41, Calif.
Karhan, Raymond Albert (J'42) (BJMSYK), Devel. Engr., York Corp., Roosevelt Ave., York, Pa.
Karla, Gustav ('41) (OSMD), Owner, Firm of Gustav Karla, Cons. Engr., 509—2nd Ave., Seattle 4, Wash.
Karlson, Charles B. ('22; '35) (EHJLOS), Pres., Steel Products Corp., 52 Vanderbilt Ave., New York 17, N.Y.
Karlson, Karl W. ('29) (FSKER), Fuel Engr., Republic Coal & Coke Co., 8 S. Michigan Ave., Chicago 3, Ill.; *for mail*, 708 Arthur Ave., Racine, Wis.
Karlsson, Hilmer ('27; '35; F'46) (SFKB), Ch. Engr., Air Preheater Corp., 121 S. Main St.; *for mail*, 102 Early St., Wellsville, N.Y.
Karmazin, John ('14; 35), Cons. Engr., Frididaire Div., Gen. Motors Corp., Huntington, Ind., Pres., Karmazin Engr. Co. Grosse Ile.; *for mail*, Grosse Ile, Mich.
Karp, Daniel S. (J'40) (JMCD), V.P., Karp Metal Products Co., Inc., 129—30th St., Brooklyn 32, N.Y.
Karp, Martin Everett (J'42) (BEHJM), Ensign, U.S.N., U.S.S. *Marquette*, AKA-95; *for mail*, 194—133rd St., Belle Harbor, L.I., N.Y.
Karpov, A. V. ('41), Cons. Engr., Century Bldg., Pittsburgh 22, Pa.; *for mail*, 40 Exchange Pl., New York 5, N.Y.
Karr, James Joseph (J'40) (BSHNLM), Mech. Engr., Kearney & Trecker Corp., Milwaukee 14, Wis.
Karre, William A. ('23; '32) (DZKLYC), Plant Engr., Mathieson Alkali Works, Inc.; *for mail*, 2246 Pierce Ave., Niagara Falls, N.Y.
Karsten, Walter F. R. ('43) (CKL), Mgr., Indus., Mar. & Contr. Equip. Div., Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.
Karsunky, Wm. Karl ('22; '31) (OSKHFE), Cons. Engr., 1223 Connecticut Ave., N.W., Washington 6, D.C.; *for mail*, Three Oaks, Kensington, Md.
Kase, Lester Michael (J'42) (FRS), Sgt., Hdq. Det., Engr. Sec., Com. Z, European Theater of Operas., Normandy Base Sec., A.P.O. 562, c/o Postmaster, New York 4, N.Y.; *for mail*, 916 S. Austin Blvd., Oak Park, Ill.
Kaser, Arthur J. (J'33) (EFS), 3551 W. 147th St., Cleveland, Ohio.
Kasonik, John, Jr. (J'41), 431 Patton St., Wilmerding, Pa.
Kass, Philip (J'42) (FKS), Apt. 32, 1226 Sherman Ave., Bronx 56, N.Y.
Kassander, Arno Richard ('36) (CM), C.P.A., Lybrand, Ross Bros. & Montgomery, 90 Broad St., New York 4, N.Y.
Kassebohm, Walter H. ('30; '34; '35) (CJMB NO), Gen. Mgr., Prod. Engrg. Co., 940 Dwight Way, Berkeley 2; *for mail*, 78 The Uplands, Berkeley 5, Calif.
Kast, Francis Xavier (J'44) (AKMNH), Engrg. Officer, U.S.N., U.S.S. *Cowpens*, F.P.O., San Francisco; *for mail*, 37 Bolinas Ave., San Anselmo, Calif.
Kast, William Gregory (J'44) (CSMKN), Gunnery Officer, U.S.N., U.S.S. *Antietam*, F.P.O., San Francisco; *for mail*, 37 Bolinas Ave., San Anselmo, Calif.
Kasten, Fred McRae (J'44) (NAMZ), Mech. Engr., Precision Thread Grinding Co., 6651 Tireman Ave., Detroit; *for mail*, 12930 Asbury Pk., Detroit 27, Mich.
Kastle, Clarence McSherry (A'46), Indus. Sales Engr., Socony-Vacuum Oil Co., Inc., Box 869, Charleston 23, W.Va.; *for mail*, Box 3-E, Richmond 6, Va.
Katcher, Morris ('19; '30) (NOMCL), Prin., Modern Engrg. Co., 597—5th Ave., New York 17, N.Y.
Katelus, Geo. J. (J'39) (CJM), Equip. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail*, 155 Court St., Elizabeth, N.J.
Katerndahl, Dean Richard (J'37) (YA), 1st Lt., Army Air Forces, Det. 111, 139th Army Airways Communication System Sqd., A.P.O. 919, c/o Postmaster, San Francisco, Calif.; *for mail*, 50 N. Long Ave., Chicago 44, Ill.
Kates, Edgar J. ('16; '21) (EOFS), *Director-at-Large*, '45-'49; Cons. Engr., Diesel Specialist, 415 Lexington Ave., New York 17, N.Y.
Kathmann, Alphonse J. (J'45) (BDL), Mch. Designer, ABC Packaging Mach. Co., 330 S. 6th St.; *for mail*, 219 S. 11th St., Quincy, Ill.
Katow, M. Smoot (J'43) (NHBJML), Devel. Engr., Chicago Pump Co., 2300 W. Wolfram St.; *for mail*, 3548 N. Halsted St., Chicago 13, Ill.
Kattelle, Laurence Watson ('36) (BDR), Engr., Walworth Co., 60 E. 42nd St., New York 17, N.Y.
Kattelmann, Harry Richard (J'43) (NBJM), Design Engr., Marchant Calculating Mch. Co., Oakland 8; *for mail*, 5287 Broadway Terrace, Oakland 11, Calif.
Katz, Donald LaVerne ('45) (KELHF), Prof. Chem. Engrg., Univ. of Mich., 2028 E. Engrg. Bldg., Ann Arbor, Mich.
Katz, Gerald Harold (J'45) (KSE), Engr., Layout & Installation, Allen Plumbing & Htg. Co., 345 Massachusetts Ave., Boston; *for mail*, 100 Grandville Ave., Malden 48, Mass.
Katz, Israel (J'41) (BEM), Asst. Prof. Aircraft Eng. Design, Sibley College, Cornell Univ.; *for mail*, R.D. 2, Varna Locality, Ithaca, N.Y.
Katz, Maurice (J'45) (OWACNB), Cons. Engr., 86 Elm Hill Ave., Roxbury 21, Mass.
Katz, Nelson J. (J'45) (SNBLCO), Detail Piping Draftsman, Atlantic Refining Co., 260 S. Broad St., Philadelphia 2; *for mail*, 5449 Arlington St., Philadelphia 31, Pa.
Katz, Peter (J'45), Designer, Remington Rand, Inc., 131 Flatbush Ave. Ext., Brooklyn; *for mail*, 155 S. Grove St., Freeport, L.I., N.Y.
Katzenstein, Martin L. ('03; '15) (HKS), Pres., Warren Engrg. Co., 117 Liberty St., New York 6, N.Y.
Kaufelt, Curtis Lindner (J'45), Asst. Planning Engr., Milwaukee Works, Internatl. Harvester Co., 1714 W. Bruce St., Milwaukee 1; *for mail*, 2942 N. 60th St., Milwaukee 10, Wis.
Kauffeld, Theodore John ('40) (CMHK), V.P., Dir., Devenco, Inc., 150 Broadway, New York 7, N.Y.; *for mail*, Punch Bowl Rd., Convent, N.J.
Kauffman, Herbert P. ('23; '35) (MLJNG), Sales Engr., J. W. Greer Co., 119 Windsor St., Cambridge; *for mail*, 97 Mt. Vernon, Boston 8, Mass.
Kauffman, Wilbur W. (J'38) (CFJ), Lt. Comdr., Naval Ord. Plant, Macon, Ga.
Kauffmann, William Martin (J'32) (EB), Asst. to Ch. Engr., Mack Mfg. Corp., Empire State Bldg., 5th Ave. & 34th St., New York 1, N.Y.; *for mail*, 143 Sanford Ave., Plainfield, N.J.
Kaufman, Daniel (J'44) (CSMKN), Electronic Technician's Mate, 1st Class, U.S.N.R., Navy 3002, Box 46, F.P.O., San Francisco, Calif.; *for mail*, 3735—90th St., Jackson Heights, N.Y.
Kaufman, George Edward (J'44) (NMOC), Jr. Engr., Design & Devel., Hoover Co., North Canton; *for mail*, 1164 Lexington Ave., Akron 10, Ohio.
Kaufman, Milton ('34; '45) (NHBZKE), Ord. Engr., Naval Gun Factory, Washington; *for mail*, 1422 Webster St., N.W., Washington 11, D.C.
Kauppinen, Tenho Savel (J'40) (MNGSB), Instr., Univ. of N.H.; *for mail*, 1 Main St., Durham, N.H.
Kaveny, John Gordon ('45) (KM), Pres., Gen. Mgr., Roos Tool & Mfg. Co. & Quiet Auto Burner Corp., 17-19 Grove St., Montclair; *for mail*, Fairfield Rd., Little Falls, N.J.
Kaveny, Thomas, Jr. ('28; '34; '35) (CLBM), Pres., Herman Pneumatic Mch. Co., 1806 Union Bank Bldg., Pittsburgh 22, Pa.
Kay, Lloyd D. ('32), Pres., Kay-Brunner Steel Products, Inc., 999 Meridian Ave., Alhambra, Calif.
Kayan, Carl F. ('25; '40) (KLZBSE), Assoc. Prof., Columbia Univ., Morningside Heights, New York 27, N.Y.
Kaye, James W. ('40) (KMS), Ch. Engr., Power Serv. Div., Thos. A. Edison, Inc., Main St.; *for mail*, 33 Franklin Ave., West Orange, N.J.
Kaye, Joseph (J'41) (BAKES), Asst. Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge 39, Mass.
Kayes, William Joseph ('39; '44) (MCNBWK), Prod. Engr., Serv. Mgr., Trailmobile Can. Ltd., Box 450, Windsor, Ont., Can.
Kazen, Mitchell Christopher (J'43), *Charles T. Main Award*, '43; 3618 Caniff Ave., Detroit, Mich.
Kazik, John, Jr. (J'42) (CDLMOS), Westinghouse Elec. Corp., East Pittsburgh; *for mail*, 715 South Ave., Pittsburgh 21, Pa.
Kazutoff, Alexander ('32; '41) (ABHK), Research Engr., Factory A, Lockheed Aircraft Corp., Burbank; *for mail*, 14966 Dickens St., Sherman Oaks, Calif.

Keachie, Edward Chester (A'43) (CMG), Asst. to V.P., Newbery Elec. Corp., Los Angeles; *for mail,* 1141 W. 47th St., Los Angeles 37, Calif.
Kearney, Frank Vinton ('30) (AJK), Asst. Div. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.; *for mail,* 1896 Boulevard, Jersey City 5, N.J.
Kearney, Howard Marshall ('44) (BNKSZH), Mech. Engr., Radio Corp. of Am., Camden; *for mail,* 43 N. Drexel St., Woodbury, N.J.
Kearney, John Gordon ('44) (BES), Designer, Engine Dept., Fed. Shipbldg. & Dry Dock Co., 744 Broad St., Newark, N.J.; *for mail,* 47-34 Glenwood St., Little Neck, L.I., N.Y.
Kearney, Joseph Daniel (A'44) (LMT), 1458 Columbia Rd., N.W., Washington, D.C.
Kearney, O'Earl ('44), V.P., Lumsden & Van Stone Co., 426—1st St., South Boston 27, Mass.; *for mail,* Pershing Hotel Apts., 1428 Peachtree St., N.E., Atlanta, Ga.
Kearney, Thomas James ('34; '46) (LCKMND), Asst. Ch. Engr., Detrex Corp., 14331 Woodrow Wilson, Detroit 3; *for mail,* 17217 Westmoreland Rd., Detroit 19, Mich.
Kearns, Charles Maxwell, Jr. (J'36) (B), Ch. Research Engr., Hamilton Stand. Propellers Div., United Aircraft Corp., East Hartford; *for mail,* 84 Prospect St., Manchester, Conn.
Kearns, Michael I. ('38) (CEFKMO), Mgmt. Engr., Brown & Root, Inc., Box 3, Houston 1, Tex.
Keating, Arthur E. ('19; '23; '35) (CMEJ), Pres., Gen. Mgr., Trimont Mfg. Co., 55 Amory St., Roxbury 19, Mass.
Keating, Daniel Joseph, Jr. (J'41) (CO), Secy. & Treas., Daniel J. Keating Co., 613 S. 24th St.; *for mail,* 6440 Overbrook Ave., Philadelphia 31, Pa.
Keating, John Thomas ('44) (FLS), Ch. Oper. Engr., Procter & Gamble Co., 1232 W. North Ave., Chicago 22, Ill.
Keator, Frederic William ('22; '26; '30) (FHSZ), Assoc. Prof. Mech. Engrg., Sch. of Engrg., Yale Univ., 400 Temple St., New Haven 11, Conn.
Kedy, Stiles Fraser ('16) (SFHLT), Owner, Stiles F. Kedy Co., 185 Franklin St., Stoneham 80, Mass.
Kee, Robert James (J'42) (RKSM), 4th & Chestnut Sts., New Castle, Del.
Keech, George Bruce (J'43) (AJY), Aero. Engr. P-2, Naval Air Matl. Center, Naval Air Exper. Sta., Naval Shipyard, Philadelphia; *for mail,* 6818 N. Gratz St., Philadelphia 26, Pa.
Keefer, Kenneth Beach ('38) (CDM), Ch. Methods Engr., Automatic Elec. Co., 1033 W. Van Buren St., Chicago 7, Ill.
Keefer, Ray Albert (J'44), Apprentice Seaman, Co. 114, C-8-L, Naval Trng. Center, Sampson, N.Y.
Keel, Knut Olav ('44) (AYJFNC), Ch. Engr., Cleveland Diesel Eng. Div., Gen. Motors Corp., 2160 W. 106th St., Cleveland 11, Ohio.
Keeler, Hugh Edward ('12; '19; '22; F'41) (SEOKAZ), Prof. Mech. Engrg., Univ. of Mich., 231 West Engrg. Bldg., Ann Arbor, Mich.
Keeler, J. Fred (J'36) (C), Mgr., Product Engrg., Steel & Tubes Div., Republic Steel Corp., 224 E. 131st St., Cleveland 8, Ohio.
Keeler, Leonard Becknell (J'38) (CMDA), Asst. to Gen. Mgr., Wayne Div., Bendix Aviation Corp., Wayne; *for mail,* 16860 Kentfield, Detroit 19, Mich.
Keeley, William Carroll, Jr. ('29) (CMDYRS), V.P., Air Reduction Co., Inc., 60 E. 42nd St., New York 17, N.Y.
Keeling, Harry James ('42) (NEBZLJ), Mech. Engr., So. Counties Gas Co., 810 S. Flower St., Los Angeles 14, Calif.
Keely, Hazael Lafayette (A'43) (WCDY), Owner, Hal Keely Plywood Co., 2908 Smallman St., Pittsburgh 1; *for mail,* 4031 California Ave., Pittsburgh 12, Pa.
Keen, George W. ('24; '34) (SJKZCF), Supt., Power Prod. Stas., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Baltimore; *for mail,* 1820 Mayfield Ave., Halethorpe 27, Md.
Keen, Robert Donald (J'43) (NDJLSZ), Mech. Engr., Design, Engrg. Products Co., 1600 S. San Pedro St., Los Angeles; *for mail,* 422 W. 79th St., Los Angeles 3, Calif.
Keen, W. Newlin (J'41) (YBNZ), Mech. Engr., Synthetic Rubber Technologist, Sales & Serv., E. I. du Pont de Nemours & Co., Box 525; *for mail,* R.D. 3, Wilmington, Del.
Keenan, Frank Thomas (J'38) (CHL), 3829 W. 64th Pl., Chicago, Ill.
Keenan, Joseph Henry ('26; '30; '34) (BKSAE), Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge 39; *for mail,* 48 Old Middlesex Rd., Belmont 78, Mass.
Keenan, Walter M. ('17; '28) (SFEKBO), Cons. Engr., 114 Liberty St., New York 6, N.Y.
Keenan, William Thomas, Jr. (J'41) (YDCSA), Salesman, Mech. Div., U.S. Rubber Co., 2801 Locust St., St. Louis 3, Mo.
Keene, Burton Freeman ('29; '35) (CFKSZ), Supvg. Field Engr., Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44, Pa.
Keene, J. Albert ('27; '43) (SFHC), Asst. Supt. Prod., Ala. Power Co., 600 N. 18th St., Birmingham 2, Ala.
Keener, H. James (J'35) (CDFMNS), Plant Engr., Conmar Products Corp. & Margon Corp., 140 Thomas St., Newark 1, N.J.
Keep, Henry ('44), Col., Corps of Engrs., U.S.A., Cons. Engr. to Post Engr., Charge Engrg. Sec., Utilities Dept., Middletown Air Depot; *for mail,* 143 Emaus St., Middletown, Pa.
Keep, Philip R. (J'39), Ensign, U.S.N.R., Assembly & Repair Dept., Naval Air Sta., Navy 1955, F.P.O., New York, N.Y.
Keese, David Leslie (J'45) (CMY), Y.M.C.A., 654 Bergen Ave., Jersey City 4, N.J.
Keesing, Maurice ('44), 47 Elm St., Elizabeth 3, N.J.
Keeth, Grover ('25) (FSEC), Ch. Engr., Marathon Corp., Rothschild, Wis.
Keeth, Jacob A. ('19; '25; '31) (SFK), Mgr., Power Prod., Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City 10, Mo.
Kego, Robert ('40) (BJM), Chas. Churchill & Co., Ltd., Coventry Rd., South Yardley, Birmingham, England.
Kehl, Robt. J. ('15; '23), Linde Air Products Co., 30 E. 42nd St., New York, N.Y.
Kehoe, Arthur Henry ('44), V.P., Charge of Constr. & Shops, Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Kehoe, Daniel Edward ('45) (JCD), Fdy. Supt., Savannah Mch. & Fdy. Co., Indian St.; *for mail,* 625 E. 53rd St., Savannah, Ga.
Keim, Charles J. (J'35) (EBH), Design Engr., Oil Well Supply Co., Oil City, Pa.
Keinath, George ('43) (ZMAB), Cons. Engr., Great Am. Industries, Inc., Meriden, Conn.; *for mail,* 36 Knollwood Dr., Larchmont, N.Y.
Keiser, Augustus Charles, Jr. (J'32) (EJMLSC), Lub. Engr., Tex. Co., Box 1722, Atlanta 1; *for mail,* 218 Bolling Rd., N.E., Atlanta, Ga.
Keith, Bruce Garth (J'42), Capt., Corps of Engrs., 16th Gp., Camp Claiborne, La.
Keith, Paul (J'43) (NJ), Engr., Harnischfeger Corp., 8 Lister Ave., Newark 8; *for mail,* 4910 Hudson Ave., West New York, N.J.
Keith, William Walton (J'40) (MBRSOF), 111 Seminole Dr., Marietta, Ga.
Keithley, Joseph Fraser (J'40) (OSACEH), Supt., Keithley Constr. Co., Box 32, Manitou Springs, Colo.
Kelcec, George (J'41) (SA), 2nd Lt., Army Air Forces, 498th Bomb. Gp., Clovis, New Mex.; *for mail,* 52 Embury Ave., Ocean Grove, N.J.
Keleher, John Edward ('43) (SLKDZF), Ch. Engr., Refined Syrups & Sugars, Inc., Foot of Vark St., Yonkers 1; *for mail,* 277 Cook Ave., Yonkers 2, N.Y.
Kelkar, Anand Mahadeo (J'39) (CDMLNZ), Mech. Engr., Premier Autos. Ltd., Wittet Rd., Bombay 1, India; *for mail,* 603 E. Madison Ave., Ann Arbor, Mich.
Kell, Waldo Ralph ('43) (CKMWYJ), 1007 W. 68th Terrace, Kansas City 5, Mo.
Kellem, Jacob Charles (J'45) (OFHBSE), Lt. (j.g.), 3rd Asst. Engr., U.S. Maritime Serv., 1716 N. 42nd St., Philadelphia 4, Pa.
Keller, Alfred R. ('39), 4 Randolph Rd., Worcester, Mass.
Keller, Allen ('41) (SACM), Steam Turbine Engr., Bldg. 2-64-G, River Works, Gen. Elec. Co., 920 Western Ave., Lynn, Mass.
Keller, Edwin (J'31) (MCBD), Prod. Engr., Owens-Ill. Glass Co., Sewell St.; *for mail,* 812 Lincoln Ave., Pitman, N.J.
Keller, Ernest George ('43), Airplane Div., Research Lab., Curtiss-Wright Corp., Columbus, Ohio.
Keller, Frederic John ('31) (LMS), Dist. Ch. Engr., Mgr., Food Specialties Div., Breyer Ice Cream Co., 3409 Queens Blvd., Long Island City 1, N.Y.
Keller, Geo. D. (J'37), Noble Hill, Abington, Pa.
Keller, George Ruland (J'41) (BMRNW), Devel. Engr., W. H. Wheeler, Inc., 800 E. 136th St., Bronx 54, N.Y.
Keller, Gilbert M. (J'40) (CASR), Steam Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Keller, Harry Hexamer ('45) (OCM), Comdr., U.S.N.R., Officer in Charge Constr., Asst. to Supvr. of Shipbldg., U.S.N., Cramp Shipbldg. Co., Philadelphia; *for mail,* 423 Newbold Rd., Jenkintown, Pa.
Keller, Henry Christian ('43) (DJM), Engrg. Mgr., Lamson Corp., Syracuse 1; *for mail,* 314 Haddonfield Dr., DeWitt, N.Y.
Keller, Henry G. ('37) (NJBLTD), Engr., Link-Belt Co., 2045 W. Hunting Pk. Ave., Philadelphia 40; *for mail,* 225 Brookdale Ave., Glenside, Pa.
Keller, Hixson Samuel (J'40) (SMCW), 630 W. Ganson Ave., Jackson, Mich.
Keller, John Donald ('41) (BKFJO), 1423 Farmers Bank Bldg., Pittsburgh 22, Pa.
Keller, John Morris (J'38) (SBN), Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 2609 N. 71st St., Wauwatosa 13, Wis.
Keller, Joseph F. ('08) (AHGMNY), Pres., Equi-Flow, Inc., 1841 Broadway; *for mail,* 325 West End Ave., New York 23, N.Y.
Keller, K. T. (A'16), Pres., Chrysler Corp., 341 Massachusetts Ave., Detroit, Mich.
Keller, Leonard Frederick (J'43) (CMSDLZ), Ch. Engr., Mosslinn Div., Chicopee Mfg. Corp. of N. H., Milltown; *for mail,* 59 High St., Metuchen, N.J.
Keller, Martin W. (A'19) (CNJMDB), V.P., Charge of Mfg., Oneida Ltd., Oneida, N.Y.
Keller, Ralph (A'45) (WCYDM), V.P., Gen. Mgr., Co. of Master Craftsmen, Inc.; *for mail,* 435 Broad St., Oneida, N.Y.
Keller, Richard D. (J'31) (BCM), Asst. Sales Mgr., Pratt & Whitney Div., Niles-Bement-Pond Co., West Hartford, Conn.
Keller, Robert Daniel (J'37), 1222 Seedeva St., Clearwater, Fla.
Keller, William McKinley ('45) (BJR), Gen. Foreman, Penn. R.R., c/o Mech. Engr., Rm. 515, 15 N. 32nd St., Philadelphia; *for mail,* 526 Glenwood Rd., Merion, Pa.
Kelley, Edward F. ('30) (LEMC), 54 Marion St., Brookline, Mass.
Kelley, F. W., Jr. (J'29), 267 Main St., Catskill, N.Y.
Kelley, Herschell Wade ('37; '42) (BJD), Design Engr., Union Bag & Paper Corp.; *for mail,* Box 1895, Savannah, Ga.
Kelley, Joseph Edwin (J'42) (NHCYKM), Design Engr., RCA Victor Div., Radio Corp. of Am., New Holland Ave., *for mail,* 637 N. Marshall St., Lancaster, Pa.
Kelley, Robert T. ('29), Reduction Dept., Anaconda Copper Min. Co., Anaconda, Mont.
Kellogg, Andrew Pierce ('39) (CSWK), Business Mgr., *Schenectady Union Star,* 207 Clinton St., Schenectady 5, N.Y.
Kellogg, Chever McMann ('41) (ACF), Mgr., Mfg. Economics, Shell Oil Co., Inc., 50 W. 50th St., New York; *for mail,* 26 Tunstall Rd., Scarsdale, N.Y.
Kellogg, George Dwight, Jr. (J'40) (MAB), Master in Sci., Physics & Mech. Drawing, Hotchkiss Sch., Lakeville, Conn.
Kellogg, Harry F. ('34) (CJM), Pres., Cent. Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Kellogg, Henry Birge (J'44) (A), Aviation Cadet, Army Air Forces, Sqd. 2, Tech. Sch. Sqd., Grand Rapids, Mich.
Kellogg, Morris Woodruff ('13) (CEFKLM), Pres., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.
Kellogg, Raymond Meserve ('15) (EF), Asst. Process Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Kellogg, W. D. ('32) (BMT), Pres., Gen. Mgr., Aveco, Inc.; *for mail,* 17 Coolidge Rd., Amsterdam, N.Y.
Kelly, Dennis H. (J'39) (LAMB), Tool Designer, Diesel Equip. Div., Gen. Motors Corp., S. Division Ave., Grand Rapids; *for mail,* 348 Logan, S.E., Grand Rapids 3, Mich.
Kelly, George Francis (J'39) (HBM), Mech. Engr., Design & Devel., Allied Control Co. Inc., 2 East End Ave., New York 21, N.Y.; *for mail,* 169 Ege Ave., Jersey City 4, N.J.
Kelly, Harry J. ('23) (CKO), Mgr., Engrg. Dept., Gulf Oil Corp., Box 1166, Pittsburgh 30, Pa.
Kelly, Howard Anthony ('38; '45) (MNSDCZ), Ch. Engr., Wheeling Stamping Co., 2100 Water St.; *for mail,* 16 Forest Rd., Oakmont, Wheeling, W.Va.
Kelly, James Lawrence (J'46) (ALZ), 4464 W. 69th Pl., Cleveland, Ohio.
Kelly, John Joseph, Jr. (J'43) (BHA), 803 Lincoln St., Evanston, Ill.
Kelly, John Patrick ('29) (BR), Retired; 30 George St., Pittsfield, Mass.
Kelly, Raymond E. (J'42) (NWYEH), Mech. Engr., U.S.N. Mine Countermeasures Sta., Panama City, Fla.
Kelly, Sidney John (J'45), Mech. Engrg. Instr., Bucknell Univ., Box 462, Lewisburg, Pa.; *for mail,* 1933 E. 33rd St., Brooklyn 10, N.Y.
Kelly, Thomas Conway ('14) (ABCJKL), Staff Engr., Vulcan Copper & Supply Co., 120 Sycamore St., Cincinnati 2; *for mail,* Apt. 318, 144 Glencoe Pl., Cincinnati 19, Ohio.
Kelly, Victor Quinton (J'44) (YSLMAC), Inner-Tite Corp., 18 W. Jersey St., Elizabeth; *for mail,* 371 Seymour Ave., Newark 8, N.J.
Kelsey, Elias Isbell ('39) (CMNJGD), Indus. Engr., 91 Maple St., Milford, Conn.
Kelsey, George Wright ('21; '30) (CZHM), Gen. Sales Mgr., Builders-Providence, Inc., 9 Codding St., Providence 1; *for mail,* 140 Brown St., Providence 6, R.I.
Kelsey, Harold D. ('30), Managing Engr., Aircraft Gas Turbine Div., Gen. Elec. Co., 920 Western Ave., West Lynn 3, Mass.
Kelsey, W. H. ('23) (BCDLMN), Ch. Engr., Combined Metals Reduction Co., 218 Felt Bldg., Salt Lake City 1, Utah.
Kelso, James Milton (J'37) (JKAZM), Research Engr., Harrison Radiator Div., Gen. Motors Corp.; *for mail,* 266 Pine St., Lockport, N.Y.
Kelting, Clarence A. ('38) (S), Asst. Div. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Kemler, Emory N. ('27; '33; 35) (NBSHEZ), Head, Engrg. Research Div., So. Research Inst., 917 S. 20th St., Birmingham 5, Ala.
Kemmer, Paul Hanes ('34) (ABEHJO), Col., Army Air Forces, Deputy Ch., Engrg. Div., Air Tech. Serv. Comd., TSENG-1, Lab. Area, Wright Field, Dayton, Ohio.
Kemmish, L. W. (J'35) (CSJF), Assoc. Engr., Prod. Dept., Seattle-Tacoma Shipbldg. Corp., 2400 H Ave., Seattle; *for mail,* 3607—35th Ave., S.W., Seattle 6, Wash.
Kemp, Harold Augustus ('20; '31) (H), Dir., Sanitary Engrg., Govt. of Dist. of Columbia, Dist. Bldg., Washington 4, D.C.
Kemp, James Battaile (J'39) (CTD), E. I. du Pont de Nemours & Co.; *for mail,* 305 Maple Ave., Waynesboro, Va.

Kemp, James Taylor ('37) (JSRHO), Tech. Advisor to Mission for Economic Affairs, U.S. Embassy, 1 Grosvenor Sq., London, W. 1, England; *for mail,* 1609—31st St., N.W., Washington 7, D.C.

Kemp, Lorin W. ('29) (CFSDJL), Plant Mgr., Internatl. Smelting & Refining Co., Elm St., Perth Amboy, N.J.

Kemp, Robert Bowers (J'44) (BOHCMK), Maj., Corps of Engrs., U.S.A., Hdq., 2nd Engr. Spec. Brigade, Camp San Luis Obispo, Calif.

Kemp, William Van Antwerp ('25) (CFSJLM), W. Parish Rd., Westport, Conn.

Kempe, Walter F. (J'43) (CKMN), Engr., U.S. Rock Wool Co., 40 S. Main St., Salt Lake City; *for mail,* 1094 E. 21st St. S., Salt Lake City 5, Utah.

Kendall, Edgar Homer ('16; '26) (ODHBMJ), Cons. Engr., 628 Vincent Blvd., Alliance, Ohio.

Kendall, Ernest Earl (A'40) (AHY), Owner, Engrg. Sales Co., 1044 N. Main St., Akron 10, Ohio.

Kendall, George Hayward ('37) (NCMJKZ), Cons. Mech. Engr., Box 72, Noroton Heights, Darien, Conn.

Kendall, H. Clayton ('30) (CHM), V.P., in Charge Prod. & Pressed Metal Sales, Rockwood Sprinkler Co., 38 Harlow St.; *for mail,* 32 Buckingham St., Worcester, Mass.

Kendall, Myron A. ('16), Ch. Engr., Stephens Adamson Mfg. Co.; *for mail,* 1211 Garfield Ave., Aurora, Ill.

Kendall, Norman R. (J'39) (BCD), Indus. Engr., Detailer, U.S. Gypsum Co., 300 W. Adams St., Chicago 6; *for mail,* 504 Wesley Ave., Oak Park, Ill.

Kending, Ernest Kaiser ('35) (FCJOL), 3301—164th St., Flushing, N.Y.

Kendrick, John F. ('26; '35) (BE), 16 E. Broad St., Columbus 15, Ohio.

Kendrick, Richard Alan (J'44), 2520 Montoya Ave., Albuquerque, New Mex.

Kendrick, Miss Ruth Ford (J'46), 2520 Montoya Ave., Albuquerque, New Mex.

Kenefick, John Cooper (J'43) (RSE), Lt. (j.g.), U.S.N.R., Ch. Engr., U.S.S. *Devosa,* AKA-27, F.P.O., New York; *for mail,* 366 Norwood Ave., Buffalo 13, N.Y.

Kenerson, William H. ('04; F'36), *Vice-President,* '22-'24; 100 Morris Ave., Providence, R.I.

Kenley, Brents E. (J'35), Lt. (j.g.), U.S.N.R., 500 Dune St., Norfolk 3, Va.

Kennard, Dwight Clinton, Jr. (J'41) (ABZ), Aero. Engr., Air Tech. Serv. Comd., Wright Field, Dayton; *for mail,* 1719—5th Ave., Dayton 5, Ohio.

Kennedy, Douglas Preston ('28) (CDHORS), Gen. Mgr., Compania Azucarera Dominicana, C. por A., Estate Consuelo, San Pedro de Macoris, D.R.

Kennedy, Edwin Coe (J'46), 45 New Lawn Ave., Arlington, N.J.

Kennedy, Geoffrey Farrer ('43) (HKLORS), Partner, Cons. Engr., Kennedy & Donkin, 12 Caxton St., London, S.W.1, England.

Kennedy, Grafton S. (J'17) (BCM), Col., Ord. Dept., Ord. Sec. Hdq., 9th U.S. Army, A.P.O. 339, c/o Postmaster, New York, N.Y.; *for mail,* Nantucket Island, Mass.

Kennedy, John C. ('30) (MJC), V.P., Secy., Works Mgr., Kennedy Valve Mfg. Co., E. Water St., Elmira, N.Y.

Kennedy, Matthew Edwin ('06; '30) (CM), Pres., Treas., Kennedy Valve Mfg. Co., 1021 E. Water St.; *for mail,* Strathmont Park, Elmira, N.Y.

Kennedy, Paul August, Jr. (J'37) (DJLMW), Prod. Supvr., Panel Dept., Wayne Mfg. Corp.; *for mail,* 404 Cherry Ave., Waynesboro, Va.

Kennedy, Paul S. ('26) (WY), V.P., Finishes Div., Internatl. Corp., 224 McWhorter St., Newark 1, N.J.

Kennedy, Robt. E. ('21) (DJM), Tech. Secy., American Foundrymen's Association, 222 W. Adams St., Chicago, Ill.

Kennedy, Thomas E. ('43) (LSMC), Ch. Project Engr., Walter Kidde Constructors, Inc., 140 Cedar St., New York, N.Y.

Kennedy, Wendell C. (J'40) (CDL), 285 Terrace Apts., Maryville, Tenn.

Kennedy, William Arthur ('08; '14) (AEJLRS), Supvr. of Products, Grinnell Corp., 260 W. Exchange St., Providence 1; *for mail,* 31 Forest St., Providence 6, R.I.

Kennedy, William Craig (J'41) (ACM), 1048 Rose Circle, Park Hills, Covington, Ky.

Kennedy, William Grady (J'42) (BAKHO), Mech. Engr., United Aircraft Research Dept., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 30 Wolcott Hill Rd., Wethersfield 9, Conn.

Kennedy, William M. ('30), 335 Lyndhurst Ave., Lyndhurst, N.J.

Kennedy, William P. ('13) (CEJO), V.P., Gen. Mgr., Brodie System, Inc., 115 Clifton Pl., Brooklyn, N.Y.

Kennedy, William R. (J'38) (CEP), Pilot, U.S.N., VRF-1, Naval Air Sta., New York, N.Y.

Kennedy, William Redding, Jr. (J'42) (AFMZ SC), Engrg. Officer, U.S.N.R., F.P.O., New York, N.Y.; *for mail,* 2138 Oglethorpe Ave., Augusta, Ga.

Kennedy, Winthrop Sargent (J'42) (J), Seaman, 2nd Class, U.S.N.-1, 6401 Darlington Rd., Pittsburgh 17, Pa.

Kenney, Clarence Edward ('45) (SNRAB), Design Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 2368 S. 56th St., West Allis 14, Wis.

Kenney, James T. ('27; '35; '35) (CLMJON), Plant Engr., Johns Manville Products Corp.; *for mail,* 135 North N St., Lompoc, Calif.

Kenney, Lewis Hobert ('04; '11) (BK), Retired; Univ. Club, 16th & Locust Sts., Philadelphia 2, Pa.

Kennison, George Foster (J'42) (BJMDHL), Ensign, U.S.N.R., 28 Byfield Rd., Waban 68, Mass.

Kennon, Lorenzo ('40) (BHJNO), Cons. Engr., 1927 Evergreen St., Burbank, Calif.

Kent, Clarence Hammond ('27) (BES), Chief, Engrg. Sec., Am. Univ., Biarritz, France; *for mail,* 676 Riverside Dr., New York 31, N.Y.

Kent, Frank Joseph (J'44) (BMNY), Ch. Draftsman, Raybestos-Manhattan, Inc., Passaic; *for mail,* 20 Ashton Pl., Glen Rock, N.J.

Kent, George Charles (J'43), Engrg. Instr., U.S. Armed Forces Inst. (Manila, P.I.), A.P.O. 75, San Francisco, Calif.; *for mail,* c/o K. Schulhof, 2100 Ridgeland Ave., Berwyn, Ill.

Kent, Gordon N. (J'40) (CLM), Mgr., Atlantic Bridge Co., Ltd., Box 230, Yarmouth, N.S., Can.

Kent, Henry R. ('01), Retired; Forest Beach Rd., South Chatham, Mass.

Kent, Herbert S. ('15; '18; '21), Ch. Draftsman, Continental Gin Co., 4500—4th Ave.; *for mail,* 1909 Wellington Rd., Birmingham 9, Ala.

Kent, Lawrason Riggs ('44) (SEFB), Mech. Engr., Bailey Meter Co., 503 City Centre Bldg., Philadelphia 7, Pa.; *for mail,* 3502 Dennlyn Rd., Baltimore 15, Md.

Kent, Nicholas (J'43) (ECM), Serv. Devel. Engr., Caterpillar Tractor Co., Peoria; *for mail,* 526 California Ave., Peoria 4, Ill.

Kent, Norman Williamson ('33; '43) (MCJD), Sales Engr., Petroleum Chem. Div., Organic Chem. Dept., E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington, Del.

Kent, Robert Sayre ('01; F'42) (LFKS), Pres., Robert Sayre Kent, Inc., 383 Pearl St., Brooklyn 1; *for mail,* 7616 Ridge Blvd., Brooklyn 9, N.Y.

Kent, Robert Thurston ('05; '19; F'43) (CJM), 60 Melrose Pl., Montclair, N.J.

Kent, Robert Willard ('17; '19; '24) (CMDO), Partner, Bigelow, Kent, Willard & Co., 75 Federal St., Boston 10, Mass.

Kent, Russell H. ('21; '35) (CLMST), Exec., Engr., Kent Mfg. Co., Clifton Heights; *for mail,* 431 Riverview Rd., Swarthmore, Pa.

Kent, S. Leonard, Jr. ('27) (CHS), 630 Winsford Rd., Bryn Mawr, Pa.

Kent, Sidney Louis (J'45) (HSOKAE), Geophysicist, Mech. Research Engr., Hans Lundberg & Co., 80 Richmond St. W.; *for mail,* 71 Forest Hill Rd., Toronto, Ont., Can.

Kent, Thomas Francis (J'45) (ESOKYA), Ensign, U.S.N.R., 25 Columbus Ave., North Easton, Mass.

Kentis, George E., Jr. (J'36) (BHM), Supt., Charge Mch. Tool Div., Yoder Co., W. 55th St. & Walworth Ave., Cleveland; *for mail,* 4000 W. 204th St., Fairview Village 16, Ohio.

Kenworthy, Joseph Miller, Jr. (J'39) (MCY), Stand. Methods Engr., Firestone Tire & Rubber Co., Pottstown; *for mail,* 4523 Pine St., Philadelphia 43, Pa.

Kenyon, James Maxwell (J'35) (MCJWYN), Dir., Ewbank Works, Entwisle & Kenyon, Ltd.; *for mail,* 9 Newton Dr., Accrington, Lancs, England.

Kenyon, Reid L. ('41) (BRNJZK), Assoc. Dir. of Research, Research Labs., Am. Rolling Mill Co., Middletown, Ohio.

Kenyon, Van Leslie, Jr. ('44) (SEFKMT), Instr., Duke Univ., Durham; *for mail,* R.F.D. 3, Hillsboro, N.C.

Kenyon, Walter Clifford, Jr. (J'42) (ABDS), Flight Research Engr., Sikorsky Aircraft Div., United Aircraft Corp., South Ave., Bridgeport; *for mail,* R.F.D. 2, Pine Rock Park, Shelton, Conn.

Keplinger, Charles Henry (J'32) (BCEFHJ), Keplinger & Wanenmacher, 329 Kennedy Bldg., Tulsa 3, Okla.

Keppel, Howard Bird ('29; '35) (F), Partner, Parry Engrg. Co., 154 Nassau St., New York 7, N.Y.

Keppelman, Henry Schroeder ('45) (CJR), V.P., O. C. Duryea Corp., 1601 Mercantile Trust Bldg., Baltimore 2, Md.

Keppler, Paul William (J'26) (ZSONLF), Research & Devel. Engr., Foxboro Co., Neponset Ave.; *for mail,* 91-A Main St., Foxboro, Mass.

Ker, Alan Ward (J'45) (EJSZB), 1537 N. La Salle St., Indianapolis 1, Ind.

Kerbey, Eric A. ('30; '35) (SECJLH), Exec. V.P., Midwest Piping & Supply Co., Inc., 1450 S. 2nd St., St. Louis 4, Mo.

Kerby, Wilford Dixon, Jr. (J'43) (RKF), Field Engr., West. Union Tel. Co., Rm. 417, West. Union Bldg., Marietta, Atlanta, Ga.

Kerchner, Charles E. ('23; '35) (FSTEKO), Supt. of Power, Proximity Mfg. Co.; *for mail,* 1206—14th St., Greensboro, N.C.

Kerkau, Arthur D. (J'39) (CSZ), 1031 Borton Ave., Essexville, Mich.

Kerker, Henry F. ('28; '35) (LKNYDZ), Sales Engr., Buflovak Equip. Div., Blaw-Knox Co., 1543 Fillmore Ave., Buffalo 11; *for mail,* 35 Fernhill Ave., Buffalo 15, N.Y.

Kermer, Martin Jack ('14) (BCKL), Ch. Engr., Evaporator Dept., Buflovak Equip. Div., Blaw-Knox Co., Ferry St., Buffalo; *for mail,* 251 Comstock Ave., Buffalo 15, N.Y.

Kern, Frank, Jr. (J'39), Partner, Charge Shop & Engrg., Metal Parts Mch. Co., 1742 Rivard St., Detroit 7, Mich.

Kernander, Warren Norman (J'44) (MNBEHK), Ensign, U.S.N., Naval Mine Depot, Yorktown, Va.

Kerner, Joseph R. ('43) (SCEAKZ), Cent. Sta. Div., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.

Kerner, Leo C. ('28), V.P., Natl. Carloading Corp., 19 Rector St., New York, N.Y.

Kerns, Clinton Bowen ('37) (CDM), Box 348, Lake Wales, Fla.

Kerns, Robert Gene (J'44), Box 445, Priest River, Idaho.

Kerr, Arthur John ('21; '31) (CE), *Manager,* '44-'45, *Director-at-Large,* '45-'47; V.P., Charge of Sales, Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh, Pa.

Kerr, C. Phillips ('16; '28), Lt. Col., Gen. Staff Corps, Hdqs., II Army Corps, Wilmington, Del.

Kerr, Charles Volney ('92) (BHMOSY), 418 Ulysses St., Los Angeles 31, Calif.

Kerr, David Cushing (J'40) (CJ), 959 Oberlin Ave., Lorain, Ohio.

Kerr, Edward ('28), V.P., Davey Co., Downingtown, Pa.

Kerr, Eugene Wycliff ('02; '08) (FSKLOZ), Head, Firm of E. W. Kerr, Cons. Engr., 808 Lake Pk., Baton Rouge 8, La.

Kerr, Henry Hamilton ('25) (SFC), V.P., Charge of Operas., Toledo Edison Co., Edison Bldg.; *for mail,* 2365 Barrington Dr., Toledo 4, Ohio.

Kerr, Henry Kirby (J'36) (CFW), Sub-Lt. (E), H.M.C.S., *Ste. Therese,* F.M.O., Halifax, N.S., Can.

Kerr, John Hammond (J'46) (BC), Engr., E. I. du Pont de Nemours & Co., 626 Schuyler Ave.; *for mail,* 73 Belgrove Dr., Arlington, N.J.

Kerr, John Latta ('44) (SKCO), Sales Mgr., Turbine Div., Gen. Elec. Co., 970 Western Ave., West Lynn, Mass.

Kerr, John Russell (J'40) (OKDCSL), Asst. Branch Engr., York Corp., 5051 Santa Fe Ave., Los Angeles; *for mail,* 7109 Marconi St., Huntington Park, Calif.

Kerr, S. Logan ('21; '26; '34) (HOBCMZ), *Junior Award,* '21; Pres., S. Logan Kerr & Co., Inc., Bankers Security Bldg., Walnut & Juniper Sts., Philadelphia 7; *for mail,* 36 Westview St., Philadelphia 19, Pa.

Kerr, Thomas Herron ('17; '21) (BFO), Assoc. Prof., Engrg. Exper. Sta., Ohio State Univ., Columbus 10; *for mail,* 1753 Bedford Rd., Columbus 8, Ohio.

Kerr, Wallace E. ('29; '35) (JNB), Cons. Engr., 310 W. Madison Ave., Youngstown 4, Ohio.

Kerr, William A. ('45) (CZ), Sales Mgr., Instrument Div., Am. Optical Co., 14 Mechanic St., Southbridge; *for mail,* 348 Payson Rd., Belmont 78, Mass.

Kerrick, Joseph Homet ('40) (FSDR), Fuel Engr., Utilization Serv., Philadelphia & Reading Coal & Iron Co., Reading Terminal Bldg., Philadelphia 5; *for mail,* 925 E. Dorset St., Philadelphia 19, Pa.

Kershaw, Stephen Emeric ('42; '45) (COHMD), Owner, Indus. Am. Consultants, 21 Portage Ave., Detroit 3, Mich.

Kershner, Stuart G. (J'37) (CEH), Asst. to Mgmt., Tex. Pipe Line Co., Box 2420, Tulsa, Okla.

Kertesz, Carl (J'44) (BCM), Pres., Die-Jig Corp., 80 Washington St., New York; *for mail,* 80-44 —190th St., Jamaica, L.I., N.Y.

Kerzel, August (J'32) (NDWOMJ), Design Engr., Willamette Iron & Steel Corp., 3060 N.W. Front Ave., Portland; *for mail,* 1735 S.E. 11th Ave., Portland 14, Ore.

Kessinger, Jackson Arthur (J'43) (AK), 9502 Old Bladensburg Rd., Silver Spring, Md.

Kessler, Armin George ('09; '25) (CEJMOY), V.P., Farrel-Birmingham Co., Inc., 2710—1st Cent. Tower, Akron, Ohio.

Kessler, Henry Reginald ('22; '26; '35) (SFL), Mgr., Republic Flow Meters Co., 420 Lexington Ave., New York 17; *for mail,* 6779 Dartmouth St., Forest Hills, N.Y.

Kessler, Herbert H. ('15; '20; '27) (LDEFSR), Basic Industries Specialist, Allis-Chalmers Mfg. Co., Broad St. Sta. Bldg., Philadelphia 3; *for mail,* 406 Old York Rd., Abington, Pa.

Kessler, Martin ('44) (ABCJNY), Research Physicist, Pollak Mfg. Co., 541 Devon St., Arlington; *for mail,* 106 Wildwood Ave., Upper Montclair, N.J.

Kesti, Andrew Ilmar (J'41) (MCNEAD), Process Engr., Continental Motors Corp., 305-15 Market St.; *for mail,* 512 Houston Ave., Muskegon, Mich.

Ketcham, Christian B. ('43) (CDSZ), Branch Mgr., Westinghouse Elec. Corp., 207 W. 3rd St., Cincinnati 2; *for mail,* 2747 Werkastle Lane, Cincinnati 11, Ohio.

Ketcham, Henry Hendricks ('28) (MRW), Box 221, Piqua, Ohio.

Ketchel, John Raymond ('44) (CFKS), Gen. Supt. of Prod., Jersey Cent. Power & Light Co., 501 Grand Ave., Asbury Park; *for mail*, Box 159, South Amboy, N.J.
Ketchpel, Paul A. ('36) (CHNOR), Owner, Mgr., Ketchpel Engrg. Co., 1401-5 Palisades Ave.; *for mail*, 291 Winthrop Rd., West Englewood, N.J.
Ketchum, Eugene Rupert ('44) (SBEHK), Mech. Engr., Dow Chem. Co.; *for mail*, 1322 W. 5th St., Freeport, Tex.
Ketchum, Gardner Mason (J'44) (SEKHBF), Instr., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Ketchum, Kenneth Woodrow (J'42) (MHJ), Westinghouse Elec. Co. of Brazil, Caixa Postal 1320, Rio de Janeiro, Brazil, S.A.
Ketchum, Samuel ('08; '17) (BKHMCR), Engrg. Dept., Carrier Corp., S. Geddes St., Syracuse 1; *for mail*, 128 Twin Hills Dr., Syracuse 4, N.Y.
Ketler, Charles Pletz ('42) (EAMCH), 117 Maple Ave., Rockville Centre, N.Y.
Ketner, Donald Morrow (J'46), Tenn. Eastman Corp., Oak Ridge; *for mail*, 914 Sterling Ave., Chattanooga, Tenn.
Keto, Jorma Raymond (J'40) (MNDC), Mech. Engr., Design Dept., Coldwell-Philadelphia Lawn Mower Div., Portable Products Corp., Newburgh, N.Y.
Ketter, H. Edward (J'40) (DFS), Ensign, U.S.N.R.; Box 385, Gary, W.Va.
Kettering, Charles F. ('15) (AEFRSY), *A.S.M.E. Medallist*, '40; Gen. Mgr., Research Labs. Div., Gen. Motors Corp., 485 Milwaukee Ave., Detroit 2, Mich.
Kettler, Jack Raymond (J'44), Calif. Inst. of Tech., 1201 E. California St., Pasadena; *for mail*, Box 402, Route 1, Huntington Beach, Calif.
Keuffel, Adolf W. ('30) (JMW), Asst. Secy., Keuffel & Esser Co., 300 Adams St., Hoboken, N.J.
Keuffel, Carl W. ('13; '26), Exec., Mfg. Dept., Keuffel & Esser Co., 300 Adams St., Hoboken, N.J.
Keusch, Richard Buck (J'40) (BCJ), Box 92, Riverdale Sta., Dayton, Ohio.
Keyak, Karl S. (J'40) (HOKNSM), Supvr., Design Scien. Gp., Design Sec., Naval Shipyard, San Francisco 24, Calif.
Keyes, Frederic Hale ('02) (CGRWM), Admin. Asst., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39; *for mail*, 73 Elm Rd., Newtonville 60, Mass.
Keyes, Henry Morgan (J'36) (BNKEHS), N. Main St., Stony Creek, Conn.
Keyes, John Humphrey ('30; '35) (MNCA), Design & Prod. Engr., Shaffer Mfg. Co., Box 3126, Tulsa 8, Okla.
Keys, Douglas Leighon ('19; '25) (BCHM), Lub. Engr., Tex. Co., 135 E. 42nd St., New York 17; *for mail*, 1966 E. 23rd St., Brooklyn 29, N.Y.
Keyserling, Ben Hirsch (J'37) (CJMS), Maj., U.S.A., Gen. Supply Officer, Erie Ord. Depot, Lacarne, Ohio; *for mail*, 1 S. Crest Rd., Chattanooga 4, Tenn.
Kezios, Stothe Peter (J'43) (KHBASE), Ensign, Civil Engr. Corps, U.S.N.R., 31st Naval Constr. Bn.; *for mail*, 5759 W. Dakin St., Chicago 34, Ill.
Kiachif, Messoud (J'39) (TLNYBM), Mech. Engr., Spruance Plant, E. I. du Pont de Nemours & Co., Richmond; *for mail*, 805 W. Franklin St., Richmond 20, Va.
Kice, Murray Stancliffe ('43) (LOSCJB), Ch. Engr., Am. Blower Corp., Detroit 32, Mich.
Kiclanes, Charles E. (J'42) (MJSAFH), Lt. (j.g.), U.S.N.R., LST-41, F.P.O., San Francisco, Calif.; *for mail*, 11 S. Austin Blvd., Chicago 44, Ill.
Kidd, Alexander ('30; '34) (BDJMCK), V.P., Charge of Mfg., M. W. Kellogg Co., Danforth Ave., Jersey City 3, N.J.
Kidd, George Edwin ('42) (BKLNYE), Ch., Design Sec., Field Research Dept., Magnolia Petroleum Co., Box 900, Dallas 1, Tex.
Kidd, Geo. F. ('15) (DES), Mech. & Struc. Engr., 61 Hollywood Ave., East Orange, N.J.
Kidder, Wilbur Elwyn (J'34) (CEM), Personnel Mgr., Behr-Manning Corp., Troy, N.Y.
Kidwell, Guy S., Jr. (J'43) (AEFZGB), 1st Lt., Army Air Forces, B-17 Airplane Comdr., Box 440, Army Air Field, Roswell, New Mex.; *for mail*, 45 E. Irvin Ave., Hagerstown, Md.
Kieckhefer, Robert John, Jr. (J'45) (NWHC), Yeoman, U.S.N.R., Personnel Separation Center, Bldg. 1200, Great Lakes, Ill.; *for mail*, Brookfield, Wis.
Kiefer, Carl J. ('20), V.P., Charge Prod., Schenley Distillers Corp., 26 E. 6th St., Cincinnati, Ohio.
Kiefer, Paul James ('15; '20) (BEHKS), Sr. Prof. Mech. & Mar. Engrg., U.S.N. Postgraduate Sch., Annapolis, Md.
Kiefer, Paul Walter ('39; F'45) (RYZEFK), Ch. Engr., M.P. & Rolling Stock, N.Y. Cent. System, Rm. 1462, 466 Lexington Ave., New York 17, N.Y.
Kiehnle, William Albert ('29; '35) (O), Secy., William V. Kiehnle Co., 3606 Park Ave., New York 56, N.Y.
Kielczewski, Edward Louis (J'44) (CBMJNL), Tool Designer & Checker, Krautter Weber Tool Co., 226 Morris Ave., Newark 3; *for mail*, 12 Chestnut St., Kearny, N.J.
Kienholz, Robert Andrew (J'39) (BCSLN), Designer, E. I. du Pont de Nemours & Co., Wilmington 98, Del.; *for mail*, 6314 Haverford Ave., Philadelphia 31, Pa.
Kiernan, Bernard Lacoste, Jr. (J'42), Kay Engrg. Serv., 202 Interstate Bank Bldg., New Orleans 12, La.
Kiernan, Francis Raymond (J'35) (CMODSW), Comdr., U.S.N.R., Bur. of Ord., Navy Dept., Washington 25, D.C.; *for mail*, 1814 Abingdon Dr., Alexandria, Va.
Kierulff, John Lorenz (J'43) (DSFH), Matls. Engr., Bonneville Power Admin., 1300 N.E. Union Ave., Portland; *for mail*, 3333 N.E. 60th Ave., Portland 13, Ore.
Kiesel, George Frederick, Jr. (J'43), 3251 Hawthorne, St. Louis, Mo.
Kiesel, John S. ('27) (CJBHMY), Cons. Engr., 1220 Woodmont Ave., Williamsport 16, Pa.
Kiesel, William F., Jr. ('07) (R), Retired; 909 Penn. St., Hollidaysburg, Pa.
Kieselbach, Henry August ('14; '29), Gen. Mgr., Insulation Dept., Johns-Manville Sales Corp., 22 E. 40th St., New York 16, N.Y.; *for mail*, 43 Myrtle Ave., Montclair, N.J.
Kievit, William Martin (J'42) (CMRA), Engrg. Metal Inspr., Stand. Oil Co. of N.J., Box 16, Elizabeth; *for mail*, 1134 Stiles St., Linden, N.J.
Kiewit, Alfred L. (J'40), Research Engr., Cincinnati Gas & Elec. Co., 4th & Main; *for mail*, 4326 Ridgeview Ave., Cincinnati 5, Ohio.
Kihn, Wm. J. ('20; '35), R.F.D. 3, Morristown, N.J.
Kilbourne, Hubert G. ('42) (HM), 176 Federal St., Boston, Mass.
Kilby, Hubert S. ('42) (CES), V.P., West. Light & Tel. Co., Inc., Box 763, Great Bend, Kan.
Kilcoin, Frank B. (J'43) (JSA), Rm. 706, Y.M.C.A., Ft. Wayne, Ind.
Kilduff, Frederic William (A'39) (CMDA), Sr. Administrator, Bankers Trust Co., 16 Wall St., New York; *for mail*, 61 W. 9th St., New York 11, N.Y.
Kilgore, Rather Bowline, Jr. (J'39) (LCF), Project Engr., Stand. Oil Co. (Ohio), Midland Bldg., Cleveland 15; *for mail*, 3548 W. 129th St., Cleveland 11, Ohio.
Kilgore, Russell Ward (J'38) (RE), Project Engr., Electro-Motive Div., Gen. Motors Corp., La Grange; *for mail*, 1834 S. 2nd Ave., Maywood, Ill.
Killam, Kenneth Amasa ('39) (ONMBCJ), Pres., Mgr., Sterling Engrg. Co., Inc., 90 Mill St.; *for mail*, 22 Fenton Ave., Laconia, N.H.
Killinger, Clarence E. ('44), Maint. Supt., Dept. 36, Bell Aircraft Corp., Bellcraft; *for mail*, 1608 Hillside Dr., Marietta, Ga.
Killmer, Herbert Phill (J'44), 1634½ S. Williams St., South Bend, Ind.
Killoran, Clarence Henry (J'45) (HOS), Jr. Engr., H. G. Acres & Co., 2135 Culp St.; *for mail*, 1902 Drummond Rd., Niagara Falls, Ont., Can.
Kilner, Donald Dee (J'42) (BEANSZ), Sr. Technician-Engr., Cleveland Diesel Eng. Div., Gen. Motors Corp., 2160 W. 106th St., Cleveland; *for mail*, 13968 Lake Ave., Lakewood 7, Ohio.
Kilpatrick, A. Vern ('38) (EHM), Assoc. Prof. Mech. Engrg., Mo. Sch. of Mines & Metal., Rolla, Mo.
Kilroy, Michael J. (J'37) (FKS), Lt., U.S.N.R., 3311 Windsor Ave., Kansas City, Mo.
Kimball, C. Winston (J'38), 481st Ord. Co., Hickam Field, T.H.
Kimball, Dexter Simpson ('00; F'36; H'39) (BC), *Manager*, '19-'21, *President*, '22; *Worcester Reed Warner Medallist*, '33; Belleayre Apts., Ithaca, N.Y.
Kimball, Dexter Simpson, Jr. ('28; '39) (CMDJ), Factory Mgr., Bendix-Westinghouse Automotive Air Brake Co., 901 Cleveland Rd., Elyria, Ohio.
Kimball, Edwin Elliot ('42) (RS), Retired; 903 Bedford Rd., Schenectady 8, N.Y.
Kimball, Henry Baldwin (J'35) (MCBA), 8 Jefferson St., Marblehead, Mass.
Kimball, Richard David ('44) (CFS), Plant Engr., Maint., New Britain Mch. Co., New Britain, Conn.
Kimball, Richard Hayes, Jr. (J'42) (WZJHDT), Ensign, O-V(S), U.S.N.R., 71 Union St., Bridgewater, Mass.
Kimball, Robert Schaeffer (J'36) (DKL), 1st Lt., Corps of Engrs., Hdq., 1777th Engr. Combat Bn., Ft. Lewis, Wash.
Kimber, Harry A. ('21), Dist. Sales Agt., Sims Co., Erie, Pa.; *for mail*, 33 Elk Ave., New Rochelle, N.Y.
Kimberlin, Paul Hubert (J'30) (KFSCWY), Maint. Supvr., K. M. & S. Co., 2229 Ave. F; *for mail*, Box 1286, Bay City, Tex.
Kimbrell, Jack Theodore (J'44), 1820—74th Ave., Elmwood Park, Ill.
Kimbrough, Jacob C. ('37; '43) (NJMBD), Mech. Engr., Am. Lava Corp., Manufacturers Rd., Chattanooga 5; *for mail*, 220 Hillcrest Ave., Chattanooga 4, Tenn.
Kimel, William R. (J'44), 335 E. Tallmadge Ave., Akron 10, Ohio.
Kimmel, Alfred William ('16; '19; '35) (CDL), Owner, Ready Mixed Corp., 20 Keowee St., Dayton 2, Ohio.
Kimmell, Phillip Michael ('43) (DSKLCH), Mech. Engr., Hawk-Eye Works, Eastman Kodak Co., 1447 St. Paul St., Rochester 5; *for mail*, 33 Stonecliff Dr., Rochester 12, N.Y.
Kimmich, Herman Siegfried (J'44) (MCBN), Tool Designer, Keuffel & Esser Co., 3rd & Adam Sts., Hoboken; *for mail*, 325—6th Ave., Newark 7, N.J.
Kimmlinger, Kenneth Thomas ('45) (SFKB), Sales Engr., M. H. Detrick Co., 111 W. Washington St., Chicago 2; *for mail*, 5044 Berenice Ave., Chicago 41, Ill.
Kincade, Elmer C. ('13; '35) (CEP), V.P., Gulf Refining Co., Box 2100, Houston, Tex.
Kincaid, Leroy Miller (A'44) (CMTB), Mfrs. Rep., Power Plant Equip., 1117 Liberty Life Bldg., Charlotte; *for mail*, Box 116, Erlanger, N.C.
Kind, Daniel A. (J'42) (CDM), 911 W. 29th St., Wilmington, Del.
Kinderman, Walter J. ('41) (Z), Research Engr., Yarnall-Waring Co., Mermaid Lane, Chestnut Hill, Philadelphia 18; *for mail*, 109 W. Nippon St., Philadelphia, Pa.
Kindermann, Wilfred Julius (J'45) (NYZJC), Mech. Engr., Bell Tel. Labs., Inc., 463 West St., New York 14; *for mail*, 7 LaBelle Rd., Mt. Vernon, N.Y.
Kindl, Fred Henry, Sr. ('45) (ECKMZH), Ch. Engr., Am. Dist. Steam Co.; *for mail*, 809 Goundry St., North Tonawanda, N.Y.
Kindler, Ralph August (J'43) (ACW), 16350 Nordhoff St., San Fernando, Calif.
King, Alfred Taylor ('40) (GHN), Supt., Holt-Weld, 647 N. Cannon Ave.; *for mail*, 615 York Ave., Lansdale, Pa.
King, Arthur Charles ('38) (EFKOSZ), Cons. Engr., 35 S. Dearborn St., Chicago 3, Ill.
King, Charles Burton ('12) (CDJMN), Retired; 725 Westholme Ave., Los Angeles 24, Calif.
King, Charles F. (J'39) (ZLEOHJ), Project Engr., Fisher Governor Co., Marshalltown, Iowa.
King, Douglas Henry (J'45) (BCJ), Capt., "Aingarth," Muston Rd., Filey, East Yorkshire, England.
King, E. Ross (J'44) (CJL), 30 Penrose St., Quakertown, Pa.
King, Edward L., Jr. (J'39) (AEH), Lt., U.S.N.R., Acorn 45, F.P.O., San Francisco, Calif.
King, Frederick James ('27), Ch. Engr., Linde Air Products Co., 30 E. 42nd St., New York 17, N.Y.
King, George Ilgenfritz ('01) (BR), 15 Stratford Rd., Brooklyn 18, N.Y.
King, Harold Merriam ('14; '35) (SCEA), Asst. Engr., Steam Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail*, 90 Walker Rd., Swampscott, Mass.
King, Henry F. ('33; '35) (CM), Ch. Engr., Tech. Devel. Div., Sears, Roebuck & Co., Homan Ave., Chicago, Ill.
King, James Edward (J'43), Lt. (j.g.), U.S.N.R., 6640 Marietta St., Pittsburgh 6, Pa.
King, Joe Judson ('28; '37) (AE), Texas Co., Box 2332, Houston, Tex.
King, John Andrew ('25; '35) (SFJKLE), Dist. Refractory Engr., Carborundum Co., Box 268, Perth Amboy, N.J.; *for mail*, Box 744, Worcester 1, Mass.
King, John Aubrey ('23; '28; '31) (SBFKE), Prof. & Head, Mech. Engrg. Dept., College of Applied Sci., Syracuse Univ., Syracuse 10, N.Y.
King, John Chilton (J'42) (AM), Engrg. Designer B, No. Am. Aviation, Inc.; *for mail*, 3536 W. 112th St., Inglewood, Calif.
King, John Payne ('45) (CJM), Treas., Alfred B. King & Co., Blatchley Ave. & River St., New Haven 13, Conn.
King, Marcello Aberdeen ('19; '21; '35) (CSJNB), Exec. Engr., Moore Steam Turbine Div., Worthington Pump & Mchy Corp., Wellsville, N.Y.
King, Mou-Hui (J'44) (CFHKS), Engr., Chinese Supply Comm., 2311 Massachusetts Ave., Washington 8, D.C.
King, Norman M. ('17; '22), 9949 Shore Rd., Brooklyn 9, N.Y.
King, Peter Monk ('19; '35) (HSFBEN), Retired; 5337 Coliseum St., Los Angeles 16, Calif.
King, Ralph Marsh (J'27) (CONMD), Ch. Field Engr., Layne-Cent. Co., Box 215, Hollywood, Memphis 8; *for mail*, 963 Maury, Memphis 7, Tenn.
King, Roy Stevenson ('04; '10) (FKSREM), Prof. & Head, Mech. Engrg. Dept., Ga. Sch. of Tech.; *for mail*, 1293 Oxford Rd., N.E., Atlanta, Ga.
King, Russell Noel (J'40) (NMJCBH), Mech. Engr., Nash Engrg. Co., Wilson Rd.; *for mail*, Richmond Hill Rd., South Norwalk, Conn.
King, Samuel Leo, Jr. (J'41) (ACY), 158 W. Main St., Elkton, Md.
King, Thomas Benton (J'41) (BYH), Research & Devel. Engr., Winters, Crampton Corp., Grandville, Mich.
King, Vernon C. ('18; '24), Indus Engr., Wickwire Spencer Steel Co. Div., Colo. Fuel & Iron Corp., 56 Sterling St., Clinton; *for mail*, 20 Englewood Ave., Worcester 3, Mass.
King, William Julian ('44) (EFSARC), *Melville Medallist*, '45; Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
King, William O'Connor (J'43), Mech. Engr., Testing, Natl. Bur. of Stands., Connecticut Ave. at Upton, N.W., Washington, D.C.; *for mail*, 8405—16th St., Silver Spring, Md.
King, Williams Vincent ('30; '37) (SOFECH), Project Mgr., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.
Kingman, William W. ('39) (DCSHME), V.P., Treas., Edgar Bros. Co., Metuchen, N.J.

Kingsbury, J. Grant ('06) (BHM), Pres., Grant Mfg. & Mch. Co., 90 Silliman Ave., Bridgeport, Conn.; *for mail*, Box L-1, Fall Brook, Calif.
Kingsbury, Robert Clair (J'38) (MNC), Sales Engr., Dodge Mfg. Corp., Mishawaka; *for mail*, 1402 Kessler Pl., South Bend 16, Ind.
Kingston, Victor Michael (J'38) (HKSAF), 4907 S. Hoover St., Los Angeles 37, Calif.
Kinkead, Robert Emerson ('27) (BKO), Cons. Engr., 3441 Lee Rd., Cleveland 20, Ohio.
Kinker, Clarence Carl ('44) (BCLMNY), Ch. Engr., Defiance Mch. Works, Inc.; *for mail*, 205 Jefferson, Defiance, Ohio.
Kinley, Jack (J'38) (MNC), Asst. Mgr., M. M. Kinley Co., Box 6177, Houston; *for mail*, 1922 Olympia, Houston 6, Tex.
Kinnard, James A. ('29), Cons. Engr., 11-269 Gen. Motors Bldg., Detroit 2, Mich.
Kinnavy, Martin Gerald (J'44) (ABS), 3331 N. Harding Ave., Chicago 18, Ill.
Kinne, Clarence Evelyn ('99) HNS), Retired; 116 S. Indiana Ave., Watertown, N.Y.
Kinney, Aldon Monroe ('35) (CELMOS), Pres., A. M. Kinney, Inc., 1211 Enquirer Bldg., Cincinnati 2, Ohio.
Kinney, David Whittemore (J'42), 2 Woodland Rd., Malden, Mass.
Kinney, James Joseph ('41) (NMBJ), Ord. Engr., Sec. 3, Design, U.S. Naval Gun Factory, Admin. Bldg., Washington, D.C.
Kinney, Joseph N. (A'15) (DJY), V.P., Kinney Motors, Inc., 1001 Quentin Rd., Brooklyn, N.Y.; *for mail*, 92 Midland Ave., Montclair, N.J.
Kinney, Watson Fredericks (J'40) (SK), Lt. Comdr., U.S.N.R., Officer-in-Charge, Crossroads Project, Naval Engrg. Exper. Sta., Annapolis, Md.; *for mail*, 16335 Lyndon St., Detroit 27, Mich.
Kinney, William Shelby ('43) (SFHOKZ), Mech. Engr., Power Dept., Schenley Distillers Corp., 26 E. 6th St., Cincinnati 2, Ohio.
Kinnison, Court ('36; '45) (SNDELM), Indus. Engr., 119 Kansas St., San Francisco, Calif.
Kinsman, Richard Ellard ('14; '25) (AEW), 3 Strathmont Pk., Elmira, N.Y.
Kinzbach, Robert Benton ('45), V.P., Kinzbach Tool Co., Inc., Box 312, Houston 1, Tex.
Kinzie, Phillip Alexander ('43) (HAREMS), Littlerock, Calif.
Kipp, Harold Lyman (J'37) (KBNEAS), Prof. Mech. Engrg., Tex. Tech. College, Lubbock, Tex.
Kippenhan, Charles Jacob (J'42) (BKHSEL), Lt., U.S.N.R., 1607 Muscatine Ave., Iowa City, Iowa.
Kiproff, Peter (J'43) (ASH), Aero. Engr., Aero. Eng. Lab., Naval Air Matl. Center, Naval Shipyard, Philadelphia 12, Pa.; *for mail*, 20 E. Cedar St., Mt. Vernon, N.Y.
Kirby, Cornelius C. (J'37) (SFEL), Engr., Power Plants, Chem. Constr. Corp., 350—5th Ave., New York 1, N.Y.
Kirby, William Colvin ('38) (KS), Contrg. Engr., Grinnell Co., Inc., 240 N. Highland St., N.E., Atlanta 2, Ga.
Kirby, William Kenneth ('38) (EFKJ), Cons. Petroleum Engr., 1964 Glenview Terrace, Altadena, Calif.
Kirchner, Carl Otto ('41) (ACM), Mch. Shop Foreman, Leggett & Platt Spring Bed & Mfg. Co.; *for mail*, 114 N. Garrison Ave., Carthage, Mo.
Kirgan, John Francis ('19; '21; '27) (KHBCSM), Cons. Engr., Heat Transfer Products, Inc., Box 516, Carbondale, Pa.; *for mail*, R.D. 1, Columbia, N.J.
Kirk, George Leon ('30) (NMZCB), Ch. Engr., Elgin Natl. Watch Co., 107 National St.; *for mail*. 1125 Bellevue Ave., Elgin, Ill.
Kirk, Walter Blanchard ('45) (BEJLYZ), Engr. of Tests, Westinghouse Air Brake Co., Wilmerding, Pa.
Kirkby, Norman Otto (J'45) (RBNCKM), Asst. to Mech. Engr., Chicago, Burlington & Quincy R.R., 547 W. Jackson Blvd., Chicago 6, Ill.
Kirkland, Randolph Withers, Jr. (J'41) (AES), Lt., Hdq. Co., 2nd Bn., 511th Parachute Inf., A.P.O. 468, c/o Postmaster, San Francisco, Calif.
Kirkpatrick, Alton ('18; '22) (SLFOK), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston, Mass.; *for mail*, 671 S. Coronado St., Los Angeles 5, Calif.
Kirkpatrick, Edward Marshall ('41) (ARS), Dist. Turbine Specialist, Gen. Elec. Co., 4966 Woodland Ave., Cleveland 4, Ohio.
Kirkpatrick, Floyd Major ('45) (MJ), Asst. to Supt., Gen. Elec. Co., 2901 E. Lake Rd.; *for mail*, 724 E. 29th St., Erie, Pa.
Kirkpatrick, James Wellington (J'44) (AFNC JH), Engr., Aircraft Power Plants, Transcontinental & West. Air, Inc., 10 Richards Rd., Mun. Airport, Kansas City; *for mail*, 3729 Summit St., Kansas City 2, Mo.
Kirkpatrick, Ruel Lee (J'38) (HBEM), Field Engr., Clark Bros. Co., Inc., Olean, N.Y.; *for mail*, 1923 Brun Ave., Houston, Tex.
Kirkup, Joseph Parry ('08; '12) (CFSMO), 11 Murray Ave., Port Washington, N.Y.
Kirkup, Richard Paynter (J'45), 11 Murray Ave., Port Washington, N.Y.
Kirkwood, Arthur Carter ('23; '36) (ESOFHK), Assoc. Engr., Burns & McDonnell Engrg. Co., 107 W. Linwood Blvd., Kansas City 2; *for mail*, 5632 Charlotte St., Kansas City 4, Mo.
Kirloskar, S. L. ('27; '37) (CEH), Deputy Gen. Mgr., Kirloskar Bros., Ltd., Kirloskarvadi, Dist. Satara, India.

Kirsch, Carl W. ('30; '35) (BCLMKJ), Tool Engr., Hoover Co., North Canton; *for mail*, 1 Superior Apts., 817—5th St., N.W., Canton 3, Ohio.
Kirsch, Marvin (J'40) (AHJC), Ensign, U.S.N.R., A.G.O. 611, Naval Aviation Tech. Trng. Center, Jacksonville, Fla.; *for mail*, 1585 E. 19th St., Brooklyn 30, N.Y.
Kirschke, Norman Walter (J'42) (BCENJY), Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 710 Bedford Rd., Schenectady, N.Y.
Kirsten, Frederick Kurt ('29) (AESY), Prof. Aero. Engrg., Univ. of Washington, Seattle 2, Wash.
Kirwan, Kenneth K., Jr. (J'35) (CJM), Constr. Supt., Raymond Concrete Pile Co., 140 Cedar St., New York, N.Y.; *for mail*, 209 Chancery Rd., Baltimore 18, Md.
Kirwin, Thomas Joseph, Jr. (J'42) (ZL), Instrument Engr., E. I. du Pont de Nemours & Co., Belle; *for mail*, 815 Bridge Rd., Charleston 4, W.Va.
Kisa, Oscar A. ('36) (OLSMFE), Cons. Engr., Oscar A. Kisa & Associates, 2932 N. Newhall St., Milwaukee 11, Wis.
Kish, George D. (J'45) (NJLH), Devel. Engr., Dresser Mfg. Div., Dresser Industries; *for mail*, 60 S. Kendall Ave., Bradford, Pa.
Kishbaugh, T. L. (J'44), 545 N. Sunnyslope, Pasadena 8, Calif.
Kishel, Chester Joseph (J'44) (MJCFA), Prod. Control Mgr., A. W. Hecker Co., E. 71st St. & Euclid Ave.; *for mail*, 4317 Clybourne Ave., Cleveland, Ohio.
Kishibay, Charles Ohannes (J'43) (BNRAM), Instr. in Engrg. & Physics, Pratt Inst. of Sci. & Tech., 195 Grand Ave., Brooklyn; *for mail*, 101 Lexington Ave., New York 16, N.Y.
Kisner, Albert G. ('38; '41) (OSMTAC), Sales Engr., Gen. Elec. Co., 1405 Locust St., Philadelphia, Pa.
Kispert, Edwin G. (J'41), Lt. (j.g.), U.S.N.R., 69 Herman St., Fall River, Mass.
Kissam, William Matteson (J'35) (CHDMOA), Lt. (j.g.), U.S.N.R., Bur. of Ord., Navy Dept., Rm. 3016, Navy Bldg., Washington 25; *for mail*, 17 Danbury St., S.W., Washington 20, D.C.
Kistler, Paul Norman ('24; '30) (SFKEZJ), Prof. Mech. Engrg., Brown Univ., Providence, R.I.
Kite, Henry Jahke ('15; '23) (SKFZ), Asst. Supvg. Engr., Strawbridge & Clothier, 801 Market St., Philadelphia 5; *for mail*, 20 Lodges Lane, Cynwyd, Pa.
Kitt, Howard Geraldine ('42) (SLCF), Power Engr., United Engrs. & Constructors, 1401 Arch St., Philadelphia, Pa.
Kittner, Harry (J'44) (CMDLOZ), Lt., U.S.M.C.R., Hdq. & Serv. Co., 1st Engr. Bn., 1st Mar. Div., F.P.O., San Francisco, Calif.; *for mail*, Box 12, Weldon, N.C.
Kittredge, Arthur Edmund ('44) (NKBSLA), V.P., Ch. Engr., Cochrane Corp., 17th & Allegheny, Philadelphia 32, Pa.
Kittredge, John Mahon (J'35) (A), Asst. Project Engr., Wright Aero. Corp., 329—19th Ave. at Vreeland, Paterson, N.J.
Kittredge, John W. ('16; '35) (ZED), Mch. Designer, M. H. Rhodes, Inc., 30 Bartholemew St., Hartford; *for mail*, 130 Kenyon St., Hartford 5, Conn.
Kittrell, James Barnard (J'42) (CMDW), 3rd Asst. Engr., U.S. Maritime Serv., Am. S. African Line, Inc., New York; *for mail*, 91 Village Lane, Rochester 10, N.Y.
Kittrell, Oliver Terry (J'43) (ACM), 2510 Belmont Blvd., Nashville, Tenn.
Klafstad, Erling ('31; '35) (CSEBMZ), Ch. Engr., Crosby Steam Gage & Valve Co., 10 Roland St., Charlestown 29; *for mail*, 18 Brookside Ave., Belmont 79, Mass.
Klapes, Michael Charles (J'43) (ACFO), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8, Conn.
Klappenbach, Hugo Edwin (J'44) (CZSW), Asst. Mgr., Real Estate Dept., 1st Wis. Natl. Bank, 735 N. Water St., Milwaukee 2; *for mail*, 1736 S. 53rd St., Milwaukee 14, Wis.
Klatchko, Manuel (J'41) (ANYJMC), Design Engr., Consltd. Vultee Aircraft Corp., Lindbergh Field, San Diego; *for mail*, 6430 Acorn St. San Diego 5, Calif.
Klauber, Laurence Monroe ('19) (SCF), V.P., Gen. Mgr., San Diego Gas & Elec. Co., Box 1831, San Diego 12, Calif.
Klauberg, Walter Edward (J'44), Design Engr., A. O. Smith Corp., 240 Mellie Esperson Bldg., Houston 2; *for mail*, 1235 W. 25th St., Houston 8, Tex.
Klaus, Albert Joseph (J'45) (BLM), Precision Instrument Maker, Keuffel & Esser Co.; *for mail*, 369 New York Ave., Jersey City, N.J.
Klaus, Jules William (J'44) (BAONDC), Engr., Broderick & Bascom Rope Co., 4203 Union Blvd., St. Louis; *for mail*, 3347 Lawn Ave., St. Louis 9, Mo.
Kleffel, Harrison Edward ('23; '35) (SFEH), Supt. Constr., Union Elec. Co., 315 N. 12th Blvd., St. Louis 1, Mo.
Klein, Alfred (J'45) (BCDJMN), Planning Engr., Lewyt Corp., 60 Broadway, Brooklyn 11; *for mail*, 200 W. 16th St., New York 11, N.Y.
Klein, Allwin S. (J'41) (ABM), Jr. Engr., Aetna Stand. Engrg. Co., 275 W. Federal St., Youngstown; *for mail*, 1553 Sheridan Ave., Warren, Ohio.

Klein, Arthur Warner ('03; '11) (SLEBAK), Prof. Mech. Engrg., Lehigh Univ.; *for mail*, 43 Wall St., Bethlehem, Pa.
Klein, August Clarence ('28; F'44) (SEFDOK), Engrg. Mgr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Klein, Bernard A. ('24; '35) (FKS), Serv. Engr., Bricksteel Refractory Co., 1029 Clinton St., Hoboken; *for mail*, 334 Grove St., Montclair, N.J.
Klein, Bernard D. ('23), Pres., Gas Purifying Matls. Co., Inc., Foot of Halsey St.; *for mail*, 30-34—36th St., Long Island City, N.Y.
Klein, Edward William ('18; '24) (KFH), Southeast. Rep., Warren Webster & Co., 152 Nassau St., N.W., Atlanta 3, Ga.
Klein, Edward William, Jr. (J'38) (KOC), Partner with Edward William Klein (Sr.), Mfrs. Agent & Cons. Engr., 152 Nassau St., N.W., Atlanta 3, Ga.
Klein, Henry G. ('39) (FSKBJ), Dist. Serv. Engr., Babcock & Wilcox Co., 2730 Koppers Bldg., Pittsburgh 19, Pa.
Klein, Julian J. (J'33; M'45) (RMS), 607 S. Sandusky Ave., Bucyrus, Ohio.
Klein, Leonard Martin (J'38) (NLTD), Assoc. Agric. Engr., Bur. of Plant Industry, Soils & Agric. Engrg., U.S. Dept. of Agric., Agric. Engrg. Bldg., Ore. State College; *for mail*, 3750 Oak Creek Rd., Corvallis, Ore.
Klein, Milton (J'42) (JMC), Mech. Engr., David Bogen Co., Inc., 663 Broadway, New York 12; *for mail*, 1499 Coney Island Ave., Brooklyn 30, N.Y.
Klein, Rulof ('23) (OSEF), 35 Parkway, Montclair, N.J.
Klein, Samuel J. (J'43) (AMNHC), Partner, Klein, Lawson & Co., 418 Murphy Bldg., Detroit 26, Mich.
Klein, Walter (J'43) (MNBECL), 1507 Popham Ave., Bronx 53, N.Y.
Kleinman, Harold Atchison ('28) (SFE), Elec. Prod. Engr., Iowa-Ill. Gas & Elec. Co., 320—18th St., Rock Island; *for mail*, 2305—14th Ave., Moline, Ill.
Kleinmann, Earl Edward (J'42) (LHCZYK), Sr. Engr., Chem. Engrg. Dept., Phillips Petroleum Co., 206 Osage; *for mail*, Box 1407, Bartlesville, Okla.
Kleinschmidt, Arthur Charles (J'46), Box 105, College, Alaska.
Kleinschmidt, Kenneth Theodore (J'43) (BDY), Engr., Raybestos-Manhattan, Inc., Willett St., Passaic; *for mail*, 21 De Mott Ave., Clifton, N.J.
Kleinschmidt, Robt. V. ('21; '29), 20 East St., Stoneham, Mass.
Kleist, Dale ('42) (CMTYJB), Mgr., Engrg. Labs., Owens-Corning Fiberglas Corp., Case Ave., Newark, Ohio.
Klekotka, John Aloysius (J'44) (AGM), Rev., Order of St. Augustine; Instr., Villanova College, Villanova, Pa.
Klemin, Alexander ('19; '31) (ABEHWN), Aero. Consultant, Anderson Rd., Greenwich, Conn.
Klemmer, Kenneth Rinn (J'44), Ensign, LCS (L) (3) 20, F.P.O., San Francisco, Calif.
Klep, Martin C. ('30; '35) (ADG), Engr. Asst., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail*, 47 S. Mill St., Nyack, N.Y.
Klepetko, Ernest, Jr. (J'42) (FKSECA), Equip. Inspr., Lago Oil & Trans. Co. Ltd., Aruba, N.W.I.
Kleppe, Carl Albert (J'45) (SLMEH), Student Engr., Graduate Trng. Course, Allis-Chalmers Mfg. Co., Milwaukee; *for mail*, 2311 N. Buffum St., Milwaukee 12, Wis.
Klespies, Frank Joseph (J'43) (MSL), Engr., Design & Constr., B. F. Goodrich Co., Main St., Akron; *for mail*, 467 Madison Ave., Akron 2, Ohio.
Kliebhan, Frank Henry (J'44) (ABE), 1007 E. Potter Ave., Milwaukee 7, Wis.
Klimas, Claude Paul ('45), Dist. Engr., Combustion Engrg. Co., Inc., 80 Federal St., Boston; *for mail*, 43 Choate Rd., Belmont, Mass.
Kliment, William P. ('43) (SHKN), Engr. of Stands., Crane Co., 836 S. Michigan Ave., Chicago 5; *for mail*, 1643 S. 61st Court, Cicero 50, Ill.
Klinck, John Henry ('04) (RSATY), Retired; Box 9206, Tampa 4, Fla.
Kline, Clement LeRoy ('39) (SKFZHD), Mech. Engr., with G. W. Williams, Cons. Engr., 627 Architects Bldg.; *for mail*, 719 E. 34th St., Indianapolis, Ind.
Kline, G. M. ('41), Chief, Organic Plastic Sec., Natl. Bur. of Stands., Washington, D.C.; *for mail*, 109 Battery Lane, Bethesda, Md.
Kline, John Henry (J'42), Grad. Trng., Prod. Control, Allis-Chalmers Mfg. Co.; *for mail*, 1970 S. Layton Blvd., Milwaukee, Wis.
Kline, Lee Andrew ('26; '41) (FSEKZ), Power & Fuel Engr., Carnegie-Ill. Steel Corp., 912 Salt Springs Rd., Youngstown 9; *for mail*, Box 657, R.F.D. 1, Niles, Ohio.
Kline, Mason E. ('31) (AW), V.P., Union Lumber Co., 1010 Crocker Bldg., San Francisco, Calif.
Kline, Paul A. (J'34) (DJM), 1423 Sheffield Rd., South Euclid 21, Ohio.
Klinedinst, L. M. ('29), Gen. Mgr., Indus. Div., Timken Roller Bearing Co., Canton, Ohio.
Klint, Robert Victor (J'45) (EABOCS), Test Engr., Gen. Elec. Co., Schenectady; *for mail*, 2435 Guilderland Ave., Schenectady 6, N.Y.
Klintman, Pehr Christian ('29; '35) (EFHRS), Comdr., U.S.N.R., 47 Ardmore Ave., Lansdowne, Pa.

Klonoski, Arthur Francis (J'43) (BJM), 157 Central Ave., Torrington, Conn.
Klopsch, Otto Zwicker ('38) (KMCD), Asst. to Pres., Phelps Dodge Copper Products Corp., 40 Wall St., New York, N.Y.
Klotz, Harry John ('13; '20; '35) (SFEKOZ), Ch. Power Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Klotz, Walter (J'40) (BJ), Devel. Engr., Walter Kidde & Co., Inc., 675 Main St., Belleville 9; *for mail,* 1 Fuller Pl., Arlington, N.J.
Kluesener, Hugo H. ('34; '35), Mch. Design, Am. Can Co.; *for mail,* 206 Leslie St., Newark, N.J.
Klumpp, John B. ('09) (OFCM), Cons. Engr., 123 S. Broad St., Philadelphia 9, Pa.
Klutey, Frederick E. ('25; '43) (BKCLHM), Acting Head, Chem. Engrg. Sec., Tech. Div., Engrg. Dept., E. I. du Pont de Nemours & Co., Wilmington 98; *for mail,* 3203 Swarthmore Rd., Wilmington 79, Del.
Knabe, Carl F. ('41) (ABK), Sales Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 1940 Albany St., Schenectady, N.Y.
Knabe, Fred Smith (J'35) (EMR), Diesel Maintainer, Bethlehem Steel Co., Sparrows Point; *for mail,* 2747 Dunglen Court, Liberty Gardens, Dundalk, Md.
Knape, Harold Oscar ('43) (ELM), Gen. Mgr., Wix Accessories Corp.; *for mail,* Box 471, Gastonia, N.C.
Knapp, Charles Andrew (J'42) (AJBNMF), Project Engr., Westinghouse Elec. Corp., Lester; *for mail,* 206 Avon Rd., Springfield, Del. Co., Pa.
Knapp, Edwin C. ('93; '32) (BFK), Retired; 1124 Parkwood Blvd., Schenectady 8, N.Y.
Knapp, Hans John (J'43), 410—9th Ave., N.E., Puyallup, Wash.
Knapp, Peter Richard ('45) (SCKEFO), Gen. Supt., Toledo Edison Co., Edison Bldg., Toledo 4, Ohio.
Knapp, Robert Talbot (J'21) (HBAKNE), Assoc. Prof. Hyd. Engrg., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.
Knapp, S. Arthur, Jr. (J'37) (EJP), Oil Well Supply Co.; *for mail,* 310 Central Ave., Oil City, Pa.
Knapp, Walter ('15; '23) (BCDGJN), Ch. Mech. Engr., Equip. Devel., RCA Victor Div., Radio Corp. of Am.; *for mail,* 718 N. Shippen St., Lancaster, Pa.
Knauer, Ebert G. ('19) (BCDJMN), Research Engr., Burroughs Adding Mch. Co., Detroit 32; *for mail,* 4269 Fullerton Ave., Detroit 4, Mich.
Knaus, William Lloyd ('39; '41) (BKJN), Engr., Cent. Plant Engrg. Sec., Air Conditioning Dept., Gen. Elec. Co., 5 Lawrence St., Bloomfield; *for mail,* 138 Alexander Ave., Upper Montclair, N.J.
Knauss, G. Edwin (J'41) (ANHRKB), Engrg. Checker, Glenn L. Martin Co., Baltimore; *for mail,* 3112 Harview Ave., Baltimore 14, Md.
Knebel, Harvey L. (J'41) (AEFBHJ), Flight Engr., Pan Am. Airways, Mills Field, South San Francisco; *for mail,* 2763 Garber St., Berkeley 5, Calif.
Knecht, A. Wilson ('44) (OCLMSF), Project Mgr., with Clyde R. Place, Cons. Engr., Rm. 2122, 420 Lexington Ave., New York 17; *for mail,* 20 Grosbeak Rd., Yonkers 2, N.Y.
Knecht, Harry ('38) (S), Div. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.; *for mail,* 8 Colonial Terrace, Maplewood, N.J.
Knezo, John, Jr. (J'35) (ABENJ), Project Engr., Breeze Corps., Inc., 41 S. 6th St., Newark 7; *for mail,* 309 East St., Garwood, N.J.
Knickerbacker, John ('91), 30 Pleasant St., Ballston Spa, N.Y.
Knierim, Vincent LeRoy ('45) (SKLC), Ch. Draftsman, Tenn. Eastman Corp.; *for mail,* 1505 Waverly Rd., Kingsport, Tenn.
Kniese, Harold George ('17; '21; '30) (BCKOS), Dir., Indus. Relations, Cent. Ill. Light Co., 316 S. Jefferson Ave., Peoria 2, Ill.
Kniffen, Allen T. ('36; '44) (KEFL), Engr., Air Reduction Sales Co., 60 E. 42nd St., New York 17, N.Y.
Knight, Charles Adrian ('42) (OCKJDM), Ch. Engr., Magnolia Petroleum Co., Box 3311, Beaumont, Tex.
Knight, Earl Rhodes ('17; '26; '35) (BFKSZ), Professional Engr., Manchester & Hudson Co., 573 Eddy St., Providence 3; *for mail,* Corner Smith & Orchard Ave., Greenville, R.I.
Knight, Fred Daniel ('43) (SFKC), Asst. to Sr. V.P., Hartford Elec. Light Co., 266 Pearl St., Hartford 3, Conn.
Knight, George Laurence ('05; '08; F'38) (CO S), Retired; 1 Harbor Lane, Brooklyn 9, N.Y.
Knight, John Joseph (J'43) (LKS), Apprentice Engr., Gen. Chem. Co., El Segundo; *for mail,* 710 S. Redondo Blvd., Inglewood, Calif.
Knight, John M. ('44) (RYKJNB), Mech. Engr., Am. Car & Foundry Co., 30 Church St., New York 8, N.Y.
Knight, Louis Perolio (J'44) (CJM), 1804 Enolam Blvd., Decatur, Ala.
Knight, Seymour H. ('19) (DLWS), Supvg. Engr., Strawbridge & Clothier, 8th & Market Sts., Philadelphia 5; *for mail,* 615 Manoa Rd., Havertown P.O., Pa.
Knight, Sidney ('41) (ACL), Assoc., McKinsey & Co., 60 E. 42nd St., New York; *for mail,* 1 Highwood Way, Larchmont, N.Y.
Knipping, Rainer Hubert ('19; '26) (EFHKS), R. H. Knipping Co., 623 Aztec Bldg., San Antonio 5, Tex.
Kniskern, Walter Hamlin ('05; '12; F'45) (LBK SDE), Consultant, Nitrogen Div., Solvay Process Co., Hopewell, Va.
Knoblock, Alvin F. ('14) (AEM), Chmn. of Bd., Bundy Tubing Co., Hern Ave.; *for mail,* 2850 Cambridge, Detroit, Mich.
Knoblock, John Edward (J'45) (ACK), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland, Ohio; *for mail,* 19145 Ardmore Ave., Detroit 21, Mich.
Knoblock, Kenneth David('43) (OCKF), V.P., Ch. Engr., E. B. Mathewson Co., 612 N. Michigan Ave., Chicago 11, Ill.
Knocke, Louis T. ('20; '22) (BEJ), Asst. Ch. Engr., Engrg. Dept., Dodge Bros. Corp., 7900 Jos. Campau Ave.; *for mail,* 20050 Renfrew Rd., Detroit, Mich.
Knoedler, Elmer L., Jr. ('41; '45) (OSLKH), Engr., Sheppard T. Powell, 330 N. Charles St., Baltimore 1; *for mail,* 3120 St. Paul St., Baltimore 18, Md.
Knoll, Herman (J'36), 125—2nd St., Garden City, N.Y.
Knoll, Herman J. (J'40) (LDJSBC), Plant Engr., Del. Works, Gen. Chem. Co., Marcus Hook; *for mail,* 308 Pomeroy St., Ridley Park, Pa.
Knoll, Rudolph Julius (J'35) (JEDKOS), Plant Engr., Laclede Steel Co., Madison, Ill.; *for mail,* 310 Bellerive Blvd., St. Louis 11, Mo.
Knoop, Theo. M. ('04; '16) (HKS), V.P., Otto Knoop Lumber & Realty Co., Ltd., 1201 Julia St., New Orleans, La.; *for mail,* 25 Clarkson Ave., Brooklyn, N.Y.
Knopf, Kenneth Kermit ('46) (EEL), Asst. to Mgr., Engrg. & Sales, Joshua Hendy Iron Works, Sunnyvale; *for mail,* 201 W. Poplar Ave., San Mateo, Calif.
Knopp, Walter Vincent (J'44) (JMCLNZ), Research Fellow, Stevens Metal Powder Lab., 1 Elysian Pk., Hoboken, N.J.; *for mail,* 38 Seely Pl., Scarsdale, N.Y.
Knorring, Edward (J'46), 212 S. 18th Ave., Maywood, Ill.
Knott, Maurice John (J'39) (BMJ), Mch. Design Engr., Brown & Sharpe Mfg. Co., Providence; *for mail,* 336 Prospect St., Pawtucket, R.I.
Knowles, Asa Smallidge ('44) (CDM), Dean, Sch. of Business Admin. & Dir., Gen. College Extension, R.I. State College, Kingston, R.I.
Knowles, Carroll ('40) (NMLHBJ), Ch. Engr., Mchy. Div., Pratt & Whitney Div., Niles-Bement-Pond Co., West Hartford; *for mail,* 444 Farmington Ave., Hartford, Conn.
Knowles, Dennis Henry (J'44) (COY), Lt. (E), Royal Navy, 2 Woodlands Rd., Bexley Heath, Kent, England.
Knowles, Douglas Wigham (J'42) (ENBK), Head, Lab. Design Sec., Turbo Research, Ltd., Leaside; *for mail,* 194 Grenadier Rd., Toronto 3, Ont., Can.
Knowles, Gregory W. (J'46) (HOW), Draftsman, Mchy., Wheeler Shipbldg. Corp., Foot of 154th St., Whitestone; *for mail,* 70 S. Bayles Ave., Port Washington, N.Y.
Knowles, Miss Jeanette B. (J'40) (CM), V.P., Knowles Tool Corp., 800 S. 7th St.; *for mail,* 305 S. 22nd St., Richmond, Ind.
Knowlton, Archer Eben ('43) (SKEHFO), Assoc. Editor, *Electrical World,* McGraw-Hill Publ. Co., 330 W. 42nd St., New York 18, N.Y.; *for mail,* Box 213, Short Beach, Conn.
Knowlton, Lauriston Edward ('39) (FCE), Engr., Providence Gas Co., 100 Weybosset St., Providence 1; *for mail,* 15 Hall Pl., Edgewood 5, R.I.
Knowlton, Paul Holland, Jr. ('30; '37) (SEAR CB), Asst. Design Engr., Turbine & Generator Div., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Knox, S. L. G. ('92; '01; F'41), Cons. Engr., McGlynn, Hays & Co., 328 Belleville Turnpike, Arlington, N.J.; *for mail,* 17 E. 95th St., New York, N.Y.
Knudsen, William S. (H'46), Dir., Gen. Motors Corp., Detroit; *for mail,* 1501 Balmoral Dr., Detroit 3, Mich.
Knust, Herman Russell (J'41) (ACJ), Field Foreman, Mech. Dept., Bethlehem Steel Co., Sparrows Point; *for mail,* Apt. C, 7008 Morningtou Rd., Dundalk, Md.
Koch, Albert Herman ('38) (TZCLK), Southeast. Regional Mgr., Minneapolis-Honeywell Regulator Co., 5 Rhodes Center, Atlanta, Ga.
Koch, Bruno F. ('21; '26) (SFEDK), Power Plant Designing Engr., Anaconda Copper Min. Co., 25 Broadway, New York 4, N.Y.
Koch, Charles Herbert, Jr. (J'43) (DJMNY), Asst. Ch. Engr., United Specialties Co., Shelmire & Edmund St., Philadelphia 36; *for mail,* 3370 Vaux St., Philadelphia 29, Pa.
Koch, Edward Gaulbert ('26; '35) (CEKMRW), Pres., Gen. Mgr., Miranda Sugar Estates, Miranda, Oriente, Cuba.
Koch, Geo. B. ('09), Retired; Box 566, Altoona, Pa.
Koch, Herman G. (J'33) (BLS), Sec. Head, Johns-Manville Research Lab., Manville; *for mail,* Country Club Rd., Somerville, N.J.
Koch, Theodore Frederick (J'39) (COJKHA), 2 Stoll's Alley, Charleston 1, S.C.
Koch, William Martin (J'45), Pvt., U.S.A., Box 1722, Houston, Tex.
Kocher, E. J. (J'42) (HJK), 3438 N. 3rd St., Milwaukee, Wis.
Kocher, Edward Henry ('30) (C), Consultant, Bijur Lub. Corp., 43-01—22nd St., Long Island City, N.Y.; *for mail,* Reserve St., Boonton, N.J.
Kochlacs, Alexander John (J'41) (R), Design Engr., C. F. Braun & Co., Alhambra; *for mail,* 681 S. Norton St., Los Angeles 5, Calif.
Kocmit, Otto, Jr. (J'40) (CBDM), Aviation Radio Technician, U.S.N., Bks. 1, Sec. 46-45, Naval Aviation Tech. Trng. Center, Ward Island, Corpus Christi, Tex.; *for mail,* 15827 Friend Ave., Maple Heights, Bedford, Ohio.
Kocsis, Julius (J'31), Draftsman, Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 125 Brown Ave., Turtle Creek, Pa.
Kocyan, George Henry, Jr. (J'45) (EOR), Designer, Bldg. B-14, Baldwin Loco. Works, Paschall P.O., Philadelphia 42, Pa.
Koechlein, George John (J'38) (SE), Lt. (j.g.), U.S.N.R.; *for mail,* 217 Edgewood Ave., Westfield, N.J.
Koehler, Christopher L. ('19; '30) (CDHLY), Works Mgr., Schaible Co., 1086 Summer St., Cincinnati 4; *for mail,* 2940 Wold Ave., Cincinnati, Ohio.
Koehler, Oscar Ernest ('39) (BJLNRZ), Ch. Engr., Cutting Tool Div., Greenfield Tap & Die Corp., Sanderson St.; *for mail,* 32 Maple St., Greenfield, Mass.
Koellish, Wilfred Marvin (J'41) (ACM), Sr. Flight Test Engr., Glenn L. Martin Co., Baltimore 3; *for mail,* 1203 Shore Rd., Middle River 20, Md.
Koenig, C. Fred, III (J'42) (ABKSE), 226 Arlington Ave., Milmont Park, Pa.
Koenig, Edward Carl (J'36) (MZ), Serv. Mgr., Stanley Elec. Tool Div., Stanley Works, 61 W. Kinzie St.; *for mail,* 3635 N. Seeley St., Chicago, Ill.
Koenig, Eugene Herman ('32; '39) (CDMNYZ), Plant Engr., Keuffel & Esser Co., Adams & 3rd St., Hoboken; *for mail,* 518 S. Forest Dr., Teaneck, N.J.
Koenig, Fred Victor (J'41) (SBAKZ), Asst. Mchy. Engr., Steel Shipbldg. Div., United Concrete Pipe Corp., 1700 S. 9th St., Long Beach; *for mail,* 455 E. Ocean Blvd., Long Beach 2, Calif.
Koenig, Lloyd Roesch ('23; '34) (BNGO), Mem. Firm, Haynes & Koenig, 818 Olive St., St. Louis 1; Asst. Prof. Mech. Engrg., Wash. Univ., St. Louis, Mo.
Koenig, Louis, Jr. (J'43) (KHBNEZ), Test & Research Engr., Cargocaire Engrg. Corp., 223 Clifton Blvd., Clifton, N.J.; *for mail,* 66-14 Woodside Ave., Woodside, N.Y.
Koenig, William Carl (J'41) (ABCKNY), Mech. Engr., War Dept., Wright Field; *for mail,* 32 Vandergrift Dr., Dayton, Ohio.
Koenig, William Walter (J'42) (NLMJBZ), Design Engr., Internatl. Business Mchs. Lab., 310—5th Ave., New York; *for mail,* Hotel St. George, Brooklyn 2, N.Y.
Koepke, Charles A. ('29; '35) (CMD), Owner & Head, Firm of C. A. Koepke, Cons. Engr., Box 623, St. Paul, Minn.
Koerner, Arthur Edward (J'42), Lt. (j.g.), U.S.N.R., 1102 Long Pond Rd., Rochester 13, N.Y.
Koerper, Erhardt Carl (J'41) (CBLZ), Charge Spec. Engrg., A. O. Smith Corp.; *for mail,* 5204 W. Washington Blvd., Milwaukee 1, Wis.
Koerwer, Richard John (J'42) (SAOEC), 69 E. Essex Ave., Lansdowne, Pa.
Koester, Carl John ('38) (CLDNYZ), Works Engr., Natl. Carbon Co., Inc., Fostoria, Ohio.
Koester, Herman ('11; '13) (CMZJ), V.P., Works Mgr., Bristol Co., Waterbury; *for mail,* 41 Columbia Blvd., Waterbury 33, Conn.
Koester, Wilbur Fred ('39; '44) (MDJS), Asst. Plant Engr., Allis-Chalmers Mfg. Co., West Allis 14; *for mail,* 7407 W. North Ave., Wauwatosa 13, Wis.
Koethe, Werner Emil (J'42) (ACBMJ), Aircraft Engrg. Officer, Army Air Forces; *for mail,* 210 Owen Ave., Lansdowne, Pa.
Koetter, Frederick William (J'45) (AFHEK), Carburetor Test Engr., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 315 Pearl St., Hartford 4, Conn.
Koetting, John Lawrence (J'45) (EKS), Thermodynamics Engr., Westinghouse Elec. Corp., Lester; *for mail,* 919—12th Ave., Prospect Park, Pa.
Koffmann, Eugen (J'41) (BRS), Mech. Design Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 422 Morewood Ave., Pittsburgh 13, Pa.
Koffsky, Samuel ('42) (BCM), Ch. Engr., Simmons Mch. Tool Corp., N. Broadway; *for mail,* 845 Park Ave., Albany, N.Y.
Kogan, Zuce ('43) (FOS), Cons. Engr., 724 Sheridan Rd., Chicago 13, Ill.
Kohanow, Nicholas (J'37) (ABG), Ch. Draftsman, Warren McArthur Corp., Bantam; *for mail,* Box 136, Lincoln Pl., Litchfield, Conn.
Kohl, Earl Cecil (J'41) (HAC), Pump Engr., Kohl Bros., 21st & Greenwood St., Harrisburg; *for mail,* 1 Jefferson Ave., Myerstown, Pa.
Kohl, Frederick Sanford (J'34) (SCRE), Sales Engr., Turbine Div., Gen. Elec. Co., 166 Broad St., Fitchburg, Mass.
Kohler, Albert Joseph ('31) (SKFEHM), Apt. D-514, 3551—39th St., Washington 16, D.C.
Kohler, Anthony M. ('17; '23) (CMLDFZ), Gen. Mgr., Refractories Div., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.

Kohler, John H. ('44) (AJM), c/o United-Carr Fastener Corp., 31 Ames St., Cambridge, Mass.
Kohlhepp, Dean H. (J'37) (BJM), Ensign, U.S.N.R.; 213 Ash St., Ridgway, Pa.
Kohlmann, Gunter ('33; '44) (ESF), Sales Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7, Wis.
Kohn, David F. (J'45) Chem. Engr., Food Research Labs., 48-14—33rd St., Long Island City; *for mail,* 68-64 Yellowstone Blvd., Forest Hills, N.Y.
Kohn, Emanuel Joseph ('35) (FHBS), Ch., Bur. of Steam Engrg., Tenn. Coal, Iron & R.R. Co., Ensley Sta., Birmingham 8, Ala.
Kohn, Julius Albert (J'43) (ASCBM), Design Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 814 Woodland Ave., Schenectady 8, N.Y.
Kohnen, Bernard William (J'41) (ZBH), Devel. Engr., Dole Valve Co., 1923 W. Carroll Ave., Chicago 12; *for mail,* 3946 W. Dakin St., Chicago 18, Ill.
Kohner, John A. (J'39), Power Indus. Sec., Gen. Motors Overseas Operas. Div., Gen. Motors Corp., 1575 Broadway, New York, N.Y.
Kohut, Frank J. ('34), Sales Mgr., C. M. Kemp Mfg. Co., 405 E. Oliver St., Baltimore, Md.
Koke, Leonard Carl (J'42) (BJ), 535 Meade St., Denver 4, Colo.
Kolb, Robert Edwin (J'40) (ABK), 16516 Wisconsin Ave., Detroit 21, Mich.
Kolb, Robert Phillips ('26; '30) (SKEF), Prof. Heat-Power Engrg., Worcester Poly. Inst., Boynton St., Worcester 2, Mass.
Kolbe, Gervase Henry (J'38) (HLNWBO), Mech. Devel. Engr., Kimberly-Clark Corp.; *for mail,* 313 Forest Ave., Neenah, Wis.
Kolesh, Victor Anthony (J'41) (A), 4 View St., Worcester 4, Mass.
Kolflat, Alf ('45), Partner, Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Kollberg, Gustaf Leonard ('10) (OAEHC), Cons. Engr., 924 E. Juneau Ave., Milwaukee 2, Wis.
Kolzing, Henry H. ('44) (CEH), Oliver Corp.; *for mail,* Neal Hotel, Shelbyville, Ill.
Kominsky, Jack David (J'42) (ABS), Sr. Draftsman, Jack D. Kominsky, 60 E. 42nd St., New York 17; *for mail,* 244—8th Ave., New York, N.Y.
Kongelbeck, Sverre ('41) (BHJKMZ), Ch. Engr., Infilco Inc., 325 W. 25th Pl., Chicago 16, Ill.
Konheim, Harvey S. ('32; '35) (CMJTLB), Planning Engr., West. Elec. Co., Inc., Jersey City, N.J.; *for mail,* 500 West End Ave., New York 24, N.Y.
Konigsberg, Irwin (J'43), 917 Sheridan Ave., Bronx 56, N.Y.
Konnerth, Herman (J'40) (NDCHK), Mech. Engr., Bechtel Bros. McCone Co., Box 162, Vernon Branch; *for mail,* 11439 S. Vermont Ave., Los Angeles 44, Calif.
Konstan, Paul (J'41) (HKSED), Jr. Mech. Engr., Shell Devel. Co., 100 Bush St., San Francisco 6, Calif.
Koontz, D. Lionel ('45) (BMO), U.S. Custom House, Denver; *for mail,* 810—14th St., Denver 2, Colo.
Koontz, Glen Millard (J'43) (AEBJ), Serv. Rep., Airplane Div., Plant 2, Curtiss-Wright Corp., Genesee St., Buffalo 5, N.Y.; *for mail,* Haven, Kan.
Koontz, Lamont Burton (J'33) (ZHB), Pat. Agt., Minneapolis-Honeywell Regulator Co., 2753—4th Ave. S., Minneapolis; *for mail,* 4216—13th Ave. S., Minneapolis 7, Minn.
Koopman, Paul (J'39) (SKM), Safety Engr., Boiler Inspr., Hartford Steam Boiler Inspc. & Ins. Co., Hartford, Conn.; *for mail,* 1321 Sherman St., Grand Rapids 6, Mich.
Koos, Karl Kenyon (J'42), 8248 Ingleside Ave., Chicago 19, Ill.
Kop, Peter Everard (J'37) (BNCMS), Ch. Draftsman, Earle Gear & Mch. Co., 4707 Stenton Ave., Philadelphia 44, Pa.; *for mail,* 8 Harcourt Pl., Trenton 10, N.J.
Kopec, Casimir S. (J'39) (NCMBEJ), 19758 Pennington Dr., Detroit 21, Mich.
Kopeck, William W. ('25; '35) (CFS), Maint. Mgr., Gimbel Bros., Inc., 1275 Broadway, New York; *for mail,* 2327 Lafayette Ave., Bronx, N.Y.
Koper, Fred George ('45) (CDF), V.P., Coal Preparation Engr., Pierce Mgmt., Inc., Scranton Elec. Bldg., Scranton, Pa.; *for mail,* 3542—80th St., Jackson Heights, N.Y.
Kopetz, George E. (J'39) (KLNYZ), Ch. Engr., Chem. Plants Div., Blaw-Knox Constr. Co., Box 5087, E. Liberty Sta.; *for mail,* 1628 Chislett St., Pittsburgh 6, Pa.
Kopetz, Walter H. (J'41) (NLCB), Design Engr., Swift & Co., Union Stock Yards, Chicago; *for mail,* 1136 E. Leafland Ave., Decatur 19, Ill.
Kopf, Joseph L. ('31; '35) (CLMNZ), *Assistant Treasurer,* A.S.M.E., '44 to date; Pres., Jabez Burns & Sons, Inc., 600 W. 43rd St., New York 18, N.Y.
Kopf, William Frederick (J'24) (BS), 626 Park Ave., Elizabeth 3, N.J.
Koplin, Robert D. ('11; '21; '35) (SBKFDH), Engr., Power Div., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia; *for mail,* 309 Roslyn Ave., Glenside, Pa.
Kopp, Paul Joseph (A'44), Lt. Col., U.S.A., Office of Chief of Chem. Warfare Serv., Washington 25, D.C.; *for mail,* 1305 N. Adams St., Arlington, Va.
Kopp, Sigmund ('36; '38) (KCSLH), Asst. Ch. Engr., Alco Products Div., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Koppy, Michael (J'44) (SKHELN), Mech. Engr., Puget Sound Naval Shipyard; *for mail,* Gen. Delivery, Bremerton, Wash.
Koran, Stephen Edward (J'42) (SEFKJH), Prof. Mech. Engrg., Youngstown College, Youngstown; *for mail,* 29 Rebecca Ave., Hubbard, Ohio.
Korb, Fred B. ('30; '35) (KFJBAE), Ch. Engr., Indus. Gas Engrg. Co., 231 W. Huron St., Chicago 10, Ill.
Kories, Otto V. ('23) (ESY), Comdr., Ch. Engr., U.S. War Shipping Admin., Washington, D.C.; *for mail,* 1471 Bellevue Ave., Los Angeles 26, Calif.
Kormendy, Louis ('42) (EFOS), Mech. Engr., H. E. Beyster Corp., Indus. Bank Bldg., Detroit 26; *for mail,* 8010 Pilgrim St., Detroit 21, Mich.
Kornylak, Andrew Thomas (J'45), Treas., Ch. Engr., James Allen Tuck Corp., 1 Exchange Pl., Jersey City 2; *for mail,* 152 Jewett Ave., Jersey City 4, N.J.
Korr, Irving (J'42) (OKHFD), Jr. Mech. Engr., Jaros, Baum & Bolles, 415 Lexington Ave., New York; *for mail,* 1415 E. New York Ave., Brooklyn 12, N.Y.
Korstian, Robert John (J'42) (AMJ), Master Sgt., Regimental Staff Sec., 60th Inf., A.P.O. 9, c/o Postmaster, New York, N.Y.; *for mail,* 4 Sylvan Rd., Durham, N.C.
Kort, Ernest George ('44) (JCAKLT), Div. Ch., Jobbing Engrg. Div., Aluminum Co. of Am.; *for mail,* 306 Keystone Dr., New Kensington, Pa.
Korten, Elmer C. ('41) (SJBYL), Asst. Ch. Engr., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford 2; *for mail,* 239 Garden St., Wethersfield 9, Conn.
Korycinski, Peter Francis (J'42) (AKM), Aero. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail,* 864 E St., Copeland Park, Newport News, Va.
Kosciuch, Edmund K. (J'40) (BKL), 4017 Potomac Ave., Chicago 51, Ill.
Kosecoff, Irving William (J'45) (CMY), 293 Hooker Ave., Poughkeepsie, N.Y.
Kosh, Joseph Albertis (J'44) (BHM), Vacuum Engr., Keller Corp., 233 Broadway, New York 7; *for mail,* c/o Herzberger, Apt. 6F, 390 Wadsworth Ave., New York 33, N.Y.
Koskan, Allen J. (J'42), Box 566, Barberton, Ohio.
Koski, Elmer Jacob ('22; '26; '35) (MN), Field Engr., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3, Ohio; *for mail,* 5412 N. Hollywood Ave., Milwaukee 11, Wis.
Koskinen, Einar T. ('18; '35) (EHMNSY), Ch. Engr., Prescott Co.; *for mail,* 2001 Broadway, Menominee, Mich.
Kosley, Raymond Walter (J'42) (ACM), 17 Elm St., Morristown, N.J.
Kosman, Milton (J'37) (CDHJMN), Owner, Oakland Engrg. Co., 481—25th St., Oakland 12, Calif.
Koss, Aloysius J. (J'41), 7453 Woodrow Wilson, Detroit, Mich.
Kostir, Edward (J'44), Jr. Engr., Vlchek Tool Co., 3001 E. 87th St., Cleveland; *for mail,* 3820 Arnold Ct., Cleveland 9, Ohio.
Kostka, Fred Paul (J'46), 27 Ashmont St., Providence 5, R.I.
Kosztyla, Camille Joseph (J'41) (JHCMT), Asst. Planning & Estimating Supt., N.Y. Naval Shipyard, Brooklyn, N.Y.
Kotar, Frank John (J'43) (CLM), Mech. Designer, Cent. Research Labs., Gen. Foods Corp., 1125 Hudson St., Hoboken; *for mail,* 5 Hasbrouck Pl., Rutherford, N.J.
Kothe, Harold B. (J'41) (CDT), 4543 Clayton Ave., St. Louis, Mo.
Kothe, Otto W. ('28), Dir. of Education, St. Louis Tech. Inst., 4543 Clayton St., St. Louis, Mo.
Kothe, Robert B. (J'40), 4543 Clayton Ave., St. Louis, Mo.
Kothera, Edward J. (J'40) (KAR), Asst. Research Engr., Young Radiator Co., 709 S. Marquette; *for mail,* Apt. 305, 1024 Main St., Racine, Wis.
Kothny, G. L. ('05; '12; F'38) (HKS), V.P., Sperry-Sun Well Surveying Co., 1608 Walnut St., Philadelphia 3, Pa.
Kotilinek, John (J'31) (BDL), Engr., Prater Pulverizer Co., 1829 S. 55th Ave., Cicero; *for mail,* 1404 S. Highland Ave., Berwyn, Ill.
Kottcamp, John Paul (F'43) (CDMZ), V.P., Johns-Manville Corp., 22 E. 40th St., New York 16; *for mail,* Hudson House, Ardsley-on-Hudson, N.Y.
Kottcamp, John Paul, Jr. ('45) (CGO), 1334 N. Chestnut St., Waukegan, Ill.
Kottman, Joseph J. ('45) (NCLYM), Asst. Ch. Engr., Ansco Div., Gen. Aniline & Film Corp., Charles St., Binghamton; *for mail,* 101 Murray St., Binghamton 43, N.Y.
Kotzebue, Meinhard Henry ('23; '25; F'43) (LMKEHP), 208 Niels Esperson Bldg., Houston 2, Tex.
Kotzen, Herbert (J'44), 15 Pasadena Rd., Grove Hall 21, Mass.
Kousmin, S. I. ('35), Apt. 37, Prospect ½, Leningrad 46, Kirovskii, U.S.S.R.
Kousnetzoff, Valerien P. ('40) (AEKN), Sr. Aero. Engr., Airplane Div., Curtiss-Wright Corp., 4300—5th Ave., Columbus; *for mail,* 235 W. Norwich Ave., Columbus 1, Ohio.
Kousz, John Joseph (J'42) (ABEJY), Stress Analyst, P-V Engrg. Forum, Inc., Sharon Hill; *for mail,* 622 W. Kingsley St., Philadelphia 44, Pa.
Kouwenhoven, Frank W. ('17; '25; '35), Assoc. in Mech. Engrg., Johns Hopkins Univ., Homewood, Baltimore 18, Md.
Kovach, Andrew J. ('23; '24; '35) (BJM), Draftsman, United Engrg. & Fdy. Co., 1st Natl. Bank Bldg., Pittsburgh; *for mail,* 505 Case St., Rochester, Pa.
Kovacs, Stephen Florian ('45) (JBMKS), Design Engr., H. K. Ferguson Co., Hanna Bldg., Cleveland; *for mail,* 3268 E. 135th St., Cleveland 20, Ohio.
Kovaleff, Serge Peter ('45) (CEMLNJ), Gen. Sales Mgr., Enterprise Eng. & Fdy. Co., 18th & Florida Sts., San Francisco 10; *for mail,* 2848 Canyon Rd., Burlingame, Calif.
Koven, Gustav H. ('41), Pres., Gen. Mgr., L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City; *for mail,* Blue Mill Farm, Green Village, N.J.
Kozacka, Joseph S. ('32) (MBJN), Tool Engr., Vascoloy-Ramet Corp., North Chicago; *for mail,* 5345 Wolfram St., Chicago 41, Ill.
Kozica, William Stephen (J'42) (ABCFNS), Lt., U.S.N.R., Engrg. Trng. Officer, Naval Air Sta., Miami, Fla.; *for mail,* 5240 S. Troy St., Chicago 32, Ill.
Krafft, Joseph Martin (J'44) (BENA), Asst. Engr., Engrg. & Research Corp., Riverdale, Md.; *for mail,* 105 Cedar St., Alexandria, Va.
Kraft, Lester L. ('31) (FKSH), Mech. Supt., Union Elec. Co. of Ill., St. Louis 16; *for mail,* 4356 Holly Hills Blvd., St. Louis, Mo.
Kraft, Russell H. (J'43) (AFBSMY), Ensign, A-V (S), U.S.N.R.; *for mail,* 1441 E. 27th St., Brooklyn 10, N.Y.
Krafve, Richard E. ('45) (CMOD), Lt. Col., Gen. Staff Corps, Economics Div., Office of U.S. Military Govt. for Germany, A.P.O. 742, c/o Postmaster, New York, N.Y.; *for mail,* 1501 N. 21st St., Superior, Wis.
Krahe, Richard Lawrence, Jr. (J'46), Ensign, U.S.N.; 322 E. 241st St., New York 66, N.Y.
Krahn, Leo G. ('29; '34; 35) (MJDCKL), Ch. Engr., Stand. Asbestos Mfg. & Insulating Co., 410 N. Olive St., Kansas City; *for mail,* 325 Ward Pkwy., Kansas City 2, Mo.
Krahulec, Fred (J'45) (ZNMYBC), Mech. Design Engr., A. C. Nielsen Co., 2101 W. Howard St., Chicago; *for mail,* 1919 W. Berteau Ave., Chicago 13, Ill.
Krajewski, Milton (J'42) (BKY), Mech. Engr., Kellex Corp., 233 Broadway, New York, N.Y.; *for mail,* 70 Essex St., Jersey City 2, N.J.
Kram, Harvey I. (J'43) (MONCJB), Pres., Cons. Engr., Kram Engrg. Associates, 475—5th Ave., New York; *for mail,* 175 Beach 148th St., Neponsit, L.I., N.Y.
Kramer, Andrew Anthony ('20), Pres., Columbian Steel Tank Co., 1509 W. 12th St., Kansas City 7, Mo.
Kramer, Bernard Louis (J'41) (NOHKSF), Lt., U.S.N.R., Naval Inspr. (Mchy.), c/o Asst. to Indus. Mgr., Navy Dept., 3050 N.W. Front, Portland, Ore.; *for mail,* 22 Ramsey Pl., Albany, N.Y.
Kramer, Chester Walter (J'27) (SHFEKL), Application & Field Serv. Engr., Worthington Pump & Mchy. Corp., 417 Watts Bldg., Birmingham 3, Ala.
Kramer, Delbert F. (J'36), 16881 Pierson Ave., Detroit 19, Mich.
Kramer, Frank Kipp, Jr. (J'41) (FSW), 714 W. Main St., Elizabeth City, N.C.
Kramer, Harold K. ('24; A'39) (C), Asst. V.P., Borden Co., 350 Madison Ave., New York 17; *for mail,* 27 Willets Court, Plandome, N.Y.
Kramer, Karl S. ('39) (SNJB), Asst. Sec. Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 624 Delaware Ave., Norwood, Pa.
Kramer, Leo Judson ('38; '45) (MLCNYD), Engr., Product Devel., Raytheon Mfg. Co., 1065 Commonwealth Ave., Boston; *for mail,* 31 Auburn St., Brookline 36, Mass.
Kramer, Stanley Frederick (J'43) (ECM), Ensign, U.S.N., c/o Indus. Mgr., 8th Naval Dist., 1 Canal St.; *for mail,* 5347 Coliseum, New Orleans, La.
Kramer, Vincent Joseph (J'41) (MCA), Asst. Ch. Engr., Smith & Mills Co., 2889 Spring Grove Ave., Cincinnati 25; *for mail,* 4113 Smith Rd., Norwood 12, Ohio.
Kramer, Wilbur C. ('37) (SFKH), Effic. Supt., Chicago Dist. Elec. Generating Corp., Box 65, Hammond, Ind.; *for mail,* 10622 Ave. E, Chicago 17, Ill.
Kramer, William Henry, Jr. ('44) (EFSKLC), Sales Engr., Phillips Petroleum Co., 80 Broadway, New York 5, N.Y.; *for mail,* 362 Thomas St., Phillipsburg, N. J.
Kramlich, C. W. ('29), 828 N. Broadway, Milwaukee, Wis.
Kranich, Henry Oscar ('36) (BMN), Engr., Libbey-Owens-Ford Glass Co., E. Broadway, Toledo; *for mail,* 2454 Hempstead Rd., Toledo 6, Ohio.
Krannert, Herman C. ('17; '35) (CDW), Pres., Inland Container Corp., Indianapolis, Ind.
Krantz, Clayton John (J'43), 3018 Logan Ave. N., Minneapolis 11, Minn.
Krantz, LeRoy J. (J'40) (ABK), Estimating Engr., Burge Ice Mch. Co., 218-20 N. Jefferson St., Chicago; *for mail,* 122 S. Cuyler Ave., Oak Park, Ill.

Krase, John Miles (J'44) (KNBCAF), 1234 Mt. Vernon Ave., Dayton 5, Ohio.
Krasner, Morton Harold (J'44), 16 Boston St., Malden, Mass.
Kratz, Alonzo Plumsted ('13; '21) (EFK), Research Prof., Dept. of Mech. Engrg., Univ. of Ill., Urbana, Ill.
Kratzer, James Carl (J'35) (FEABK), Div. Head, Research Lab., Linde Air Products Co., E. Park Dr., Tonowanda; *for mail,* 394 Washington Ave., Kenmore 17, N.Y.
Kraus, Benjamin Franklin ('44) (CSRTHA), Dir. of Inspc., Inspc. Dept., Assoc. Factory Mutual Fire Ins. Cos., 184 High St., Boston 10; *for mail,* 246 Fairmount Ave., Hyde Park 36, Mass.
Kraus, Charles Edward ('33; '35; '35) (JMC), Owner, Kraus Engrg. Co., 228 Winton Rd. N.; *for mail,* 100 Edgemoor Rd., Rochester 10, N.Y.
Kraus, Charles Elmer ('23) (CHM), Engr., Springfield Mch. Tool Co., W. Southern Ave.; *for mail,* R.D. 1, S. Tecumseh Rd., Springfield, Ohio.
Kraus, Gerard (J'43) (YJZ), Research Fellow, Applied Sci. Research Lab., Univ. of Cincinnati, Cincinnati 21, Ohio.
Kraus, Milton Nathaniel ('36; '45) (BES), Asst. Plant Engr., City Plants, Sheffield Farms Co., Inc., 524 W. 57th St., New York; *for mail,* 74 Clinton Pl., New York 53, N.Y.
Kraus, William Robinson (J'38) (MCDN), V.P., Prod. Mgr., Plant 2, Wm. Powell Co., 3233 Colerain Ave., Cincinnati; *for mail,* 1204 N. Bend Rd., Cincinnati 24, Ohio.
Krause, James Lawrence (J'42) (NMSDJC), Mech. Engr., T. C. Wheaton Co., 3rd & G St.; *for mail,* 518 E. Main St., Millville, N.J.
Krause, Otto C., Jr. (J'37) (SZFKBH), Economy Engr., Canal Sta., Louisville Gas & Elec. Co., 2005 Northwestern Pkwy., Louisville 3, Ky.
Krause, Robert ('36; '41) (LCMSDO), Asst. Plant Mgr., Container Corp. of Am., 404 E. N. Water St., Chicago 11; *for mail,* 17827 Tipton St., Homewood, Ill.
Krause, Robert Martin ('30; '35) (AECMJN), Prod. Engr., Gen. Elec. Co., 920 Western Ave.; *for mail,* 38 Manning Rd., Lynn, Mass.
Krauss, Arthur H. ('20; '23; '35) (SFE), Mech. Engr., Ebasco Services, Inc., 2 Rector St., New York 6, N.Y.; *for mail,* 254 Lenox Ave., South Orange, N.J.
Krauss, Harry Edward, Jr. ('45) (OCMJN), Ch. Engr., Dennis Mitchell Industries, 4424 Paul St., Philadelphia; *for mail,* 7417 Miller Ave., Upper Darby, Pa.
Kraut, Hans B. ('16) (CHN), Chmn., Bd. of Dirs., Giddings & Lewis Mch. Tool Co.; *for mail,* 15 S. Park Ave., Fond du Lac, Wis.
Krauth, Joseph Anthony, Jr. (J'42) (SKHR), 8504—123rd St., Richmond Hill 18, N.Y.
Krautheim, Wm. Chas. ('42) (RBJ), Asst. Mech. Engr., Gen. Steel Castings Corp., Granite City, Ill.; *for mail,* 7522 Forest View, St. Louis, Mo.
Kreamer, William Howden (J'37) (NTKLMB), Staff Engr., Am. Viscose Corp.; *for mail,* R.D. 6, Meadville, Pa.
Krebs, Edward P. (J'40) (BJM), 1726 Hollywood St., N.E., Warren, Ohio.
Krebs, Frank J. (J'28) (CMY), Engr., U.S. Rubber Co., 1230—6th Ave., New York 20, N.Y.; *for mail,* 603 Sunderland Rd., West Englewood, N.J.
Krehbiel, Fred. A. ('02; '30) (DEKMNS), Owner, Fred. A. Krehbiel Engrs., Rm. 1454, 222 W. Adams St., Chicago 6, Ill.
Krehbiel, Homer C., Jr. (J'37), Looper, Bethlehem Steel Co., W. Andover & 28th Ave., S.W., Seattle; *for mail,* 4600 W. Lander St., Seattle 6, Wash.
Kreiberg, Theodore N. (J'38) (OCYGNJ), Elec. Engr., U.S. Bur. of Reclamation, Redding, Calif.
Kreidler, D. W. ('17), Engrg. Staff, Hudson Motor Car Co., Detroit; *for mail,* 633 Lincoln Rd., Grosse Pointe, Mich.
Kreiner, Anthony James (J'42) (AHBE), Asst. Gen. Mgr., Designers for Industry Inc., 2915 Detroit Ave., Cleveland 13; *for mail,* 399 Lloyd Rd., Wickliffe, Ohio.
Kreisinger, Robert H. (J'38) (BFS), Gilbert Associates, Inc., 412 Washington St., Reading; *for mail,* 19 Woodland Rd., Wyomissing, Pa.
Kreith, Frank (J'46), Research Engr., Inst. of Tech.; *for mail,* 722 N. Westbourne, Los Angeles 46, Calif.
Kreitler, Frank C., Jr. (J'38) (MJNCRB), Requisition Engr., Trans. Motor Eng. Div., Gen. Elec. Co., 2901 E. Lake Rd., Erie; *for mail,* 317 E. Main St., Girard, Pa.
Krejci, Emil Louis (J'26) (CRDM), Works Mgr., Am. Steel Fdys., 4831 Hohman Ave., Hammond, Ind.; *for mail,* 2548 Indiana Ave., Lansing, Ill.
Kremer, Waldemar R. ('12) (CEFSYZ), 3348 N. Lake Dr., Milwaukee 11, Wis.
Kremser, Alois ('43) (FEACSY), Tech. Adviser to Pres., Stand. Oil Co. of Calif., 225 Bush St., San Francisco; *for mail,* Box 66, Alamo, Calif.
Kresser, Leo ('22) (BA), 114 Albermarle St., Springfield 9, Mass.
Kretzmer, Minor Garfield, Jr. (J'44) (BCZNKD), Research & Devel. Engr., Hartford-Empire Co., Homestead Ave., Hartford, Conn.
Kreuter, Verner Conrad (J'38) (CDJMNS), Engr., Am. Laundry Mchy. Co., 110 Buffalo Rd., Rochester 11, N.Y.
Krewson, William Lowley (J'44), 93 S. Rolling Rd., Springfield, Pa.
Krider, Harry Read (J'33) (CMS), 941 Rosewood Dr., Grand Rapids 6, Mich.
Krieg, Edwin Holmes ('25; '33) (SFKEBL), Cons. Mech. Engr., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.; *for mail,* 40 Sherwood Rd., Ridgewood, N.J.
Krieg, J. L. ('43) (JK), 9321—202nd St., Hollis, N.Y.
Kriegsheim, Heinrich ('24) (SBJZ), Retired; 200 W. 86th St., New York 24, N.Y.
Kriek, Peter Paul ('30) (TSL), Ch. Engr., Am. Enka Corp., Enka, N.C.
Kriger, Ralph Stacy ('44) (NHMBD), Engr., Mech. Research Dept., Corning Glass Works; *for mail,* 82 E. 3rd St., Corning, N.Y.
Krinner, Ludwig (J'44) (BEJKLS), Mech. Designer, Foster Wheeler Corp., 6 Church St.; *for mail,* 155 W. 76th St., New York, N.Y.
Krishnamurthy, B. S. ('28; '35), Gen. Mgr., Mysore Implements Factory, Hassan, Mysore State, South India.
Kristl, Franklin R. (J'46), 1116 E. 3rd Ave., Mishawaka, Ind.
Kriva, John Anton (J'43) (CODNMJ), Plant Layout Engr., J. I. Case Co., 700 State St.; *for mail,* 2316 Ashland Ave., Racine 2, Wis.
Kroder, Ernest Andrew (J'45) (DCFHZB), Plant Layout Engr., Bridgeport Brass Co., 30 Grand St., Bridgeport 2; *for mail,* 18 Hadley St., Bridgeport 8, Conn.
Kroeger, Edwin John (J'37) (ACKFYJ), Branch Head, Air, Sea Rescue & Safety Equip., Engrg. Div., 2W40 Navy Dept., Washington 25, D.C.
Krohn, Philip (J'43) (MCS), 2766 Barnes Ave., Bronx 67, N.Y.
Krol, Peter Frank ('43) (EJKHCF), Engr., Tide Water Associated Oil Co., E. 22nd St., Bayonne; *for mail,* 16 Doering Way, Cranford, N.J.
Kroll, Joseph (J'41) (ACM), Lt., Box 224, Advanced Cadet Det., Selman Field, Monroe, La.
Kroll, Leonard Arthur (J'35) (NJRM), Mch. Shop Instr., Gompers Vocational High Sch., 455 South Blvd., Bronx 55; *for mail,* 944 E. 13th St., Brooklyn 30, N.Y.
Kromer, William F. ('24) (RELOB), Ch. Mech. Engr., Charge of Sales & Engrg., H. K. Porter Co., Inc., 49th & Harrison Sts., Pittsburgh 6, Pa.
Kroner, Ernst Frederic ('28; '35) (AEH), Gen. Delivery, Fryeburg, Me.
Kroon, R. P. ('32; '41) (ABH), Mgr., Devel. Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* Box 308, 1141 Muhlenberg Ave., Swarthmore, Pa.
Krooss, John H. (J'25) (YHJB), Ch. Engr., Victaulic Co. of Am., 30 Rockefeller Plaza, New York 20, N.Y.
Kropp, Einar (J'43) (LKHFEG), Gp. Engr., Tech. Serv. Div., Stand. Oil Co. of Ohio, Midland Bldg., Cleveland 15; *for mail,* County Line Rd., Gates Mills, Ohio.
Kropper, Herman Julius ('43) (HESC), Prin. Engrg. Asst., Chief of Engrg. Div., Providence Dist., U.S. Engr. Dept., War Dept., 819 Indus. Trust Bldg., Providence 2, R.I.
Krosse, George Theo. ('40) (ZSKNF), Effic. Engr., Cent. Ill. Light Co., 1126 W. Washington St., Peoria 8; *for mail,* 1109—1st Ave., Peoria 6, Ill.
Kroto, George ('08) (CEFLMY), 43 Washington Ave., Westport, Conn.
Kroto, Stanley George (J'40) (JND), Plant Engr., Rolock, Inc., Fairfield; *for mail,* 43 Washington Ave., Westport, Conn.
Kroupsky, Vladimir Alexander (J'42), Exper. Machinist, W. L. Maxson Corp., 460 W. 34th St., New York 1; *for mail,* 224 E. 11th St., New York 3, N.Y.
Kruchen, William S. (J'44) (CMTDG), Prod. Mgr., Troy Laundry Co., 5th & Yarnall St.; *for mail,* 1400 Honan St., Highland Gardens, Chester, Pa.
Krueger, Forrest J. ('22; '35) (CJS), Engrg. Mgr., Natl. Aniline Div., Allied Chem. & Dye Corp., 40 Rector St., New York 6, N.Y.
Krueger, George Arthur (J'43) (AMS), 10130 S. Leavitt St., Chicago 43, Ill.
Krueger, Harold Frederick ('29; '38) (CDMJO), Indus. Engr., Murray Ohio Mfg. Co., 1115 E. 152nd St., Cleveland 10; *for mail,* 3105 Keswick Rd., Shaker Heights 20, Ohio.
Krueger, John Wood ('45) (CDM), Gen. Supt., Plant 2, Frigidaire Div., Gen. Motors Corp., Taylor St., Dayton 1; *for mail,* 204 Lookout Dr., Dayton 9, Ohio.
Krueger, Joseph William (J'35) (EAC), 3022 N. Cramer, Milwaukee 11, Wis.
Krug, German L. (J'45) (EKM), Cia. Salitrera de Tarapaca Y Antofagasta, Casilla 7-D, Iquique, Chile, S.A.
Kruger, Paul F. ('22), Gen. Mgr., Cia. Salitrera Anglo-Chilena, Casilla 96-D, Santiago, Chile, S.A.
Kruger, Walter Herbert (J'43) (ALM), 803—9th, Helena, Mont.
Krummel, Louis C. ('01), Brightwaters, L.I., N.Y.
Krupnick, Sydney R. ('42) (HLN), Ch. Engr., Harsyd Chemicals, Inc., 397 W. 21st St., Holland, Mich.
Krupp, John Oscar (J'39) (AMYC), Research Analyst, No. Am. Aviation, Inc., Imperial Highway, Inglewood, Calif.
Kruse, John F. (J'39) (MCO), Staff Indus. Engr., Colt's Pat. Fire Arms Mfg. Co., 15 Van Dyke Ave., Hartford 17; *for mail,* 8 Woodrow St., West Hartford 7, Conn.
Kruse, John R. ('38) (EFKSBJ), Ch. Engr., Mar. Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Kruse, Lowell Frederick (J'35) (CDML), Supvg. Engr., Stevenson, Jordan & Harrison, Inc., 19 W. 44th St., New York 18, N.Y.
Krygier, John Walter (J'45) (ECHMNB), Sr. Engr., Detroit Diesel Eng. Div., Gen. Motors Corp., 13400 W. Outer Dr., Detroit 23; *for mail,* 3596—29th, Detroit 10, Mich.
Ku, Y. C. ('45), Dir., Natl. Bur. of Indus. Research, Chungking, China; *for mail,* c/o Prof. P. B. Eaton, Lehigh Univ., Easton, Pa.
Kuba, George (J'35) (SNCOHJ), Equip. Engr., E. I. du Pont de Nemours & Co., Wilmington 99; *for mail,* 608 W. 20th St., Wilmington 223, Del.
Kubacki, Wallace (J'37) (CMBHNZ), Research & Devel. Engr., with R. M. Urquhart, R.D., Wayne; *for mail,* 606 Bingaman St, Reading, Pa.
Kubli, Robert Arnold (J'44) (JKRNAM), Devel. Engr., Linde Air Products Co., 646 Frelinghuysen Ave., Newark; *for mail,* 211 Belleview Terrace, Hillside, N.J.
Kucera, John J., Jr. (J'41) (CMB), Foreman, Union Spec. Mch. Co., 400 N. Franklin St., Chicago; *for mail,* 2144 N. Kilbourn Ave., Chicago 39, Ill.
Kuchler, T. C. ('29; '35; '35) (BER), Baldwin Loco. Works, Eddystone; *for mail,* 308 Oak Ave., Sharon Hill, Pa.
Kuechler, Gustave Henry ('45) (ADFGSW), Export Div., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 9130 Springfield Blvd., Queens Village 8, N.Y.
Kuehling, Irwin V. (J'41) (DKL), 2843-A Victor St., St. Louis, Mo.
Kuehn, Howard Edwin (J'44) (TNYACG), Asst. to Ch. Engr., Atwood Div., Farrel-Birmingham Co., Inc., Water St.; *for mail,* 23 Elm St., Stonington, Conn.
Kuehn, Hugo Richard (J'17) (LFBMSJ), Mech. Engr., Walgreen Co., 744 Bowen Ave., Chicago 15; *for mail,* 5916 W. Superior St., Chicago 44, Ill.
Kuehn, Kurt F. (J'40) (EFSJK), Assoc. Mar. Engr., (Diesel Option), Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* 1025—17th St. S., Arlington, Va.
Kuen, William E. ('15; '35) (SEFN), 462 Forrest Ave., Drexel Hill, Pa.
Kuenzel, Carl John (J'36) (BDKSZ), Equip. Engr., Plant Engrg. Dept., Am. Seating Co., 9th & Broadway, Grand Rapids 2; *for mail,* 920 Chatham St., Grand Rapids 4, Mich.
Kuenzel, Herbert ('30; '42) (JKSBMN), Asst. Prof. Mech. Engrg., Washington Univ., Skinker & Lindell Blvds., St. Louis 5; *for mail,* 645 N. Forest Ave., Webster Groves 19, Mo.
Kugel, H. K. ('13; '22) (FSE), Ch. Engr., Div. of Smoke Regulation & Boiler Inspc., Dist. of Columbia Govt., Dist. Bldg., Washington 4, D.C.
Kugler, Arthur Noble ('25; '37) (JSBNAM), Mech. Engr., Tech. Sales Div., Air Reduction Sales Co., 60 E. 42nd St., New York 17, N.Y.; *for mail,* 118 Sherwood Rd., Ridgewood, N.J.
Kuh, Louis Morton (J'46) (COZ), Instr. in Engrg. Drawing, Univ. of Colo., Boulder, Colo.
Kuhlen, Fred'k ('16; '23; '35), Assoc. Prof., Chmn., Mech. Engrg. Dept., College of City of N.Y., New York, N.Y.; *for mail,* 491 Main St., Hackensack, N.J.
Kuhler, Otto Augustus ('40) (BGRY), Design Engr., Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.
Kuhn, George Wilfrid ('22; '25) (ACMHKE), Mech. Engr. P-1, N.Y. Naval Shipyard, Brooklyn; *for mail,* 471 Willoughby Ave., Brooklyn 6, N.Y.
Kuhn, Shannon ('43) (FRS), Deputy Dir., Office of Defense Trans., Rm. 1121, Interstate Commerce Comm. Bldg., Washington, D.C.; *for mail,* 1270 Cleveland Heights Blvd., Cleveland 21, Ohio.
Kuhn, Werner Willy (J'46) (ABL), 283 Eastern Ave., Springfield, Mass.
Kuhner Max Henry ('37) (SFKE), Ch. Mech. Engr., Riley Stoker Corp., 9 Neponset St., Worcester 8, Mass.
Kuhnow, Bernard F. L. ('22; '35) (SEFKB), Mech. Engr., City of New York Dept. of Pub. Works, 1400 Municipal Bldg., New York 7; *for mail,* 666 S. 7th Ave., Mt. Vernon, N.Y.
Kuhns, Joseph Hay (J'40) (LSROFM), Supvr. Mech. Utilities, Electrochemicals Dept., E. I. du Pont de Nemours & Co., Buffalo Ave.; *for mail,* 2709 Linwood Ave., Niagara Falls, N.Y.
Kuhns, Robert Lake ('39; '46) (SKCBLF), Maint. Div. Supt., E. I. du Pont de Nemours & Co., Box 993, Charleston; *for mail,* 106—52nd St, S. E., Charleston 4, W. Va.
Kuldell, Rudolph C. ('23) (C), 5100 Fannin St., Houston, Tex.
Kulick, Sidney (J'43) (CJDMNB), Asst. Secy., Michael Flynn Mfg. Co., 2263 E. Allegheny Ave., Philadelphia 34, Pa.
Kulieke, Frederick Charles, Jr. (J'41) (RJNMCB), Lt. (j.g.), U.S.N.R., Sec. Re2G, Rm. 0419, Bur. of Ord., Navy Dept., Washington, D.C.; *for mail,* 511 S. Lincoln Ave., Alliance, Ohio.
Kulig, Constantine Walter (J'44) (ACK), 27 Moore St., Central Falls, R.I.
Kulik, Nicholas (J'41) (BEK), B-H Aircraft Co., Inc., 27-01 Bridge Plaza N., Long Island City 1, N.Y.

Kuljian, Harry A. ('21; '24; '27; F'43) (EFLO ST), Owner, H. A. Kuljian & Co., 1518 Walnut St., Philadelphia 2, Pa.
Kulka, Siegmund (J'44) (ZCHAW), Exper. Engr., Engrg. Labs., Vickers, Inc., 1400 Oakman Blvd., Detroit; *for mail,* 2070 W. Euclid, Detroit 6, Mich.
Kullmer, Frank ('32; '35) (CMB), Ch. Engr., Soule Steel Co., 1750 Army St., San Francisco 24, Calif.
Kumming, Emil ('21; '35), Charge Mech. Engrg. Div., Century Elec. Co., 1806 Pine St., St. Louis, Mo.
Kun, Irving B. (J'43) (AHL), Engr., I. P. Morris Dept., Baldwin Loco. Works, Eddystone; *for mail,* 5624 Lebanon Ave., Philadelphia 31, Pa.
Kunen, Alfred Edward (J'41) (AJ), Cons. Mech. Engr., 17 E. 42nd St., New York, N.Y.
Kunen, Herbert Kenneth (J'38) (AO), Partner, Kunen Engrg. Co., 17 E. 42nd St., New York 17, N.Y.
Kunkel, George Merrill ('21; '37) (NLDC), Assoc. Prof., Mech. Engrg. Dept., Bucknell Univ.; *for mail,* 118 Brown St., Lewisburg, Pa.
Kunkel, Raymond Erich (J'41) (SFHKDC), Asst. to Power Engr., Am. Smelting & Refining Co., Garfield; *for mail,* 1846 Euclid Ave., Salt Lake City 4, Utah.
Kuntny, Gregor Walter ('41) (BMNZ), Brown Instrument Co., Wayne & Roberts Ave., Philadelphia 44; *for mail,* 56 W. Gowen Circle, Mt. Airy, Philadelphia 19, Pa.
Kunz, W. E. (J'40), 1017 Bales Ave., Kansas City, Mo.
Kunz, Warren (J'44) (ASCEKZ), Asst. Project Engr., Exper. Jet Propulsion, Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 46 Wyllys St., Glastonbury, Conn.
Kunz, William J. ('16; '19; '35) (CFKS), Mgr. of Drafting, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 15 Essex Court, Port Washington, L.I., N.Y.
Kuppenheimer, John Daniel ('32; '44) (SKF), Proposition Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Kuprick, William (J'41) (AEM), c/o Runyon Park, 38 MacArthur Dr., Fords, N.J.
Kurkjian, Armen Stapan ('22; '35) (BCW), Partner & Sales Mgr., Oliver Mchy. Co., 1025 Clancy St., N.E., Grand Rapids 2; *for mail,* 552 Gladstone St., S.E., Grand Rapids 6, Mich.
Kurrein, Max ('34) (LMJZBC), Prof., Indus. Engrg., Supvr. Workshops, Hebrew Tech. Inst., Box 910; *for mail,* Box 955, Haifa, Palestine.
Kurth, Carl Hanns ('28) (BEM), Cons. Engr., Fulton Iron Works Co., Delaware Ave., St. Louis, Mo.
Kurth, Franz J. ('38), Tech. Dir., Anemostat Corp. of Am., 10 E. 39th St., New York; *for mail,* 510 Cortlandt Ave., Mamaroneck, N.Y.
Kurtis, Alan Abraham (J'43) (AFM), 1039 Intervale Ave., New York, N.Y.
Kurtz, Henry Frederick ('37) (BCM), Sr. Supervisory Engr., Bausch & Lomb Optical Co., 635 St. Paul St., Rochester 2, N.Y.
Kurtz, John Winfield ('40) (GJNOKB), Head, Engrg. Dept., Univ. of Omaha, 62nd & Dodge, Omaha; *for mail,* 5204 Jones St., Omaha 6, Neb.
Kurzinski, Edward Francis (J'42) (JNKF), Ensign, U.S.N.R., Commanding Officer, LSM-87, F.P.O., San Francisco, Calif.; *for mail,* 19 St. Marks Pl., Linden, N.J.
Kurzweil, Robert (J'43) (NOMJHR), Ch. Engr., Morey Mchy. Co., 4-57—26th Ave., Astoria 2; *for mail,* 84-10—126th St., Kew Gardens 15, N.Y.
Kushner, Albert (J'43) (NZMB), Jr. Research Engr., Carbide & Carbon Chemicals Corp., 3280 Broadway, New York 27; *for mail,* 615 Ocean Ave., Brooklyn 26, N.Y.
Kushnick, Jerome L. (J'42) (KAHMFB), Ensign, U.S.N.R., 1914 University Ave., Palo Alto, Calif.
Kushnick, William Henry ('26; '37) (CD), Exec. Dir., Wallace Clark & Co., 521—5th Ave., New York; *for mail,* 60 Algonquin Rd., Tuckahoe, N.Y.
Kuska, Milton (J'43) (ACNBEM), Flight Test Engr., Airplane Div., Curtiss-Wright Corp., Columbus; *for mail,* 133 W. Oakland Ave., Columbus 1, Ohio.
Kuska, Norman (J'43) (A), Detailer, Curtiss-Wright Corp., Columbus; *for mail,* 1448 Neil Ave., Columbus 1, Ohio.
Kussman, William (J'44) (OJFKRY), Ensign, U.S.N., "F" Div., U.S.S. *Duluth,* F.P.O., San Francisco, Calif.; *for mail,* 23-52—122nd St., College Point, N.Y.
Kustas, George J. (J'40) (BKM), Apprentice Die Designer, Doehler Die Casting Co., Box 400; *for mail,* 18 Richmond Ave., Batavia, N.Y.
Kuszczak, Leo (J'43) (BNAJHS), Interviewer, Counselor, Detached Enlisted Men's List, 2nd Serv. Comd., 18th Co., Counseling Sec., Separation Center, Ft. Dix, N.J.
Kut, Walter Stanley (J'38) (EKAFS), Lt. (j.g.), A-V(S), U.S.N.R., Naval Air Sta., Box G, Key West, Fla.; *for mail,* 1 Landry St., New Bedford, Mass.
Kutaj, Walter Casimar (J'44) (BNAGKO), Struc. Steel Draftsman, M. H. Detrick Co., 111 W. Washington St., Chicago 2; *for mail,* 2311 S. Whipple St., Chicago 23, Ill.
Kuthe, Charles H. (J'32) (ABJN), Tech. Adviser, Revere Copper & Brass, Inc., 5851 W. Jefferson Ave., Detroit 9, Mich.
Kutter, Herman L. ('02; '07), Pres., Gen. Mgr., Black-Clawson Co., 2nd & Vine Sts.; *for mail,* R.R. 6, Hamilton, Ohio.
Kutter, Rudolf Ludwig (J'35) (CMJ), Asst. to Pres., Black-Clawson Co., 2nd & Vine Sts.; *for mail,* 811 Lawn Ave., Hamilton, Ohio.
Kuttler, John B. ('28; '34) (SFCZJK), Asst. Ch. Engr., Prudential Ins. Co. of Am., 763 Broad St., Newark 2; *for mail,* 642 Scotch Plains Ave., Westfield, N.J.
Kutz, Harry Russell ('44) (CDMY), Brig. Gen., U.S.A., Chief, Military Plans & Trng. Serv., Office of the Chief of Ord., War Dept., 5D-400 The Pentagon, Washington 25; *for mail,* 1200—16th St., N.W., Washington 6, D.C.
Kutzelman, Elmer George (J'42) (CMSJB), Planning Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 190 Bergen Ave., Jersey City 5, N.J.
Kuvin, Leonard ('45), Administrative Asst. to Pres., Tubize Rayon Corp., 2 Park Ave., New York 16, N.Y.
Kuwada, Gompei ('00) (JMT), 3 Yoshida Kaguraokacho, Sakyoku, Kyoto, Japan.
Kux, Andrew ('44) (EHK), c/o Occidental Elec. Co., Box 466, 409 Grand Ave., Nogales, Ariz.
Kuylenstjerna, Adolf Leonard ('22), Retired; 232 Iona Ave., Narberth, Pa.
Kuzman, Joseph Anthony (J'44) (ABOYNM), Electronics & Mechanics Exper. Work, 354 Cornelia St., Brooklyn 27, N.Y.
Kwang, Kwong Yung ('99), 39 Race Course Rd., British Concession, Tientsin, China.
Kyburz, Walter Warren ('40) (LMZCOY), Test Engr., Swift & Co., Packers & Exchange Ave.; *for mail,* 4243 N. Winchester Ave., Chicago 13, Ill.
Kyhl, Louis Christian ('44) (NODKHB), Cons. Engr., with Irwin Koenig, 714 Olympic Blvd., Los Angeles 7, Calif.
Kyle, John J. (J'42) (BJKLMS), Project Leader, W. K. Mitchell & Co., Inc., 2940 Ellsworth St.; *for mail,* 2821 Annin, Philadelphia 46, Pa.
Kyle, Joseph Hertzler (J'41) (MCSERW), Reedsville, Pa.
Kyle, Peter Edward ('44) (JBN), Assoc. Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge 39, Mass.
Kyropoulos, Peter R. (J'40) (EAHKBS), Instr. in Mech. Engrg., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.

L

Laabs, Eric Harry ('17; '22; '29) (BJG), Supvg. Engr., Cutler-Hammer, Inc., 12th & St. Paul Ave., Milwaukee 1; *for mail,* 2365 N. 91st St., Wauwatosa 13, Wis.
LaBarre, Floyd, Jr. (J'41) (CMABGS), Unit Head, Tech. Data Field Serv. Dept., Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* Box 126, 140 Fox Hill Rd., Rainbow Lakes, Boonton R.D., N.J.
Labastie, Albert Henry (J'46) (BHJ), Asst. Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 2 Village Hill Rd., Belmont, Mass.
Labberton, John Madison ('37) (SOKRF), Prof. Mar. Engrg., N.Y. Univ., 180th St.; *for mail,* 114 W. 183rd St., New York 53, N.Y.
Laboissonniere, Eugene Wilfred (J'42) (BNMJ HY), Austin Ave., Greenville, R.I.
Laboulais, Jean Louis ('36) (BHKNSZ), Project Engr., Columbia Mch. Works, 255 Chestnut St., Brooklyn 8, N.Y.; *for mail,* 25 W. Elm St., Greenwich, Conn.
Labounsky, Nicholas N. (J'36) (BMS), Ch. Engr., Keystone Driller Co., 2001-2021—8th Ave., Beaver Falls, Pa.
LaBreque, R. J. (J'34) (CLT), Secy., Dir., Prod. Mgr., Acton Vale Silk Mills, Ltd., Acton Vale, Que., Can.
LaCerda, Donald Joseph (J'44) (SKBOEN), Master Sgt., Army Air Forces, Photogrammetrist, 64th Engr. Topographic Bn., A.P.O. 181, Tokyo, Japan, c/o Postmaster, San Francisco, Calif.; *for mail,* 165 E. Cottage St., Dorchester 25, Mass.
Lachman, Louis Albert (J'43) (TNBJZ), Devel. Engr., Atwood Div., Farrel-Birmingham Co., Inc., Water St., Stonington; *for mail,* 202 Thames St., New London, Conn.
LaClair, Darwin John (J'44) (OLMDZ), Sales Engr., Alan Porter Lee Inc., 136 Liberty St., New York 6; *for mail,* 7200 Ridge Blvd., Brooklyn 9, N.Y.
LaCroix, Arthur Joseph ('14; '35) (CMJ), Pres., Charge Prod. & Pur., Hyde Mfg. Co., 54 Eastford Rd., Southbridge, Mass.
LaCrosse, Emmart ('21) (CHLOSY), V.P., Dir. of Engrg., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
LaCrosse, Emmart, Jr. (J'45) (DOC), Sales Engr., Link-Belt Co., 300 W. Pershing Rd., Chicago, Ill.; *for mail,* 6136 Crittenden Ave., Indianapolis 5, Ind.
Lacy, James Wortham (J'38) (NEKA), Asst. Prof. Mech. Engrg., Sch. of Engrg., So. Methodist Univ., Dallas 6, Tex.
Ladd, George Herbert ('31; '44) (CDLMOY), Works Mgr., Inland Container Corp., Nelbar; *for mail,* 510—14th Ave., Middletown, Ohio.
Laderach, Otto William (J'43) (BMHEKS), Lt. Comdr. (E), Royal Canadian Voluntary Reserve, H.M.C.S. *Warrior,* c/o G.P.O., London, England; *for mail,* 62 Lansdowne Ave., Peterborough, Ont., Can.
Laemle, Milton Myrtill ('25; '33) (CTM), Asst. Mgr., Jessup Plant, Barbizon Corp., 148 Madison Ave., New York; *for mail,* 84-51 Beverly Rd., Kew Gardens, N.Y.
Laessle, Frank Willard (J'42) (BGM), Ensign, U.S.N.R., Navy 1925 (Salvage), F.P.O., New York, N.Y.
La Fetra, Clinton W. ('20; '35) (CMN), Mgr., Estimating Dept., Ford Instrument Co., Long Island City; *for mail,* 103 Kellogg St., Oyster Bay, N.Y.
Lafferty, Edward C. (J'46) Hoppes Mfg. Co.; *for mail,* 2000 Maiden Lane, Springfield, Ohio.
Lafferty, Hugh C. (J'41) (EKR), 215 Dryden Rd., Ithaca, N.Y.
Laffoon, Carthrae Merrette ('44) (ABCJKS), Mgr., A.C. Engrg. Dept., Westinghouse Elec. Corp., East Pittsburgh, Pa.
LaFore, John A. ('04) (FHS), Retired; Penn Valley Farm, Narberth, Pa.
LaForge, Robert Mallory ('32; '45) (MWOD), Ch. Engr., U.S. Homes, Inc., Marietta, Ga.
La France, Raymond ('44) (ACM), V.P., Charge Plant Operas., S. K. Wellman Co., 1374 E. 51st St., Cleveland 13, Ohio.
LaGambina, Joseph Claude (J'45) (NYJHML), Asst. Project Engr., Process Research Dept., Sperry Gyroscope Co., Inc., Clinton Rd. & Stewart Ave., Garden City; *for mail,* 29 Clifford St., Malverne, N.Y.
Lager, Rollin A. (J'37) (SLEOWA), Application Engr., Steam Plant Equip. Co., 4433 W. North Ave., Milwaukee 8; *for mail,* 2846 N. Farwell Ave., Milwaukee 11, Wis.
LaGrange, William Albert (J'44) (JCNLZ), Application Engr., Carrier Corp., 710 N. Harwood St., Dallas 1, Tex.
La Grant, Roy George (J'45) (SMCJHB), Lt. (j.g.), U.S.N.R., Asst. Matl. Procurement Officer, Office of Asst. to Indus. Mgr., 11th Naval District, 3000 W. Seaside Dr., Terminal Island, Calif.
Lai, William (J'45) (ENSBFK), Engr., Cooper-Bessemer Corp., Mt. Vernon, Ohio; *for mail,* 553 Craig St., Grove City, Pa.
Lail, George Gray, Jr. (J'44), 264 Arden Rd., Pittsburgh 16, Pa.
Laiming, Harry John ('34; '44) (ABJNOY), Mech. Engr., RCA Victor Div., Radio Corp. of Am., Camden, N.J.; *for mail,* 1016 S. 45th St., Philadelphia 43, Pa.
Laird, Alan Douglas Kenneth (J'40) (BKH), Jr. Engr., Leek & Co., 1111 Homer St., Vancouver, B.C., Can.
Laird, Alton Wilson ('31; '35) (HBMNJA), Dir. of Engrg., Hyd. Div., N.Y. Air Brake Co., Starbuck Ave., Watertown, N.Y.
Laird, J. Packard (J'42) (ABG), *Student Award,* '42; Jr. Engr., Carl L. Norden, Inc., 80 Lafayette St., New York; *for mail,* 42 Claremont Ave., Mt. Vernon, N.Y.
Lais, Irwin Michel (J'41) (KJSBAM), Capt., U.S.A.; *for mail,* 5710 Memphis St., New Orleans 19, La.
Lake, Albert Homer ('43) (RBJNMD), Sr. Designer, Electro-Motive Div., Gen. Motors Corp., La Grange, Ill.
Lake, Charles W., Jr. (J'46), Engr. in Trng., R. R. Donnelley & Sons Co., 350 E. 22nd St., Chicago 16; *for mail,* 4623 Ellis Ave., Chicago 15, Ill.
Lake, Eugene Tilton ('44) (NHBJE), Leading Draftsman, with George G. Sharp, Naval Arch., 30 Church St., New York, N.Y.; *for mail,* 219 Central Ave., Cranford, N.J.
Lake, Robert B. (J'40) (BHM), Jr. Engr., John S. Barnes Co., 301 S. Water St.; *for mail,* 409 Royal Ave., Rockford, Ill.
Lake, Simeon T., Jr. (J'38) (KNCO), Engr., Tex. Pub. Utilities Corp.; *for mail,* 5328 Miller Ave., Dallas 6, Tex.
Lakey, Arthur B. ('13; '35) (BHSMJA), Ch. Engr., Mem. Mgmt. Com., Kingsbury Mch. Works, Inc., 4320 Tackawanna St., Philadelphia 24; *for mail,* 4504 Fitler St., Philadelphia 14, Pa.
Lamb, Albert Chester (J'38) (SC), Supt., Engrg. Dept., Ocean Accident & Guarantee Corp. Ltd., 315 Montgomery St., San Francisco 4, Calif.
Lamb, Donald B. (J'41) (NSM), Design Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia 42; *for mail,* 32 E. Springfield Rd., Springfield, Del. Co., Pa.
Lamb, George G. ('41) (AE), Prof., Dept. of Chem. Engrg., Northwestern Tech. Inst., Evanston, Ill.
Lamb, Hawthorne M. ('17; '35) (CMS), Plant Investment Engr., Mathieson Alkali Works, Inc., Box 896; *for mail,* 830—11th St., Lake Charles, La.
Lamb, Joseph Francis ('23; '28) (ER), Supreme Secy., Gen. Mgr., Knights of Columbus, P.O. Drawer 1670, New Haven 7, Conn.
Lambelet, Carl Henry (J'13) (CMD), V.P., Charge of Mfg., Am. Mch. & Fdy. Co., 511—5th Ave., New York 17, N.Y.; *for mail,* 563 Queen Anne Rd., Teaneck, N.J.
Lambert, Francis Moore (J'28) (BJK), 626 Moreno Rd., Penn Valley, Narberth, Pa.
Lambertine, Joseph A. (J'27) (LKMO), Cons. Engr., F. R. Hormann & Co., 186 Joralemon St., Brooklyn 2; *for mail,* 181-33 Kruger Rd., Jamaica 3, N.Y.

Lambertus, Harold (J'42) (JMCDSE), Lt., U.S.N.R., Calif. Inst. of Tech., Box 86, Pasadena 4, Calif.; *for mail,* 5273 N. Illinois St., Indianapolis, Ind.
Lambie, Aaron Louis ('38) (CLODMA), Asst. Treas., Blaw-Knox Co., Box 1198, Pittsburgh, Pa.
Lamm, Earl Stanley (J'45) (BCGDM), Pvt., Casual Co. 31, A.P.O. 26041, c/o Postmaster, San Francisco, Calif.
Lammi, Raymond Elias (J'45) (HCAKTN), 66 Leighton St., Fitchburg, Mass.
Lamont, Neil C. ('23) (CDM), Works Mgr., Natl. Elec. Products Corp., 14th St., Ambridge; *for mail,* 513 Maple Lane, Edgeworth, Pa.
LaMothe, Kenneth F. ('37), Ch. Estimator, Alco Products, Inc., 30 Church St., New York; *for mail,* 97 Webster Ave., Manhasset, N.Y.
La Motte, William R. ('26), Pub. Serv. Terminal, 80 Park Pl., Newark 1, N.J.
Lamson, Otis Floyd, Jr., (J'40) (CMNJAO), Owner, Mgr., Lamson Products Co., 365 E. 45th St., Seattle 5; *for mail,* 2157 Boyer, Seattle 2, Wash.
Lancaon, Juan S. (J'45) (BCM), V.P., Ch. Engr., Philippine Hawaiian Engrg. Co., Ltd.; *for mail,* 1117 O'Donnell St., Sta. Cruz, Manila, P.I.
Lancaster, John Earle (J'45), Jr. Hyd. Engr., Research & Devel. Dept., Worthington Pump & Mchy. Corp., Harrison, N.J.; *for mail,* 422 E. 58th St., New York 22, N.Y.
Lancaster, Robert Gail (J'43) (NESAKB), Packaging Engr., Naval Ord. Plant, 21st & Arlington; *for mail,* 1329 N. Wallace St., Indianapolis, Ind.
Lancaster, Warren Jewel (J'43) (BKHZ), Exper. Test Engr., Vilter Mfg. Co., 2217 S. 1st St., Milwaukee; *for mail,* 1903 East Park Pl., Milwaukee 11, Wis.
Land, Emory S. (H'44), Vice Admiral, U.S.N., Retired; Pres., Air Transport Association, 1515 Massachusetts Ave., N.W., Washington 5, D.C.
Land, George William (J'44) (FSRC), Research & Combustion Engr., West Ky. Coal Co., Earlington, Ky.
Land, Malcolm Louis (J'45), Ch. Analyst, Research & Devel. Dept., Elliott Co., Jeannette 1, Pa.
Lande, Clarence C. ('37) (CHM), Kimlark Assembly Plant Mgr., Ord. Div., Kimberly Clark Corp., Neenah; *for mail,* 618 W. Seymour St., Appleton, Wis.
Landen, Sigurd ('42) (CDFH), Ch. Engr., Gary Steel Works, Carnegie Ill. Steel Corp., N. Broadway; *for mail,* 224 W. 46th Ave., Gary, Ind.
Lander, Raymond Alfred, Jr. (J'44), 200 Corwin Rd., Rochester, N.Y.
Landes, Benjamin Daniel ('37) (CJOY), Partner, Landes, Zachary & Peterson, Denver, Colo.
Landes, Robert Jellison (J'44), Ensign, U.S.N.; 72 N. Wolcott, Salt Lake City, Utah.
Landes, Thayer E. (J'41) (MSJC), 6703 Dunham Ave., Cleveland 3, Ohio.
Landis, Charles William (J'39) (LGDSNM), Engrg. Draftsman, Sharp & Dohme, Inc., 640 N. Broad St., Philadelphia, Pa.
Landis, Franklin Forman (J'43) (NMCOHJ), Designing Engr., Monarch Mch. Tool Co., 17 June St., Dayton 3; *for mail,* 1515 Canfield Ave., Dayton 6, Ohio.
Landis, J. Noble ('25; '27; '33) (SFC), *Director-at-Large,* '45-'49; Asst. Mech. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Landis, John Prince (J'44) (AEF), Test Engr., Research & Devel. Labs., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail,* 42 Hopkins St., Woodbury, N.J.
Landis, Mark Homer ('13; '35) (NZLHD), Pres., Gen. Mgr., Erd Co., Inc., 225-233 Ringgold St., Waynesboro, Pa.
Landis, Robert Potts (J'35) (ANBCJM), Design Consultant, Serv. Dept., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford; *for mail,* 20 Jean Rd., Manchester, Conn.
Landis, Warren Corle ('42; '45) (EFASRC), Shell Oil Co., 50 W. 50th St., New York, N.Y.
Landon, Duane ('45) (MOECZS), Indus. Engr., St. Joseph Light & Power Co., St. Joseph 2; *for mail,* 3005 Jules, St. Joseph, Mo.
Landow, Ernest Wm. (J'39) (J), Ch. Engr., Castings Div., Universal Castings Corp., 5821 W. 66th St.; *for mail,* 5458 S. Menard Ave., Chicago 38, Ill.
Landry, Bertrand Andre ('42) (FKESRB), Asst. Supvr., Fuels Div., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Landvoigt, Thomas Elmer ('17) (KFGCS), Mech. Engr., Mech. Equip. Sec., Fed. Housing Admin., Washington, D.C.; *for mail,* 201 N. Jackson St., Arlington, Va.
Lane, Abbott Atwood ('17; '24; '35) (EMLSNO), Consultant, Prod. Dept., Am. Agric. Chem. Co., 50 Church St., New York 7, N.Y.; *for mail,* 572 Ridgewood Ave., Glen Ridge, N.J.
Lane, Buster Wallace (J'37) (AMC), Res. Elec. Engr., DeSoto-Warren Plant, Chrysler Corp., 8101 W. Warren Ave., Detroit 31; *for mail,* 124 Farrand Park, Highland Park 3, Mich.
Lane, Clarence William (J'44) (KM), 6205 S. Miro St., New Orleans 15, La.
Lane, Donald Frederick ('40; '44) (CNBDLR), Dir. of Trng., Bethlehem Steel Co., Sparrows Point, Baltimore 19; *for mail,* 2905 Dunglow Rd., Baltimore 22, Md.
Lane, Edward J. H. (J'39) (MYB), Ord. Engr., Naval Gun Factory, Naval Shipyard, Washington, D.C.; *for mail,* 5702 Greenleaf Rd., Cheverly, Md.
Lane, F. Howard ('23) (SEHF), Mgr., Design Engrg., Pub. Utility Engrg. & Serv. Corp., 231 S. La Salle St., Chicago 4, Ill.
Lane, Henry M. ('00), Cons. Engr., Gray Gables, Grosse Ile, Mich.
Lane, Philip Ellison (J'42) (CMKJ), Internatl. Mutoscope Corp., 44-01—11th St., Long Island City; *for mail,* Apt. 2G, 135 Hawthorne St., Brooklyn 25, N.Y.
Lane, Richard Kelvin ('36) (S), Pres., Pub. Serv. Co. of Okla., Box 201, Tulsa 2, Okla.
Lane, Robert S. ('21; '23; '35), Pocahontas Fuel Co., Inc., 340 E. 131st St., Cleveland 8, Ohio.
Lane, William T. ('42) (RAEB), Asst. West. Sales Mgr., Franklin Ry. Supply Co., 332 S. Michigan Ave., Chicago 4, Ill.
Laney, Frank R. ('18; '28) (BCGNYZ), Dir., J. M. Wright Tech. Sch., 1 Schuyler Ave., Stamford, Conn.
Lang, Edward H. (J'39) (ACMN), Prin., Syracuse Tech. & Indus. High Sch., 214 Park Ave., Syracuse 4, N.Y.
Lang, Frederic Allison (J'40) (LCTYMD), Devel. Engr., E. I. du Pont de Nemours & Co., Newburgh, N.Y.
Lang, Hans Joachim (J'41) (BHKLOD), Staff Engr., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2; *for mail,* R.D. 2, Kennett Square, Pa.
Lang, Henry W. (J'26) (AC), Installer, N.Y. Tel. Co., 101 Willoughby St., Brooklyn; *for mail,* 336—88th St., Brooklyn 9, N.Y.
Lang, Herbert Chester ('44) (MNLZHG), Ch. Draftsman, Mason-Neilan Regulator Co., 1190 Adams St., Boston 24; *for mail,* 45 Alvin Ave., East Milton 86, Mass.
Lang, Herman Alexander (J'43) (BAKHZN), Mechanics Instr., Cornell Univ., Ithaca, N.Y.
Lang, John ('32) (CJP), Gen. Mgr., Lang Co., 267 W. 1st St. S., Salt Lake City; *for mail,* 125 S. 13th St., Salt Lake City 2, Utah.
Lang, John Biggart ('16; '26) (CDJKMN), Dir., John Lang & Sons Ltd., Johnstone near Glasgow; *for mail,* Rossall, Kilmacolm, Renfrewshire, Scotland.
Lang, Leonard F. ('38) (FSLC), Mgr., Midwestern Territory, West. Precipitation Corp., 1 N. La Salle St., Chicago 2; *for mail,* 4435 N. Winchester Ave., Chicago 40, Ill.
Lang, Otto Carl ('31) (ECJKS), Head, Maint. Engrg. Dept., Carbide & Carbon Chemicals Corp., McCorkle Ave., South Charleston; *for mail,* 1 Grandview Dr., Charleston 2, W.Va.
Lang, Rudolph Charles ('29) (LMDCAB), Devel. Engr., Natl. Biscuit Co., 449 W. 14th St., New York; Instr., Pratt Inst., 195 Ryerson Ave., Brooklyn, N.Y.
Lang, William Albert (J'45) (BSZ), Sr. Design Engr., Brown Instrument Co., Wayne & Roberts Ave., Philadelphia 44; *for mail,* 4507 N. 19th St., Philadelphia 40, Pa.
Lang, William C. (J'39) (OMCL), Plant Engr., Synthetic Rubber Div., Firestone Tire & Rubber Co., Wilbeth Rd.; *for mail,* 1691 Marigold Ave., Akron 1, Ohio.
Langdon, Howard Harland ('25; '35) (BNSTW), Head, Research & Devel., Spec. Mch. Div., Consltd. Mch. Tool Corp.; *for mail,* 153 Windemere Rd., Rochester 10, N.Y.
Langdon, Robert Kenneth (J'42) (CZJD), Sales Engr., Walter Gerlinger Inc., 610 W. Michigan Ave., Milwaukee; *for mail,* 232 Wabash Ave., Waukesha, Wis.
Langdon, Robert Sylvester, Jr. (J'42) (NMJ), Lt. (j.g.), U.S.N., Navy 3002, Box 34, F.P.O. San Francisco, Calif.; *for mail,* 8816 Langston Rd., Seattle 88, Wash.
Langdon, Roy Alger (J'41) (SLJ), Tool Designer, Gisholt Mch. Co., 1245 E. Washington Ave., Madison; *for mail,* 723 W. Johnson, Madison 5, Wis.
Langdon, Warren Ross (J'43) (NKJM), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 706 Altamont Ave., Schenectady 3, N.Y.
Lange, Clifford Brandt (J'45), Lt., Supply Corps, U.S.N.R., 239 Filmore, Twin Falls, Idaho.
Lange, Forrest Fay ('24; '25; '29) (OCMLSN), Loan Engr., Reconstr. Finance Corp., 55 Tremont St., Boston 9; *for mail,* 27 Owatonna St., Auburndale, Mass.
Lange, Henry B. ('30) (CDMO), Partner, Henry B. Lange & Co., Cons. Engrs., 453 S. Spring St., Los Angeles 13, Calif.
Lange, John Henry (J'37) (BDENKH), Lt. (j.g.), Naval Shipyard, San Francisco, Calif.; *for mail,* 3817 Glenwood Rd., Brooklyn, N.Y.
Lange, Joseph O. ('38) (CBHJMA), Engr. of Pats., Crane Co., 836 S. Michigan Ave., Chicago 5; *for mail,* 5835 N. Maplewood Ave., Chicago 45, Ill.
Lange, Max E. ('25) (BMCZ), Engr., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3, Ohio.
Lange, Paul H. ('14), Engr., 50 Beacon St., Bridgeport, Conn.
Langenderfer, Raymond Charles (J'40) (MDCLAJ), R.F.D. 3, Swanton, Ohio.
Langer, Bernard F. ('43) (RBNA), Mgr., Mech. Sec., Trans. Engrg. Dept., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Langfitt, Joseph K. ('20) (DJM), Works Engr., Link-Belt Co., 220 S. Belmont Ave., Indianapolis; *for mail,* Box 106, R.R. 17, Indianapolis 44, Ind.
Langham, Henry T. ('43; '44) (CMGZYS), Engrg. Consultant, F. W. Dodge Crop., 119 W. 40th St., New York 18, N.Y.
Langhorst, Richard Theodore (J'38) (HELSCD), Application Engr., Allis-Chalmers Mfg. Co., Forest Ave., Norwood 12; *for mail,* 4166 Allendale Dr., Cincinnati 9, Ohio.
Langill, Ross Earl ('42) (CHW), Secy., Treas., Prescott Co., Broadway & Wabash, Menominee, Mich.
Langille, Herbert B. ('15) (ENSW), Assoc. Prof. Mech. Engrg., Emeritus, Univ. of Calif.; *for mail,* 2418 Dana St., Berkeley 4, Calif.
Langley, James Max (J'41) (CES), Indus. Lub., Petroleum Advisers, 60 Wall Tower, 70 Pine St., New York 5, N.Y.
Langmuir, Irving (Non-Member), *Holley Medalist* '34; Assoc. Dir., Research Lab., Gen. Elec. Co.; *for mail,* 1176 Stratford Rd., Schenectady, N.Y.
Langner, Frederick William ('27; 33; '35) (EFS), c/o Socony Vacuum Oil Co., Inc., 1407 Tribune Bldg., 154 Nassau St., New York 7, N.Y.
Langsdorf, Alexander S. ('20) (BCD), Dean, Schs. of Engrg. & Arch., Washington Univ., St. Louis 5; *for mail,* 5187 Cabanne, St. Louis 13, Mo.
Langsner, Adolph ('22) (COMLWZ), Chief, Research Div., Langsner Engrg. Associates, 2640 W. Coyle Ave., Chicago 45, Ill.
Langston, Ernest Richard (J'42) (ONHJ), Mech. Engr., Libby, McNeill & Libby, Box 1140, Honolulu; *for mail,* 1523A Kapiolani St., Honolulu 25, T.H.
Langstroth, Clifford Barnes ('12; '19) (JEBAR), Boulder Acres, R.D. 1, Danbury, Conn.
Langtry, Wilmer Dyer ('43) (FSO), Pres., Commerical Testing & Engrg. Co., 307 N. Michigan Ave., Chicago 1, Ill.
Langvand, I. L. (A'09), Asst. Supt., Babcock & Wilcox Co.; *for mail,* 526 Lloyd St., Barberton, Ohio.
Langworthy, Ross Andrew ('19) (DEFKOS), Mech. Engr., Sanderson & Porter, 52 William St., New York 5; *for mail,* 68-10—108th St., Forest Hills, N.Y.
Langworthy, William Probasco ('23) (JMC), Managing Partner, Magnetic Metals Co., 21st St. & Hayes Ave., Camden, N.J.
Lanham, Charles Warren, Jr. (J'44) (AES), 1343 Weldon Ave., Baltimore 11, Md.
Lanham, Paul Trueman (J'41) (ZHM), Lanham, Md.
Lanier, Haskell DuBose ('36), Exec. Engr., Gen. Sugar Estates, 801 Metro. Bldg., Havana, Cuba.
Lanigan, Thos. M., Jr. ('22; '27; '35), 5509 Nebraska Ave., N.W., Washington, D.C.
Lanning, John Edward ('29) (JKD), Ch. Mech. Engr., Morenci Branch, Phelps Dodge Corp.; *for mail,* Box 346, Morenci, Ariz.
Lanno, Edward Carl (J'37) (CERSMH), Prod. Engr., Diesel Eng. Div., Gen. Motors Corp., 13400 Outer Drive W., Detroit 23; *for mail,* 15985 Woodland Dr., Dearborn, Mich.
Lansing, Charles Bridgen ('27) (CDMO), Indus. Consultant, Cleveland Trust Co., Cleveland 1; *for mail,* R.F.D. 1, Chagrin Falls, Ohio.
Lansing, James Haswell ('44) (JNMDCB), Cons. Engr., Malleable Founders' Soc., Union Commerce Bldg., Cleveland 14, Ohio.
Lantau, Martin, Jr. (J'43) (AENSW), Maint., Compressor Dept., Tenn. Gas & Transmission Co., Commerce Bldg., Houston; *for mail,* Box 2511, Houston 6, Tex.
Lanzisera, Joseph Crescent ('26; '31; '35) (BCDMNO), Cons. Engr., 185 Erasmus St., Brooklyn, N.Y.
Lapidas, Manuel (J'43) (AJM), Research Assoc., Harvard College, Cambridge; *for mail,* 10 Charles Ave., Revere, Mass.
Lapides, Robert Edwin (J'39) (MCDJ), V.P., Charge Prod., United Smelting & Aluminum Co., Inc., 187 Commerce St., New Haven 10; *for mail,* 1200 Ridge Rd., Hamden 14, Conn.
Lapidus, Solomon (J'46) (BHM), Mech. Engr., Firm of Harry W. Faeber, 70 E. 45th St., New York; *for mail,* 1040 Bushwick Ave., Brooklyn 21, N.Y.
Lapinski, Lucian Stanley (J'42) (ABJ), 492 Atlantic St., Bridgeport, Conn.
La Porta, Henry E. (J'43), Assoc., Associated Indus. Consultants, Broad St. Bank Bldg., Trenton, N.J.; *for mail,* 35 Hamilton Pl., New York 31, N.Y.
Lapp, Robert Jacob (J'45), Engr., Mead Paper Corp.; *for mail,* R.F.D. 3, Chillicothe, Ohio.
Lappin, Joseph ('21; '22; '35) (SKF), V.P., Ch. Engr., Interboro Co., 109 E. 29th St., New York, N.Y.
Lardis, Nicholas J. (J'41), Capt., 185th Engr. Combat Bn., Ft. Jackson, S.C.
Lardner, Henry Ackley ('01; F'38), V.P., Treas., J. G. White Engrg. Corp., 80 Broad St., New York 4, N.Y.

Larew, J. Lee ('21; '35) (SENMJF), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14, N.Y.; *for mail,* 835 Main St., South Amboy, N.J.
Larinoff, Michael William (J'41) (EABNJZ), Engr., Design Sec., Diesel Power Plant, Ebasco Services Inc., Rm. 1100, 2 Rector St., New York, N.Y.
Larkin, Albert Chesebro ('95; '05) (DJNZ), Ch. Draftsman, Continental Can Co. of Can., Ltd., St. Laurent St.; *for mail,* 5465 Bourret Ave., Montreal 29, Que., Can.
Larkin, David ('21; F'45) (CJN), *Vice-President; '44-'45, Director-at-Large, '45-'46;* V.P., Gen. Mgr., Broderick & Bascom Rope Co., 4203 N. Union Blvd., St. Louis 15, Mo.
Larkin, Fred Viall ('15; F'45) (CLMNZ), *Manager, '43;* Dir., Mech. & Indus. Engrg., Lehigh Univ., Bethlehem, Pa.
Larkin, William Harrison ('28; '38) (SFCK), Engr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., 420 Lexington Ave., New York; *for mail,* 50 Ausable Ave., Staten Island 1, N.Y.
Larner, Hugh Randolph (J'38) (MCDN), Indus. Engr., Time Study Dept., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3; *for mail,* 1423 Parkview Dr., Lyndhurst, Ohio.
Larocco, Michael Jerome (J'45), Lt. (j.g.), Ship Engrg. Officer, 4961 N. Kilpatrick Ave., Chicago 30, Ill.
LaRoche, Gilbert L. (J'41), 496 S. Elm Ave., Kankakee, Ill.
Laroche, Joseph John (J'42) (CMDA), Ensign, U.S.N.R., Asst. Contract Termination Officer, c/o Inspr. of Naval Matl., U.S.N., Buffalo Dist., 220 Delaware Ave., Buffalo 2, N.Y.
LaRocque, Arthur E. (J'37) (NCM), Supvr., Inspc. & Test, Gen. Elec. Co., 2231 E. State St., Trenton; *for mail,* 12 Nordacs St., Trenton 8, N.J.
Larsen, A. M. ('29; '35), Asst. Design Engr., Solvay Process Co., Solvay; *for mail,* 216 Parsons Dr., Syracuse, N.Y.
Larsen, Arild F. (J'36) (CMDOLJ), Ch. Indus. Engr., David Bradley Mfg. Works, Kankakee; *for mail,* 103 N. Maple, Momence, Ill.
Larsen, Fred Roger (J'45) (BJKNAM), Engr., Watts Regulator Co., 10 Embankment St., Lawrence; *for mail,* 36 Fairmount Ave., Somerville, Mass.
Larsen, G. Sinding ('29) (CFK), V.P., Charge Engrg., Pittsburgh Piping & Equip. Co., 10—43rd St., Pittsburgh 1, Pa.
Larsh, Wayne Eugene (J'46), 513 School St., Houma, La.
Larson, Bert Edwin ('40) (RFJ), Ch. Engr., Loco. Firebox Co., 310 S. Michigan Ave., Chicago 4, Ill.
Larson, Charles Roy (J'44) (ERNM), Draftsman, Electro-Motive Div., Gen. Motors Corp., McCook; *for mail,* 102 S. Stone Ave., La Grange, Ill.
Larson, Clifford M. ('25) (EBART), Ch. Cons. Engr., Sinclair Refining Co., 630—5th Ave., New York 20, N.Y.
Larson, Emil Lambert ('29) (RBASJN), Ch. Draftsman, Chicago, Burlington & Quincy R.R. Co., 547 W. Jackson Blvd., Chicago 6; *for mail,* 5468 S. Woodlawn Ave., Chicago 15, Ill.
Larson, G. L. ('15) (EKS), Chmn., Mech. Engrg. Dept., Univ. of Wis., Madison, Wis.
Larson, Godfrey William ('29; '36) (BKJFSZ), Ch. Engr., Sugar Creek Refinery, Stand. Oil Co. (Ind.), Sugar Creek; *for mail,* 6112 S. Benton St., Kansas City 4, Mo.
Larson, Jarl Evert ('41) (COS), Examiner, Office of Defense Plants, Reconstr. Finance Corp., 811 Vermont Ave., Washington 25, D.C.; *for mail,* 1618 Mt. Eagle Pl., Parkfairfax, Alexandria Va.
Larson, John Albert (J'36) (EJM), Sales Engr., E. C. Cooley Co., 625 Market St., San Francisco; *for mail,* 724 Mountain Blvd., Oakland 11, Calif.
Larson, Reinhold Fridtjof ('30; '39) (SEKBFA), Assoc. Prof., Mech. Engrg. Dept., Univ. of Ill. Urbana, Ill.
Larson, Robert LaFollette (J'43) (NABECM), Mathematical Technician, Cleveland Diesel Div., Gen. Motors Corp., 2160 W. 106th St., Cleveland 11, Ohio.
Larson, Theodore Earl (J'44) (C), Box 56, Route 1, Lynden, Wash.
Larsson, Thure L. F. ('05) (FHW), Retired; Center Sandwich, N.H.
LaRue, Albert Morlan (J'45) (AYNBC), Devel. Engr., De-Icer Devel. & Research Dept., B. F. Goodrich Co., 500 S. Main St., Akron; *for mail,* 1807 Chestnut Blvd., Cuyahoga Falls, Ohio.
Lasciak, Charles ('32; '38) (SKCBMN), Mar. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 829 Nostrand Ave., Brooklyn 16, N.Y.
Laserna, Alfonso (J'44), Carrera 7'-13-56, Bogotá, Colombia, S.A.
Laserson, Gregory L. (J'45) (AES), Asst., Dept. of Mech. Engrg., Columbia Univ.; *for mail,* 295 Central Park W., New York, N.Y.
Lash, Joseph Charles (J'42), Elec. Engr., Tocco Div., Ohio Crankshaft Co., 3800 Harvard Ave., Cleveland; *for mail,* 1517 Howard Ave., Cleveland 13, Ohio.
Lashbrook, Thomas S. (J'35) (CMJLWY), Indus. Engr., Owens-Ill. Glass Co., 601—36th Ave., Oakland 1; *for mail,* 5800 Balboa Dr., Oakland 11, Calif.
Lask, Fred'k ('13; '35), Adv. Mgr., A.S.M.E., 29 W. 39th St., New York 18; *for mail,* Hillandale Rd., Port Chester, N.Y.
Lasker, Frank A. (J'40), c/o Lasker Boiler & Engrg. Co., 3201 S. Wolcott St., Chicago 8, Ill.
Lasker, Harold H. C. ('36) (LDSKJE), Owner, Lasker Engrg. Co., 17 E. 42nd St., New York 17, N.Y.
Laskin, Irving (J'45) (BM), Design & Devel. Engr., Ajon Research Corp., 356 W. 36th St., New York; *for mail,* 3578 DeKalb Ave., Bronx 67, N.Y.
Laskowitz, Isidor Bernard ('30) (AHSBND), Mech. Engr., Office of Pres. of Borough of Brooklyn, Municipal Bldg., Brooklyn 2; *for mail,* 284 Eastern Pkwy., Brooklyn 25, N.Y.
Lasley, James Bernard (J'45) (FOS), Asst. Foreman, N.C. Shipbldg. Co., Wilmington, N.C.
Lasley, Robert Anderson ('46) (AEM), Works Mgr., Aircraft & Diesel Equip. Corp., 4401 N. Ravenswood Ave., Chicago 40; *for mail,* 1425 Hutchinson St., Chicago 13, Ill.
Lassagne, Theodore Hawley (A'43) (AC), Partner, Naylor & Lassagne, 2607 Russ Bldg., San Francisco 4, Calif.
Lassalle, Leo Joseph ('21) (BCO), Dean, College of Engrg., La. State Univ. & Agric. & Mech. College, University Sta., Baton Rouge 3, La.
Lassen, Ernest Julius (J'37) (CDJMNS), Mgr., Welding Sales, N.Y. Dist., Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Lassman, Benjamin ('20; '35) (NMHJB), Owner, Firm of Benjamin Lassman, Hyd. Engr., Oliver Bldg., Pittsburgh 22, Pa.
Laster, Oliver (J'43) (HEFB), Asst. Engr., Stand. Oil Co. of Calif., Rm. 470, 265 Bush St., San Francisco 4, Calif.
Lastner, Charles Andrew (J'44) CHMNOW), Project Engr., Aeromotive Engrg. Co., Maple & 9th Sts.; *for mail,* 1614 Oak St., Niles, Mich.
Latham, Allen, Jr. ('45) (LKHJNB), Charge of Engrg. Devel., Arthur D. Little, Inc., 30 Memorial Dr., Cambridge 42, Mass.
Latham, George R., III ('41) (SEFJDK), Power Engr., Millville Mfg. Co., Columbia Ave.,; *for mail,* 701 E. Pine St., Millville, N.J.
Lathrop, Earl Sigworth (J'44) (CDJM), Indus. Engr., Aluminum Co. of Am., New Kensington; *for mail,* 322—8th, Oakmont, Allegheny Co., Pa.
Lathrop, Robert Park (J'42), 4613 Windsor Lane, Bethesda 14, Md.
Latimer, Charles (J'44), 223 Boone Hall, Oak Ridge, Tenn.
Latimer, Raymond Arthur (J'42) (CMD), Indus. Engr., RCA Victor Div., Radio Corp. of Am.; *for mail,* 626 N. Duke, Lancaster, Pa.
LaTour, Francis Gerard (J'44) (CMKYDZ), Sr. Methods Engr., Irvington Varnish & Insulator Co., 6 Argyle Terrace, Irvington 11; *for mail,* 67 Washburn St., Jersey City 6, N.J.
Latshaw, Elmer ('43) (BRY), Tech. Engr., ACF-Brill Motors Co., 62nd St. & Woodland Ave., Philadelphia 42, Pa.
Lattan, Jean Edward ('45) (CBMHFS), Div. Supt., Taylor Forge & Pipe Works, Box 485, Chicago 90; *for mail,* 806 N. Kenilworth Ave., Oak Park, Ill.
Lattin, Clark P., Jr. (J'38) (CSLKE), Sales Engr., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Lattin, Judson ('91), Retired; Box 567, R.R. 2, Fontana, Calif.
Latulippe, Lucien J. (J'41) (DHJ), Mech. Engr., Hall Mchy. Co., Lansdowne St., Sherbrooke, Que., Can.
Lau, Francis Len-Sin (J'43), Pvt., Army Air Forces, 4000 Base Unit, Sqd. X, Wright Field, Dayton, Ohio.
Laubach, Howard Evans ('37) (CMGJ), Ch. Engr., Smith & Mills Co., 2889 Spring Grove Ave., Cincinnati; *for mail,* 2542 River Rd., Cincinnati 4, Ohio.
Laubenstein, Albert R. ('18; '35) (DJKN), Gen. Mgr., Laubenstein Mfg. Co., 422 S. 3rd St., Ashland, Pa.
Lauck, Lawrence Joseph (J'43) (AEBHJ), Layout & Design Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 36 Richard Rd., East Hartford, Conn.
Laudig, John Benjamin ('27; '37) (MLYDC), Mech. Engr., Packaging & Shipping Serv. Inc., 1st Natl. Bank Bldg., Scranton 1; *for mail,* 318 N. Bromley Ave., Scranton 4, Pa.
Lauer, Albert William, Jr. (J'44) (AEM), 67 Glorieux St., Irvington 11, N.J.
Lauer, William N. ('43) (CHO), Sales Engr., Dorr Co., 570 Lexington Ave., New York 22, N.Y.
Lauffer, William George ('23; '28; '35) (SKF), Mech. Plant Betterment Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Laughlin, George C. ('40; '44) (SJCDLF), Asst. Supt., Struc. & Mech. Div., Constr. Dept., Commonwealth Edison Co., 72 W. Adams St., Chicago 90, Ill.
Laughnan, Thomas G. (J'40) (KLZEAO), Sr. Tech. Engr., Barber-Colman Co., 115 Loomis St., Rockford, Ill.
Laughton, Watson B. (A'36) (CGWY), Pres., Treas., McLoughlin Bros., Inc., 45 Warwick St., Springfield 1, Mass.
Laulhere, Bernard Marcelin, Jr. (J'39) (COE), Mgr., Pacific Pipeline Constr. Co., 2268 E. Firestone, Los Angeles 2, Calif.
Lauman, Herman E. ('29; '35) (DFH), Pres., C. W. Lauman & Co., Inc. & Duro Pump Co., Inc., 50 Church St., New York 7, N.Y.
Laun, Fred C. (J'40) (SO), Draftsman, Edo Aircraft Corp., College Point; *for mail,* 145-38—19th Ave., Whitestone, L.I., N.Y.
Laurie, Albert ('21) (ESFKAL), Owner, Laurie & Lamb, 512 Trans. Bldg., 132 St. James St. W., Montreal, Que., Can.
Laursen, Alfred (J'38), c/o Erection Dept., Babcock & Wilcox Co., Barberton, Ohio.
Laursen, Milton P. (J'41) (CDJ), 725 Columbia Blvd., Minneapolis, Minn.
Laussucq, Harry Peter Lawrence ('21) (YHRJ CK), Mgr., Hyd. Dept., Birdsboro Steel Fdy. & Mch. Co., Birdsboro; *for mail,* 6 S. Los Roble Court, Reading, Pa.
Lauterbach, George Edward (J'29) (SRFCBK), Supvr., Steam Power Plants, New York Cent. System, 466 Lexington Ave., New York 17, N.Y.; *for mail,* Chestnut St., Haworth, N.J.
Laux, John P. ('34), 503—3rd St., Towanda, Pa.
La Vaute, Lester Arthur (J'38), 118 Lincoln Park Dr., Syracuse 3, N.Y.
Laverty, Harold Hazen (J'42) (JNS), Lt., U.S.N.R.; *for mail,* R.F.D. 1, Brownville, Me.
Lavold, Gerald P. (J'41) (ANBCHW), Project Engr., Foote Bros. Gear & Mch. Corp., 5225 S. Western Blvd., Chicago 9; *for mail,* 3305 W. Diversey Ave., Chicago 47, Ill.
Law, Clifford J. (J'41), 139 N. Arlington Ave., East Orange, N.J.
Lawatsch, Frank R. ('14; '35) (JNFEZ), Mech. Engr., U.S. Engrg. Office, P.O. & Custom House Bldg., St. Paul 1, Minn.
Lawitz, Leslie L. (J'34) (BJM), 5444 Ingleside Ave., Chicago 15, Ill.
Lawler, Frank Patrick ('42) (BDHMO), Cons. Engr., 1642 W. 105th Pl., Chicago 43, Ill.
Lawler, Joseph Vincent (J'41) (JMNO), Field Cost Engr., E. I. du Pont de Nemours & Co.; *for mail,* Box 1353, Richland, Wash.
Lawlor, Edward Reeve (J'42) (ESOCF), Port Engr., Port of Los Angeles & Long Beach, Los Angeles Tanker Opers., Inc., 365 W. 7th St.; *for mail,* 1525-C W. 17th Walk, San Pedro, Calif.
Lawlor, James J. (J'40) (HKS), 26 Central Ave., Wortendyke, N.J.
Lawrance, Arthur T. ('38; '39) (CJL), Asst. to Div. Mgr., Alco Products Div., Am. Loco Co., 30 Church St., New York, N.Y.
Lawrance, Charles Lanier ('28) (A), Dir., Eng. Research, Lawrance Aero. Corp., Stiles St., Linden, N.J.; *for mail,* 151 E. 63rd St., New York 21, N.Y.
Lawrence, Albert D. (J'35) (EFM), 10507 Lake Ave., Cleveland 2, Ohio.
Lawrence, Andrew ('45) (BCJMNZ), Ch. Engr., Hyscore Arms Corp., 25 Lafayette St., Brooklyn 1, N.Y.
Lawrence, Ernest O. (Non-Member), *Holley Medallist, '42;* Prof. of Physics, Dir., Radiation Lab., Univ. of Calif., Berkeley, Calif.
Lawrence, George Edward (J'44), Engr., Instrument Div., Bausch & Lomb Optical Co., St. Paul Blvd.; *for mail,* 62 Park Lane, Rochester, N.Y.
Lawrence, Hewson (J'44), Jr. Engr., Drafting, Aerojet Engrg. Corp., Pasadena; *for mail,* 2729 Chevy Chase Dr., Pasadena 2, Calif.
Lawrence, Howard Brewster ('18; '26; F'38) (EHS), Engr., Elec. Advisers, Inc., 70 Pine St., New York 5, N.Y.
Lawrence, Howard F. ('08; '22) (FHKS), Research Engr., Am. Engrg. Co., Aramingo Ave. & Cumberland St., Philadelphia 25, Pa.
Lawrence, James VanHorn ('36) (HJNZO), Ch. Draftsman, Ford Instrument Co., 31-10 Thompson Ave., Long Island City, N.Y.
Lawrence, John Andrew, Jr. (J'41) (ARS), Aero. & Mar. Engrg. Div., Gen. Elec. Co., 1 River Rd.; *for mail,* 17 Front St., Schenectady 5, N.Y.
Lawrence, John H. ('11; '20; F'36) (SFEHKB), *Manager, '24-'27, Vice-President, '27-'29;* Cons. Engr., Metro. Device Corp., 1250 Atlantic Ave., Brooklyn 16; *for mail,* 25 Parkview Ave., Bronxville 8, N.Y.
Lawrence, Leland E. ('29; '35) (AJNWYZ), Devel. Engr., Allen Bradley Co., Milwaukee; *for mail,* 6603 Revere Ave., Wauwatosa 13, Wis.
Lawrence, Moses P. ('24), Supt., Generating Plant, Appalachian Elec. Power Co., Glen Lyn, Va.
Lawrence, Richard Allen (J'40) (SFEKOD), Lt. (j.g.), Civil Engr. Corps, U.S.N.R., Motor Trans. Officer, 33rd Naval Constr. Bn. (Special) Navy 3149, F.P.O., San Francisco, Calif.; *for mail,* 101 Inwood Ave., Montclair, N.J.
Lawrence, Stillson Freeman ('25; '33; '35) (E), Struc. Dept., Lummus Co., 420 Lexington Ave.; *for mail,* 304 E. 42nd St., New York, N.Y.
Lawrence, Walter Welling ('32; '43) (SFYJ LM), Asst. Plant Engr., Vulcan Detinning Co., Sewaren, N.J.
Lawrence, William (J'41), Ensign, Ord. Supt's. Office, Naval Shipyard, Pearl Harbor, T.H.
Lawrence, Wm. H. ('25), Ch. Oper. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Lawrence, William Howard (J'42) (ESKCHF), Gen. Elec. Co., 560 Lexington Ave., New York; *for mail,* 50 Mayfair Ave., West Hempstead, N.Y.
Lawson, Edward Charles, Jr. (J'39) (ESFC), Lt., U.S.M.C.R.; 232—77th St., Brooklyn 9, N.Y.
Lawson, Jas. T. ('15; '19), Asst. to Gen. Supt. of Generation, Elec. Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark, N.J.
Lawson, Stanton Clowes Davison ('45) (NB JMD), Assoc. Prof. Mech. Engrg., Antioch College, Yellow Springs, Ohio.

Lawson, William O. ('22; '35) (YMCN), Asst. to Works Engr., E. I. du Pont de Nemours & Co., Arlington, N.J.
Layfield, Elwood Baker ('40) (LSDYKH), Ch. Engr., Bechtel Bros. McCone Co., 220 Montgomery St., San Francisco 4, Calif.
Layton, J. Wm. (J'40) (AHNR), Ch. Engr., Propellair, Inc., 1345 Lagonda Ave., Springfield; *for mail,* 200 E. Hebble Ave., Osborn, Ohio.
Lazan, Benjamin J. (J'42) (JONBAY), Assoc. Prof., Head of Testing & Metal. Labs., College of Applied Sci., Syracuse Univ., Syracuse, N.Y.
Lazarus, Richard A. (J'41) (CLY), 822 Marietta Ave., Lancaster, Pa.
Lazier, Morley John Campbell ('45) (BCO), Cons. Engr., 170 Bay St., Toronto 1, Ont., Can.
Lazzaro, Vito (J'42) (C), Sec. Head, Statistical & Methods Dept., Merrill, Lynch, Pierce, Fenner & Beane, 70 Pine St., New York 5; *for mail,* 539—85th St., Brooklyn 9, N.Y.
Lea, Robert Brooke ('46), V.P., Sperry Gyroscope Co., Great Neck; *for mail,* 350 Lakeville Rd., Lake Success, N.Y.
Leach, Charles Harold ('20; '25; '35), Pres., C. H. Leach Co., Box 510; *for mail,* 100 E. Palisade Ave., Englewood, N.J.
Leach, Clarence R., Jr. (J'39) (CKS), Engr., Mar. Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 330 Prospect Ave., Mamaroneck, N.Y.
Leach, Jonathan Lawrence ('43) (CDZMAK), Asst. Gen. Mgr., Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Leach, Vernon Gard ('23) (FL), Ch. Combustion Engr., Peabody Coal Co., 231 S. La Salle St., Chicago 4, Ill.
Leach, Walter Lewis ('42) (OHSLEB), Prin. Engr., Havens & Emerson, 233 Broadway, New York 7, N.Y.
Leasenfeld, Charles J. (J'36) (JESZAC), Field Engr., Metallizing Engrg. Co., 38-14—30th St., Long Island City; *for mail,* 170-19 Henley Rd., Jamaica 3, N.Y.
Leatherman, Lloyd Roscoe ('42) (SLKFC), Partner, Leatherman & Mertz, 2140 Book Bldg., Detroit 26, Mich.
Leavitt, George Edward, Jr. ('27) (LSC), Plant Engr., So. Cotton Oil Co., 160 E. 22nd St., Bayonne; *for mail,* 607 Springfield Ave., Cranford, N.J.
Leba, John Jay (J'40) (CJM), Secy., Treas., Twin City Tool Co., 1511-15 Lowry Ave. N.; *for mail,* 2424—5th St., N.E., Minneapolis, Minn.
Leba, Steven John (J'42), Pres., Twin City Tool Co., 2520 Marshall St., N.E.; *for mail,* 2551 Stinson Blvd., Minneapolis, Minn.
LeBailly, Andrew R. ('40) (EOS), Mech. Engr., Sargent & Lundy, 140 S. Dearborn St., Chicago 3; *for mail,* 934 Sunnyside Ave., Chicago 40, Ill.
LeBeau, George Bernard (J'44) (CDJM), Supervisory Engr., Thompson & Lichtner Co., Park Sq. Bldg., Boston 16, Mass.; *for mail,* 170 Hughes Ave., Pawtucket, R.I.
Le Bel, Louis Phillip (A'42) (SFKM), V.P. & Ch. Engr., Combustion Engrs., Inc., 3 American Row, Hartford 3; *for mail,* 382 Farmington Ave., Hartford 5, Conn.
Le Blanc, Joseph Hugh, Jr. (J'45), Foreman, Welding & Mfg. Co., 609 Calliope St., New Orleans, La.
Le Blanc, Julius Andrew, Jr. (J'44) (BL), Engrg. Draftsman, Ethyl Corp., Baton Rouge; *for mail,* 763 Lakeland Dr., Baton Rouge 8, La.
LeBlanc, Roy Augustin (J'42) (BHS), Mech. Engr., Tech. Supvr., Tenn. Eastman Corp., Box 590, Knoxville; *for mail,* 108 Kingfisher Lane, Oak Ridge, Tenn.
LeBlond, Chester S. (J'39) (CLM), Mfg. Planning Engr., Teletype Corp., 1400 W. Wrightwood Ave.; *for mail,* 2123 N. Meade Ave., Chicago, Ill.
LeBlond, Richard E. ('23; '36), Spec. Apprentice, R. K. LeBlond Mch. Tool Co., Madison & Edwards Rds., Cincinnati 8; *for mail,* Box 28, R.R. 1, Madisonville, Cincinnati, Ohio.
LeBlond, Richard Knight ('00) Chmn., Bd. of Dirs., R. K. LeBlond Mch. Tool Co., Madison & Edwards Rds., Cincinnati 8, Ohio.
Lebo, William Howard (J'40) (CMP), Lt. (j.g.), U.S.N.R., Carrier Aircraft Serv. Unit (F) 48, F.P.O., San Francisco, Calif.
Lechler, Bruno C. ('18; '21), Mgr., Pat. Dept., Am. Mch. & Metals, Inc., East Moline, Ill.
Lechthaler, Clinton Knox (J'40) (CDM), 6 Tudor Pl., West Hempstead, L.I., N.Y.
LeClerc, Arthur Burge (J'36) (SHKF), Dist. Engr., with T. C. Heyward, 1408 Independence Bldg., Charlotte 2, N.C.
Le Compte, Frank M. (J'37) (ASEFHN), Project Engr., Curtiss-Wright Corp., Caldwell; *for mail,* 22 John St., Chatham, N.J.
Le Coney, Horace Morgan, Jr. (J'37) (FEC), Lub. Engr., Stand. Oil Co. of N.J., Sewalls Point, Norfolk 12, Va.
LeConte, Joseph Nisbet ('06) (BHS), Prof. Mech. Engrg., Emeritus, Univ. of Calif., Berkeley 4; *for mail,* Box 1312, Carmel, Calif.
Ledbetter, Thomas Benson (J'42), U.S. Navy.; 120 Fayetteville Rd., Rockingham, N.C.
Ledden, Edward B. (J'43), Ch. Engr., Kresge Dept. Store, 715 Broad St., Newark 2; *for mail,* 251 S. Ridgewood Rd., South Orange, N.J.
Ledeen, Howard Lee (J'45) (HCDMLS), Engr., Engrg. Products Co., 1600 S. San Pedro St., Los Angeles 15; *for mail,* 840 S. Orange Grove Ave., Pasadena 2, Calif.
Ledeen, Hyman ('19; '22; '28) (CD), Pres., Engrg. Products Co., 1600 S. San Pedro St., Los Angeles 15, Calif.
Lederer, Jerome F. ('24; '30; '35) (ACB), Ch. Engr., Aero Ins. Underwriters, 111 John St., New York 7, N.Y.
Ledgerwood, Byron Keener (J'44), 407 Westminster Rd., Brooklyn, N.Y.
Ledgett, Lowell Ashton ('33; '43) (LKHBZ), Head, Process Design Dept., Cent. Engrg. Div., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City 2; *for mail,* 237 Emmett Pl., Ridgewood, N.J.
Ledin, Charles Carleton (J'41), Technician, Hdq., 15th Air Force, A.P.O. 520, c/o Postmaster, New York, N.Y.; *for mail,* 16 Bedford Pl., Stamford, Conn.
Ledingham, William Ernest (J'46) (ACM), 250 Sherwood Dr., Ottawa, Ont., Can.
Ledsham, William Henry ('35) (CFS), Ch. Engr., Leeds & Lippincott Co., Chalfonte-Haddon Hall, Atlantic City, N.J.
LeDuc, Richard James ('29; '34; '36) (WMCJ), Pres., Treas., U.S. Cutting Chain Mfg. Co., 32 S. Main St., Randolph, Mass.
Lee, Adolph Orlando (J'39) (KZFSEC), Prof. Mech. Engrg., Univ. of Minn., Minneapolis 14, Minn.
Lee, E. H. (J'40) (BNJKAH), Sr. Exper. Officer, Armament Research Dept., Ministry of Supply, London; *for mail,* 81 Sandhurst Rd., Orpington, Kent, England.
Lee, Edward R., Jr. ('34; '42) (SFK), Mech. Engr., J. G. White Engrg. Corp., 80 Broad St., New York 4, N.Y.; *for mail,* 70 Ridge Rd., Glen Rock, N.J.
Lee, Everett Samuel ('37) (ZLBCSA), Engr., Gen. Engrg. & Cons. Lab., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1350 Wendell Ave., Schenectady 8, N.Y.
Lee, Fairman B. ('21; '30; '35) (ZSJML), Owner, Fairman B. Lee Co., 309—4th & Cherry Bldg., Seattle 4, Wash.
Lee, George Gilmore (J'43) (KMCDO), 97 Commerce St., West Point, Miss.
Lee, George Hamor ('38; '43) (BZA), Assoc. Prof. Mech. Engrg., Post Graduate Sch., U.S. Naval Academy, Annapolis, Md.
Lee, George Watson (J'32) (EMSO), Mchy. Inspr., Oil Tankers, Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4; *for mail,* 4 Holly Pl., Staten Island 6, N.Y.
Lee, Gilmer T. ('16), 635 Greenwood Rd., Roanoke, Va.
Lee, Harold Clemons ('44) (HKSEJY), Sales Engr., Allan J. Harris Co., 400 Balter Bldg., New Orleans 12, La.
Lee, Henry Wai Foon (J'43) (ABE), Test Engr., Wright Aero. Corp., Paterson; *for mail,* 128 Ward St., Paterson 4, N.J.
Lee, John A. ('37; '45) (LBK), Mech. Engr., Magnolia Petroleum Co., Box 900, Dallas 1; *for mail,* 4154½ Prescott, Dallas 4, Tex.
Lee, Luen On (J'41) (EMB), Engr., Vimalert Co., Ltd., 90 Forrest, Jersey City 4, N.J.; *for mail,* 24 Mott St., New York 13, N.Y.
Lee, Nixon ('43) (CJM), Pres., Lee Finish Corp., 250 W. 54th St.; New York; *for mail,* 14 Ivy Close, Forest Hills, N.Y.
Lee, Ralph E. (J'41) (CD), Apt. 2, 2500 Hillegass Ave., Berkeley, Calif.
Lee, Richard William (J'44) (LMONCJ), Mech. Engr., Consultant, H. K. Ferguson Co., Ltd., Box 219; *for mail,* 676 King St. E., Hamilton, Ont., Can.
Lee, Robert Ernest (J'39) (SFZEL), D. H. Skeen & Co., 1 N. La Salle St., Chicago; *for mail,* 10845 S. California Ave., Chicago 43, Ill.
Lee, Robert James ('34; '43) (OKEFRS), Lt. (j.g.), U.S.N.R., 2356—77th Ave., Philadelphia 38, Pa.
Lee, Robert T. (J'36) (MLS), Consultant, Box 164, Lynchburg, Va.
Lee, Rowland C. (J'42) (AKHCBF), Design Engr., Avion, Inc., 4545 E. 50th St., Los Angeles; *for mail,* 1745 Silverlake Blvd., Los Angeles 26, Calif.
Lee, Smith ('31) (OBH), 704 S. Spring St., Los Angeles 14, Calif.
Lee, Wallace Rodgers ('13) (CR), V.P., Gregg Car Co., Ltd., 19 Rector St., New York 6, N.Y.; *for mail,* 108 Orchard Way, Rosemont, Pa.
Lee, William Knapp (J'42), 209 Brookwood Dr., Greenville, S.C.
Leebov, Nathan ('44) (OCMBJN), Cons. Engr., 6364 Phillips Ave., Pittsburgh 17, Pa.
Leeds, Jacob Howard (A'22) (SHA), Owner, Metro. Supply Co., 353 E. 2nd St., Los Angeles 12, Calif.
Leeds, Maurice A. (J'45), 2697 Euclid Heights Blvd., Cleveland Heights 6, Ohio.
Leemans, Robert Bell (J'40) (SKF), Mech. Engr., Power Plant Design, Ebasco Services Inc., 2 Rector St., New York 6, N.Y.; *for mail,* 602—17th St., Union City, N.J.
Leen, Morris ('42) (BJL), Designer, Rogers Diesel & Aircraft Corp., 1120 Leggett Ave., New York; *for mail,* 245 Echo Pl., New York 57, N.Y.
Leeper, Ralph W. ('08; '21), Mgr., Canadian Gen. Elec. Export Ltd., 212 King St. W., Toronto 2, Ont., Can.
Leerburger, Franklin J. ('38) (OESFCH), Cons. Engr., Rm. 4312, 1 Wall St., New York 5, N.Y.
Leeson, Meldon Gerald (J'35) (MCKN), Devel. Engr., York Corp.; *for mail,* 589 Madison Ave., York, Pa.
Leet, Leslie N. ('43), Prod. Control Mgr., Fed. Tel. & Radio Corp., 320 Orange St., Newark; *for mail,* 7 Evergreen Court, Mountainside, Westfield, N.J.
Leete, William Tower ('34; '44) (CMZ), Engrg. Asst. to Supt., Veeder-Root, Inc., Sargent St., Hartford; *for mail,* 37 Hilltop Dr., West Hartford 7, Conn.
Lefever, Paul M. ('39) (HSOCZ), Plant Supt., Susquehanna Elec. Co., Conowingo, Md.
Lefren, Edward K. (J'41) (MLY), Design Engr., Hercules Powder Co., Port Ewen; *for mail,* 75 Florence St., Kingston, N.Y.
LeGates, Frank E. ('43) (SJOCL), Ch. Inspr., Hartford Steam Boiler Inspc. & Ins. Co., 605 Mercantile Trust Bldg., Baltimore 2, Md.
Leggett, Ilow Waddell ('30; '38) (EHST), Sales Engr., Worthington Pump & Mchy. Corp., 131 Brevard Court, Charlotte 1, N.C.
Leggett, James Thomas, Jr. (J'43) (JMZ), Asst. Prof., Univ. of Fla., Gainesville, Fla.
Leggett, John Richard (J'39) (SFHKED), Mech. Engr., Barrett Div., Allied Chem. & Dye Corp., 86 Elizabeth Ave., Elizabeth B; *for mail,* 240 High St., Nutley, N.J.
Leggo, Wm. F. (A'17), Sales Engr., M. W. Kellogg Co., 225 Broadway, New York, N.Y.
Legier, Edward Wilton ('38) (SLWY), East. Div. Mgr., Am. Blower Corp., 50 W. 40th St., New York 18, N.Y.
Le Gros, Emile Albert (J'41) (CJL), 1048 Chicago Ave., Oak Park, Ill.
Lehman, Arthur Wismer (J'44), Sgt., 330th Army Serv. Forces Band, Ft. Jackson, S.C.; *for mail,* 220 E. Ashland St., Doylestown, Pa.
Lehman, Benjamin Joseph (J'46), c/o Zamkin, 320 W. 86th St., New York 24, N.Y.
Lehman, Edgar Adelbert ('43) (BCLNWY), V.P., Gen. Mgr., Lehwood Corp., 14th & Blaine; *for mail,* Box 672, Elkhart, Ind.
Lehman, Paul William ('39) (YOLMTC), Cons. Engr., Giffels & Vallet, Inc., 1000 Marquette Bldg., Detroit 26; *for mail,* 1386 Grayton Rd., Grosse Pointe 30, Mich.
Lehman, Werner ('16) (DEJ), Cons. Engr., Bucyrus-Erie Co.; *for mail,* 728 Michigan Ave., South Milwaukee, Wis.
Lehmberg, William Howard ('44) (TCMYAL), Engrg. Dir., Am. Felt Co., Glenville, Conn.
Lehn, Henry C. ('15; '35) (E), Cons. Engr., Buffalo Works, Worthington Pump & Mch. Corp., Buffalo; *for mail,* 30 Morris Ave., Buffalo 14, N.Y.
Lehner, John Bernhard (J'30) (EFKSZ), Pvt., 1st Class, Corps of Engrs., U.S.A., Engr. Reserve Trng. Corps, Ft. Belvoir, Va.; *for mail,* 69—4th Ave., Mineola, N.Y.
Lehoczky, Paul N. ('28; '34; '35) (CMDJO), Prof. & Chmn., Indus. Engrg., Ohio State Univ., Columbus 10, Ohio.
Lehr, Charles E. ('16) (CDJ), 1842 Kenmore Ave., Bethlehem, Pa.
Leigh, Edgar D. (J'39) (CDM), Sec. Engr., Charge Facilities, RCA Victor Div., Radio Corp. of Am., Camden; *for mail,* 1322 Keswick Ave., Haddon Heights, N.J.
Leigh, Frank Donald (J'42), Lt., U.S.N.R., 3006 La Branch St., Houston 4, Tex.
Leigh, Richard Shipman ('45) (KHBJOS), Project Engr., Devel. Dept., Chase Brass & Copper Co., Waterbury 91; *for mail,* R.F.D. 1, Woodbury, Conn.
Leighton, A. J. ('32), Babcock & Wilcox Co., Rm. 1502, 140 S. Dearborn St., Chicago, Ill.
Leilich, Frank T. ('19; '28) (CFMORS), Col., Corps of Engrs., U.S.A.; *for mail,* 2611 Chelsea Terrace, Baltimore 16, Md.
Leilich, George Morse (J'37) (RC), Train Master, Oper. Dept., Lehigh Valley R.R. Co., Wilkes-Barre; *for mail,* 34 Butler St., Forty Fort, Pa.
Leinart, Bryon Howard ('42) (CDKLS), Partner; Leinart Engrg. Co., 412 E. 5th Ave., Knoxville 5, Tenn.
Leinheiser, Richard Paul (J'39) (EASN), 40 Ave. F, Claymont, Del.
Leisenheimer, Robert Henry (J'42) (CFK), Ch. Engr., Furnace Div., Forest City Fdys. Co., 2500 W. 27th St., Cleveland; *for mail,* 15814 Parkgrove Ave., Cleveland 10, Ohio.
Leister, Fayette ('39) (BMADH), Mgr., Detroit Office, Fafnir Bearing Co., 502 New Center Bldg., Detroit 2; *for mail,* 18950 Santa Rosa, Detroit 21, Mich.
Leitch, Kelvin Douglas ('39) (HKS), Treas., Ch. Engr., Arthur S. Leitch Co. Ltd., 1123 Bay St., Toronto, Ont., Can.
Lelgeman, William ('44) (CFKMSY), Ch. Engr., Orr & Sembower, Inc., Morgantown Rd.; *for mail,* R.D. 1, Reading, Pa.
Lem, Frank Yee ('27; '34; '35), Jr. Asst. Engr., Shanghai Power Co., 95 Nanking Rd.; *for mail,* House 2, Lane 18, Jessfield Rd., Shanghai, China.
LeMay, Jack Evans (J'41) (LJYMZD), 12588 Lake Ave., Lakewood, Ohio.
Lembcke, Robert Kilbourne (J'40) (ACKMEO), Detail Engr., Exper. Dept., Allison Div., Gen. Motors Corp.; *for mail,* Box 53, R.R. 18, Indianapolis, Ind.

Lembeck, Otto A. ('13), 60 Hillcrest Ave., Summit, N.J.
Lemen, Ralph Marvin (J'39) (KOE), 21 Claremont Ave., New York 27, N.Y.
Lemery, Jack W. R. (J'35) (CLR), Partner, Lambert & Lemery, 1297—64th Ave., Oakland 3, Calif.
Lemezis, Sylvester, Jr. (J'43) (SJR), Lt. (j.g.), U.S.N.R., Heavy Duty Garage, Navy 3923, F.P.O., San Francisco, Calif.
Lemke, Albert Emerson (J'44), Mech. Engr., Designer, Draftsman, Link-Belt Co., 631 S. Anderson St., Los Angeles; *for mail,* Box 669, Whittier, Calif.
Lemley, Benjamin Willis ('15; '35) (CM), Owner, Lemley & Co., 840 Falls Ave., Cuyahoga Falls, Ohio.
Lemmer, Horace ('41) (LYZ), Owner, Mgr., Process Equip. Co., Keefer Bldg., Montreal; *for mail,* 1095 Graham Blvd., Mount Royal, Que., Can.
Lemmon, James R. ('37) (SKLCE), Dist. Mgr., Elliott Co., 801 Morris Bldg., Philadelphia 2, Pa.
Lemoine, Sidney J., Jr. (J'37) (CDJ), Box 1391, Alexandria, La.
Lempera, Edward Jos. (J'37) (BFS), Mgr., Maint. Depts., Sears, Roebuck & Co., 925 S. Homan Ave., Chicago; *for mail,* 6752 Riverside Dr., Berwyn, Ill.
Lenau, Henri, Biering ('22) (EHRT), Specialist Engr., E. B. Badger & Sons Co., 500—5th Ave., New York 18; *for mail,* 8 Herkimer Rd., Scarsdale, N.Y.
Lender, Albert (J'39) (AEBNJM), Matls. & Process Engr., Metallurgist, Supv. of Matls. Lab., U.S. Naval Air Sta., Quonset Point; *for mail,* 297 California Ave., Providence 5, R.I.
Lenderoth, A. W. ('23), Crosby Steam Gage & Valve Co., 165 Broadway, New York 6, N.Y.
Lenfest, Bertram A. ('04) (JNFCM), 130 Sterling Pl., Brooklyn 27, N.Y.
Lenfest, Harold Churchill ('40) (ESRO), Pres., Continental Engrg. Co., 30 Church St., New York 6, N.Y.
Leng, Richard B. (J'35) (CLMEDO), Supt., Methods & Time Study, RCA Victor Div., Radio Corp. of Am.; *for mail,* 1607 Oregon Pike, Lancaster, Pa.
Lengel, Albert (J'37) (HSR), Engr., Worthington Pump & Mchy. Corp., 4747 Broadway, Denver 16, Colo.
Lenihan, Thomas Joseph, Jr. (J'44) (LMCY), Lt. (j.g.), U.S.N.R., Asst. Shop Supt., Shop Supt.'s Office, Puget Sound Naval Shipyard, Bremerton, Wash.
Lennart, Robert P. (J'41) (HSZAEC), Pump Sta. Supvr., War Emergency Pipelines, Inc., Box 1638, Cincinnati 1; *for mail,* Apt. 7, 2749 Linshaw Court, Cincinnati 8, Ohio.
Lennie, Arthur Mossman (J'44) (MJD), Metal. Engr., Dow Chem. Co.; *for mail,* 405 E. Nelson, Midland, Mich.
Lenno, Emery Joseph ('31) (KOAEW), Partner, Office of David D. Kieff, Chamber of Commerce Bldg.; *for mail,* 178 Ward St., Watertown, N.Y.
Lennox, Julius Jacob, Jr. (J'43) (AFM), 1838—9th St., Port Arthur, Tex.
Lenone, José M. ('16) (SEFKL), Designing Engr., Wilson & Co., Inc., 4100 S. Ashland Ave., Chicago 9, Ill.
Lentz, Lawrence Walter (J'36) (JKOASF), Cons. Engr., R.R. 1, Algonac, Mich.
Lenzen, Ralph Jay Jr. (J'42) (MNY), Mch. Designer, Mall Tool Co., 7740 S. Chicago Ave., Chicago 19; *for mail,* 1422 E. 66th Pl., Chicago 37, Ill.
Leon, Alexander Saenz (J'42) (KLS), Mgr., Mch. Div., M. Castellvi, 150 Broadway, New York 7, N.Y.
Leonard, Albert Prince ('37) (LYBDFK), Mgr., N.Y. Office, Farrel-Birmingham Co., Chrysler Bldg., New York 17; *for mail,* 35-56—79th St., Jackson Heights, N.Y.
Leonard, Arthur Geo. ('90), Pres., Union Stock Yard & Transit Co., Admin. Bldg., Union Stock Yards, Chicago, Ill.
Leonard, Arthur George, Jr. (J'28) (GMSC), Secy., Lehon Co., Wilmington, Ill.
Leonard, Carroll M. ('37; '43) (SKFEAL), Assoc. Prof. Mech. Engrg., Okla. Agric. & Mech. College, Stillwater, Okla.
Leonard, Charles F. ('13) (MCBO), Cons. Engr., 220 Broadway, New York 7, N.Y.
Leonard, Dorris Glenn (J'43) (ABSCJ), Engr., Curtiss-Wright Corp., Lambert Field, St. Louis; *for mail,* 1936 W. Main, Jefferson City, Mo.
Leonard, Frederic Buchanan (J'46), Mar. Engr., Tech. Div., Maritime Comm., Rm. 7019, Washington, D.C.; *for mail,* 6716 N. Central Ave., Chevy Chase 15, Md.
Leonard, Ibbotson ('06; '21) (EFKS), Cons. Engr., E. Leonard & Sons Ltd., 881 York St., London, Ont., Can.
Leonard, John Sawyer (J'38) (EBNKH), Design Engr., Elec. Boat Co., Eastern Point Rd., Groton, Conn.
Leonard, Leonid Vladimir ('44) (LEKHSZ), Sr. Mech. Engr., Shell Oil Co., Inc., 1008 W. 6th St., Los Angeles 14, Calif.
Leonard, Malcolm W. ('23) (SFKZ), Asst. Mech. Div. Ch., Test Lab., Pub. Serv. Elec. & Gas Co., 200 Boyden Ave., Maplewood; *for mail,* 19 Molter Ave., Springfield, N.J.
Leonard, N. Nelson, Jr. (J'36) (ACBN), Design Engr., Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 441 N. Barrington Ave., Los Angeles 24, Calif.
Leonard, William Lee, Jr. (J'42), Apt. 1E, 3412 Colonial Ave., Norfolk 8, Va.
Leong, Steven Way (J'41) (ABFHKN), Aircraft Designer, Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 1700 Manning Ave., West Los Angeles 24, Calif.
Leonhard, Frederick John ('29; '39) (SCERFM), Plant Design Engr., Cleveland Elec. Illum. Co., 75 Public Sq., Cleveland 1; *for mail,* 2894 Huntington Rd., Shaker Heights 20, Ohio.
LePage, Clifford Blondel ('13) (MDNCLB), Asst. Secy., A.S.M.E., 29 W. 39th St., New York 18; *for mail,* 145 Greenway S., Forest Hills, L.I., N.Y.
Le Paige, Leon Ghislain ('42) (BKS), Amitas Ltd., 50 Broadway, New York; *for mail,* 60 E. 67th St., New York 21, N.Y.
Lepley, Raymond Daniel ('41) (OSEKLJ), Engr. in Charge, Tech. Div., Atlantic Refining Co., 260 S. Broad St., Philadelphia 1; *for mail,* 436 E. Durham St., Philadelphia 19, Pa.
Lerner, Milton Ernest (A'45) (YL), Editor, *Rubber Age,* 250 W. 57th St., New York 19, N.Y.
Leroy, Walter William (J'34) (CSL), Pres., Mag-Bestos Co., Inc., Box 581, Elizabethton, Tenn.
Lerse, Hans Julius (J'43) (EKSYZT), Test & Devel. Engr., Universal Camera Corp., 28 W. 23rd St., New York; *for mail,* Hotel Lombardy, 111 E. 56th St., New York 22, N.Y.
Lesch, Raymond Thomas (J'39) (SKFZBH), Mech. Engr., Helmick, Edeskuty & Lutz, 412 Essex Bldg., Minneapolis 2; *for mail,* 2712—44th Ave. N., Minneapolis 12, Minn.
Leslie, Bernard S. ('15; '25), Pres., Firm of Bernard S. Leslie, 1119 Security Bldg., Miami, Fla.
Leslie, George Clark ('41) (SFEH), Utilities Engr., Richfield Oil Corp., Box 787, Wilmington; *for mail,* 3849 Line Ave., Long Beach 7, Calif.
Leslie, John S. ('35; '45) (CZLRMS), Pres., Leslie Co., Lyndhurst, N.J.
Leslie, Peter J. (J'46), 393 West End Ave., New York, N.Y.
Lessells, John Moyes ('23) (BEJNR), Assoc. Prof. Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Lesser, Murray Leon (J'42) (ABKHZR), Flight Test Analyst, Northrop Aircraft, Inc., Hawthorne; *for mail,* 1259 W. 65th St., Los Angeles 44, Calif.
Lesser, William Henry ('44) (CDFKNOS), Mech. & Elec. Engr., Pierce Mgmt., Scranton Elec. Bldg., Scranton; *for mail,* 1322 Myrtle St., Scranton 10, Pa.
Lesser, William Henry, Jr. (J'45) (CAD), 1322 Myrtle St., Scranton 10, Pa.
Lessmann, Gerhard Paul ('43) (BJSNKC), Mgr., Mech. Sec., D.C. Generator Engrg. Dept., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 214 Overdale Rd., Pittsburgh 21, Pa.
Lester, Bernard ('41) (BDJLMT), Asst. Indus. Sales Mgr., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.
Lester, Everett J. (J'41) (CEM), U.S. Hoffman Mchy. Corp., 219 Lamson St., Syracuse 6; *for mail,* 218 Plymouth Dr., Syracuse, N.Y.
Lester, Saul P. (J'44) (BJM), Assoc. Engr., Wyrstapes Co., 200 Hudson St., New York; *for mail,* 1200 College Ave., Bronx, N.Y.
LeTart, Harold Joseph (J'42) (ECMN), Lt., U.S.N.R., Bur. of Ships, Navy Dept., Washington 11, D.C.; *for mail,* 507 Elliott St., Grand Haven, Mich.
Letchfield, Francis Thomas ('22) (AJLM), Cons. Engr., Asst. V.P., Wells Fargo Bank & Union Trust Co., 14 Montgomery St., San Francisco 20, Calif.
LeTourneau, Robert G. ('41) (BJM), Pres., R. G. LeTourneau, Inc., Peoria, Ill.
Letterman, Frank L. (J'41) (FKJSL), Staff Engr., Aluminum Co. of Am., River Rd., Edgewater, N.J.
Lettice, Richard S. (J'37) (SFH), Results Engr., Ala. Ord. Works, E. I. du Pont de Nemours & Co., Box 270, Sylacauga; *for mail,* 1843 S. 16th Ave., Birmingham 5, Ala.
Leudemann, Albert Victor ('22; '35) (EFKL), Gen. Sales Mgr., Mears-Kane-Ofeldt, Inc., 11 Park Pl., New York 7, N.Y.
Leuner, Joseph H. (J'42), 3960 Monticello Ave., Bronx 66, N.Y.
Leunis, Robert Richard (J'28) (MHY), Mech. Engr., Product Designer, A. Schrader's Son, 470 Vanderbilt Ave., Brooklyn 17; *for mail,* 11 Howard Court, West New Brighton 10, S.I., N.Y.
Leussler, Arthur J. (J'24) (CFS), Rhodes Equip. Co., 4485 Olive St., St. Louis, Mo.
Leuthesser, Fred William, Jr. ('45), Secy. & Treas., Ch. Engr., Natl. Metal Products Corp., 21 N. Loomis St., Chicago, Ill.
Leutwiler, Oscar Adolph ('04; '11) (BNS), Prof. Mech. Engrg. Design, Emeritus, Univ. of Ill.; *for mail,* 710 Pennsylvania Ave., Urbana, Ill.
Leutwiler, Richard Walter, Jr. ('41; '45) (NG BMC), Asst. Prof. Mech. Engrg., Univ. of Calif., Berkeley, Calif.
LeVan, Ambrose Edwin ('45) (ANZ), Dir. of Engrg., U.S. Gauge Co., Div. of Am. Mch. & Metals, Inc., Clymer Ave., Sellersville, Pa.
Levat, Sanford George (J'44), 199 Main Ave., Passaic, N.J.
Levenhagen, Fred'k H. (J'39), 1st Lt., Army Air Forces; *for mail,* c/o Henry F. Levenhagen, 80 Ella St., Bloomfield, N.J.
Leverett, Frank M. ('22; '29) (SCFEKH), Power Engr., Tex. Co., Box 712, Port Arthur, Tex.
Leverett, Wilton Howard (J'28) (ZLFBSJ), Box 527, Houston 1, Tex.
Leverich, Jerome W. ('19) (CDJ), 1815 Dorchester Rd., Brooklyn, N.Y.
Leveridge, Robert Henry (J'45) (COKGM), Application Engr., Carrier Corp., 405 Lexington Ave., New York 17; *for mail,* 135 Park Ave., Tuckahoe 7, N.Y.
Levert, Lee, Jerome ('23; '30; '35) (AES), Div. Engr., Consltd. Edison Co. of N.Y., Inc., Rm. 1306F, 4 Irving Pl., New York 3, N.Y.
Leveskis, Victor G. (J'39) (DKNCEF), Field Engr., U.S. Elec. Motors, 564—4th St., San Francisco; *for mail,* Canyon P.O., Calif.
Levin, Bernard S. (J'39) (NBHMD), Mech. Engr., Naval Shipyard, Philadelphia 12; *for mail,* 4619 N. Warnock St., Philadelphia 40, Pa.
Levine, Aaron (J'39) (KBS), Design Engr. & Estimator, Condenser Serv. & Engrg. Co., 95 River St., Hoboken; *for mail,* 66 Beacon Ave., Jersey City 6, N.J.
Levine, Bernard (J'39) (ABHZ), Chief, Automatic Pilot Unit, Air Tech. Serv. Comd., Army Air Forces, Wright Field, Dayton; *for mail,* 719 Salem Ave., Dayton 6, Ohio.
Levine, Boris (J'34) (JBE), Engr., Elec., Office of Chief of Engrs., War Dept., Washington; *for mail,* 3100 Wisconsin Ave., N.W., Washington 16, D.C.
Levine, Harold (J'43) (KAMLYZ), Cpl., 258th Ord. Evacuation Co., Camp Hood, Tex.; *for mail,* 11—51st St., Weehawken, N.J.
Levine, Harold Elvin (J'43) (NBJ), Ensign, U.S.N.R., Planning & Estimating Supt., Naval Shipyard, Brooklyn; *for mail,* 834 Central Ave., Far Rockaway, N.Y.
Levine, Herbert P. (J'39) (EFS), Lt., U.S.N.R., Box 636, South Fallsburg, N.Y.
Levine, Lester (J'43) (HS), Supt., H. Sand & Co., Inc., 291 Broadway, New York 7; *for mail,* 451 Summit Ave., Cedarhurst, L.I., N.Y.
Levinger, David ('28) (CM), V.P., Works Mgr., Hawthorne Sta., West. Elec. Co., Inc., Chicago, Ill.
Levinson, Herman John ('29) (OFSHD), Mech. & Elec. Engr., Assoc. Engrs., 1601 Chestnut St., Philadelphia 3; *for mail,* 4929 N. Warnock St., Philadelphia 41, Pa.
Levinson, Saul (J'45) (NBMJG), Test Engr., Gen. Elec. Co., 920 Western Ave., Lynn, Mass.; *for mail,* 34 Fife St., Forest Hills, N.Y.
Levitt, Albert Phillip (J'45) (ABO), Research Asst., Gordon McKay Lab., Harvard Univ., Cambridge, Mass.
Levitt, Joseph (J'44) (KCAH), Mech. Engr., Schwarz Labs., Inc., 202 E. 44th St., New York 17; *for mail,* 592 Stone Ave., Brooklyn 12, N.Y.
Levitt, Semond (J'43) (CMN), Cons. Engr., 5646 Sansom St., Philadelphia 39, Pa.
Levy, Albert Pierre (J'42) (BKNOAC), Asst. Mech. Engr., with H. W. Beecher, Cons. Engr. & Associates, Securities Bldg., Seattle 5, Wash.
Levy, Bernard William (J'45), 1417 Union St., Schenectady, N.Y.
Levy, Byron L. (J'41) (ABCED), 2nd Lt., Army Air Forces, Maint. Engr., 551st Serv. Sqd., A.P.O. 857, Miami, Fla.
Levy, Harry (J'43) (KSEHO), Mar. Engr., N.Y. Naval Shipyard, Brooklyn 1, N.Y.; *for mail,* 25 Johnson Ave., Newark 8, N.J.
Levy, Lionel Faraday ('22; '35) (GBZ), Partner, Tech. Dir., Max Levy & Co., Wayne Ave. & Berkley St., Philadelphia 44, Pa.
Levy, Sydney ('32; '43) (SONKBH), Asst. Engr., Sanderson & Porter, 52 William St., New York; *for mail,* 220 Highland Blvd., Brooklyn 7, N.Y.
Levy, Victor Nathan (J'45), Research Engr., York Corp.; *for mail,* 279 N. Hartley, York, Pa.
Lew, John (J'42) (ANBMZ), Engr., Design Studies, Cornell Aero. Lab., 4455 Genesee St., Buffalo 5; *for mail,* 826 Ashland Ave., Buffalo 13, N.Y.
Lewellen Marcy Thomas ('39) (ERS), Navy V-12 Instr., Ariz. State Teachers College; *for mail,* 114 N. Park, Flagstaff, Ariz.
Lewin, Hellmut Louis (J'45) (NKMCS), Plant Engr., Union Wire Die Corp., 375 Fairfield Ave.; *for mail,* 13 Rachelle Ave., Stamford, Conn.
Lewin, Perez Otis ('45) (FSRTKZ), Asst. Supt. Power, Boston Elevated Ry., 536 Harrison Ave., Boston; *for mail,* 542 East St., Dedham, Mass.
Lewin, Saul (J'44) (ABC), c/o Carlin, 1290 E. 19th St., Brooklyn, N.Y.
Lewis, Albert David (J'39) (NJBZO), Devel. Engr., Kimble Glass Co.; *for mail,* 730 Park Ave., Vineland, N.J.
Lewis, Alexander Dodge (J'39) (EDNWZL), Asst. Prof., Visiting Lecturer, Mech. Engrg., Yale Univ., New Haven 11, Conn.
Lewis, Bruce E. (J'41) (FRS), 106 Maryland Ave., Riverside, Wilmington, N.C.
Lewis, Crompton Emerson (J'43) (CJDMFB), Asst. to Gen. Supt., Canadian Gen. Elec. Co., 940 Lansdowne Ave., Toronto; *for mail,* 21 Sutherland Dr., Toronto 12, Ont., Can.
Lewis, Dartrey ('38) (JYMA), Ch. Devel. Engr., John A. Roebling's Sons Co., 640 S. Broad St., Trenton 2; *for mail,* The Lexington, 455 W. State St., Trenton 8, N.J.

Lewis, Davis D. ('40) (TSLK), Plant Engr., Millville Mfg. Co., Columbia Ave.; *for mail,* 825 Columbia Ave., Millville, N.J.
Lewis, Douglas Randolph (A'43) (NBG), Ch. Appication Engr., Durametallic Corp., 24 Commerce St., Newark 2, N.J.
Lewis, Earl Ray, Jr. (J'41) (BM), Supvr, Time .Study, Pratt & Whitney Div., Niles-Bement-Pond Co., West Hartford; *for mail,* 22 Lancaster Rd., West Hartford 7, Conn.
Lewis, Ernest Eber (J'42) (HAS), Engr., Aero-Mar. Engrg. Dept., Gen. Elec. Co., 1 River Rd.; *for mail,* 231 Seward Pl., Schenectady 5, N.Y.
Lewis, Frank M. ('28), Prof. Mar. Engrg., Mass. Inst. of Tech., Cambridge; *for mail,* 200 Boston Post Rd., Weston 93, Mass.
Lewis, George W. (Non-Member), *Spirit of St. Louis Medallist, '44;* Dir. of Aero. Research, Natl. Advis. Com. for Aero., 1500 New Hampshire Ave., Du Pont Circle, Washington 25, D.C.
Lewis, Goodrich Quigg ('15; '35) (RBJY), Ch. Engr., W. H. Miner, Inc., 667—209 La Salle St., Chicago 4, Ill.
Lewis H. I. ('16; '25) (ACG), c/o York Safe & Lock Co., York, Pa.
Lewis, Harold Sheldon (J'40) (MACBH), Design Engr., Jaros, Baum & Bolles, 415 Lexington Ave.; *for mail,* 220 W. 72nd St., New York, N.Y.
Lewis, Henry Casper ('39) (JMBC), Shop. Supt., Charleston Elec. Supply Co., Kanawha Blvd., Charleston 29; *for mail,* 1719 McClung St., Charleston 1, W.Va.
Lewis, Herbert B. ('25; '28) (MJCKNY), Mgr., Mchy. Div., Luken Steel Co., Coatesville; *for mail,* 335 N. Franklin St., West Chester, Pa.
Lewis, Herbert Francis (J'40) (JMDCAO), Sales Engr., Lukens Steel Co., Coatesville; *for mail,* 121 E. Virginia Ave., West Chester, Pa.
Lewis, Irving R. (J'34) (LKZ), Ch. Mech. Engr., Merck & Co., Inc., Rahway; *for mail,* 32 Linden Ave., Metuchen, N.J.
Lewis James Patrick (J'41) (AMKYNZ), Mech. Engr., Research Project Engr., Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 17202 Fernshaw Ave., Cleveland 11, Ohio.
Lewis, James Tyler ('19) (CMR), Tech. Sec., U.S. War Shipping Admin., Rm. 1201, 39 Broadway, New York, N.Y.
Lewis, James Warren (J'42) (ABH), Lt., Hdq. & Serv. Co., 349th Engr. Combat Bn., Ft. Lewis, Wash.
Lewis, John L. (J'41) (FKS), c/o Babcock & Wilcox, 85 Liberty St., New York 6, N.Y.
Lewis, Joseph Walters, Jr. (J'41) (CMJB), Mgr., Indus. Relations, U.S. Steel Products Co., 5100 Santa Fe Ave., Los Angeles 11; *for mail,* 639 La Loma Rd., Pasadena 2, Calif.
Lewis, Kelvin Powell ('41) (HBSMCE), Chmn. of Dirs., Kelly & Lewis Ltd., Springvale; *for mail,* 227 Brighton Rd., Elwood S. 3, Victoria, Australia.
Lewis, Mark Campbell (J'37) (ACBKJH), Application Engr., Gen. Elec. Co., 5541 E. 14th St., Oakland 1, Calif.
Lewis, Oliver Kenneth, Jr. (J'44) (NJBM), Engr., Clinton Engr. Works, Tenn. Eastman Corp., Box 590, Knoxville; *for mail,* 114 W. Tennessee Ave., Oak Ridge, Tenn.
Lewis, Orval Leroy (J'39) (LOE), Project Engr., Engrg. & Constr. Div., Jones & Laughlin Supply Co., 108 N. Trenton, Tulsa 2, Okla.
Lewis, Philip L. (J'43), Field Engr., Corrigan, Osburne & Wells, 60 E. 42nd St., New York; *for mail,* 106 Elliott Pl., Bronx, N.Y.
Lewis, Ralph Elton ('35; '36) (BGNS), Asst. Prof. Mech. Engrg., College of Engrg., Duke Univ.; *for mail,* 1308 Markham Ave., Durham, N.C.
Lewis, Robert Bartlett ('44) (RJKNBM), Sales Engr., Timken Roller Bearing Co., Canton 6; *for mail,* 285 Cordelia Rd., North Canton, Ohio.
Lewis, Robert Compton ('45) (YBARZO), Ch. Engr., Vibration Div., MB Mfg. Co., Inc., 1060 State St., New Haven 11; *for mail,* 50 Alden Ave., New Haven 15, Conn.
Lewis, Thomas Daniel (J'42) (ABK), 8402 S. Green St., Chicago 20, Ill.
Lewis Warren D. ('16; '21; '35) (FES), Ch. Engr., Hotel New Yorker, 34th St. & 8th Ave., New York 1, N.Y.
Lewis, William Dein (J'38) (ABEKRS), Instr., Engrg. Exper. Sta., Pa. State College, State College, Pa.
Lewisohn, Richard, Jr. (J'43) (CDMAWY), Asst. Secy., Am. Lead Pencil Co., 500 Willow Ave., Hoboken, N.J.
Lewiton, Samuel ('45) (JLCDYT), Indus. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 23 Brinsley St., Dorchester 21, Mass.
Leyda, Harry Louis ('37) (JCELR), V.P., Dallas Tank Co., Inc., 201 W. Commerce St., Dallas 1; *for mail,* 1125 Canterbury Court, Dallas 8, Tex.
Libbey, W. Scott ('31; '35) (CYT), Treas., W. S. Libbey Co., Box 440, Lewiston, Me.
Libby, Clarence R. ('28; '40) (HABC), Engr., Nash Engrg. Co., Wilson Ave., South Norwalk, Conn.
Libby, Richard Henry ('13; '17) (CDLMNW), Sec. Mgr., Anstice Co., Inc., 80 Boylston St., Boston 16; *for mail,* Box 76, South Acton, Mass.
Libby, Samuel Hammonds ('97; '00; F'41), Retired; 23 Whittlesey Ave., East Orange, N.J.
Libman, Frank Herbert (J'44) (CWJM), Works Chemist, Congoleum-Nairn, Inc., Cedarhurst; *for mail,* 18 New Windsor Rd., Westminster, Md.
Li Calsi, Joseph (J'45), Engrg. Dept., Am. Dist. Tel. Co., 155—6th Ave., New York 13; *for mail,* 328 E. 78th St., New York 21, N.Y.
Licht, Carl Martin (J'45) (KLCM), 1725 Wilson Ave., Chicago, Ill.
Lichten, Earl Barton (J'44) (NOJMGC), Engr., Barnes & Reinecke, 230 E. Ohio, Chicago; *for mail,* 744 S. Central Park Ave., Chicago 24, Ill.
Lichtenberg, Frank Delbrook ('43) (CMLJB), Tannenbaum Ranch, Black Forest, Colorado Springs, Colo.
Lichtenstein, Immanuel (J'43) (CM), Jr. Prod. Engr., East. Aircraft Div., Gen. Motors Corp., Linden, N.J.
Lichtenstein, Joseph ('26) (HKS), Mech. Engr., Foster Wheeler Corp., 165 Broadway, New York 6; *for mail,* 155 Henry St., Brooklyn, N.Y.
Lichtenstein, Val., Jr. (J'41) (BNKMR), 119 Gifford Ave., Jersey City 4, N.J.
Lichtman, Alexander M. (J'43) (SEFHBO), Ch. Petty Officer, U.S. Maritime Serv.; *for mail,* 1860—49th St., Brooklyn, N.Y.
Lichty, Lester Clyde ('21; '30) (EBKAF), Prof. Mech. Engrg., Yale Univ., 400 Temple St., New Haven 11, Conn.
Lidbury, Frank Austin ('17), Pres., Oldbury Electro-Chem. Co., Buffalo Ave., Niagara Falls, N.Y.
Liddington, Stanley James ('27; '35) (BSCJM), Estimating Engr., Canadian Westinghouse Co. Ltd., Sanford Ave.; *for mail,* 91 Arkell Ave., Hamilton, Ont., Can.
Liddle, Stephen Wayne (J'46), Radio Corp. of Am., 946 Drexel Ave., Indianapolis, Ind.
Lieb, Lloyd Linwood, Jr. (J'42) (FHS), Allowance Engr., Naval Shipyard, Mare Island; *for mail,* Box 187, Route 1, Novato, Calif.
Liebenthal, John Henry, Jr. (J'46) (BGH), Draftsman, Wyman Gordon Co., 147th & Wood Sts., Harvey; *for mail,* 1446 W. 94th St., Chicago 20, Ill.
Lieberg, Eugene Olav (J'44), Engr., Design Dept., Chance Vought Aircraft Div., United Aircraft Corp.; *for mail,* 58 Anson St., Stratford, Conn.
Lieberman, Arthur Arnold ('44) (AMP), Engr., Mgr., Petroleum Marketing Co., 4410 District Blvd., Los Angeles; *for mail,* 1480 Darlene Dr., La Habra Heights, Calif.
Liebl, Louis, Jr. (J'41) (BCDJMN), Process Engr., Doehler-Jarvis Corp., Washington St.; *for mail,* 618 Spruce St., Pottstown, Pa.
Liebowitz, Benjamin ('20; '25) (BJ), Box A-91, R.F.D. 2, New Canaan, Conn.
Liedel, Elmer Louis ('44) (FKSZ), Tech. Engr., Delray Power House, Detroit Edison Co., 2000 2nd Ave., Detroit 26; *for mail,* 10167 Orangelawn, Detroit 4, Mich.
Liedstrand, Earl Harry (J'38) (CMJS), Asst. M.M., Am. Can Co., 4815 Santa Fe, Vernon, Calif.
Lien, Arthur T. (J'41) (ACJYWE), Matls. Engr., Emerson Elec. Mfg. Co., 8100 Florissant, St. Louis 21; *for mail,* 813 Blossom Lane, Rock Hill Village 19, Mo.
Lien, George Eric (J'40) (SKEMZ), Asst. Engr., Mech. Engrg. Dept., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.
Lierboe, Leland Lee (J'42) (AME), Project Engr., Northwest Airlines, St. Paul; *for mail,* Angus Hotel, St. Paul 2, Minn.
Lifvergren, Eric R. ('35; '35), Capt.; 3519 Dent Pl., N.W., Washington 7, D.C.
Light, James Albert (J'41) (AEH), 1436 Magee Ave., Philadelphia 11, Pa.
Light, Sol J. (J'43) (TLCYHS), Textile Mgr., Consltd. Products Co., Inc., 33-41 Bergen St., Paterson 3; *for mail,* 12-20 Fair Lawn Ave., Fair Lawn, N.J.
Lightowler, George R. ('16; '19; '35) (CM), 551—5th Ave., New York 17, N.Y.
Ligon, Cleon (J'42) (SMB), Mech. Engr., Power Plant Design, Southwest. Pub. Serv. Co., 430 N. Main St.; *for mail,* Box 631, Borger, Tex.
Lilienthal, Frederick Charles (A'45) (CLNW DG), Resident Engr., Merchant Mutual Casualty Co., 268 Main St., Buffalo, N.Y.; *for mail,* 239 Union Ave., Scotch Plains, N.J.
Lillie, Grant Wyman ('01; '21) (ABNR), Retired; 1340 Sunnyside Ave., Salt Lake City 5, Utah.
Lilly, Clyde Alanson, Jr. (J'45) (SKAJ), Test Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia; *for mail,* 4527 Walnut St., Philadelphia 39, Pa.
Lillyblad, Robert Harry (J'43) (MJCABK), Partner, Charge Mfg. & Engrg., Camfield Mfg. Co., 718 N. 7th St., Grand Haven, Mich.
Limbacher, Howard Raymond ('36; '41) (FKJ), Devel. Engr., Ingersoll Steel & Disc Div., Borg-Warner Corp., 521 Harrison Court, Kalamazoo; *for mail,* 525 Edgemoor Ave., Kalamazoo 36, Mich.
Limes, Barney Lee (J'42) (ER), Apt. A, 716 Interdrive, University City, Mo.
Limont, Alex W., Jr. ('29), Mech. Engr., 608 Sussex Rd., Wynnewood, Pa.
Linch, Edward Philip ('02) (BCJKMN), Pres., Gem Stopper Co., Inc., 2116-26 W. Nicholas St., Philadelphia 21, Pa.
Lincoln, Jas. F. ('27), Pres., Lincoln Elec. Co., Coit Rd. & Kirby Ave., Cleveland, Ohio.
Lincoln, John C. ('31) (EJM), Box 141, R.D. 1, Scottsdale, Ariz.
Lincoln, John Dickey ('44), V.P., Va. Lincoln Corp., Marion, Va.
Lincoln, Rollo Basil ('35), Pittsburgh Testing Lab., Pittsburgh, Pa.
Lincoln, Willis Cary ('44) (HRS), Dist. Mgr., Joshua Hendy Iron Works, 50 Church St., New York, N.Y.; *for mail,* 21 Chelsea Pl., East Orange, N.J.
Lind, John Edwin ('45) (KNBCL), Head Design Engr., Votator Div., Girdler Corp., Louisville 1; *for mail,* 123 Brown Ave., Louisville 7, Ky.
Lind, Julius A. (J'39) (FHS), Mech. Engr., Buffalo Niagara Elec. Corp., Rm. 300, Elec. Bldg., Buffalo; *for mail,* 34 Truesdale Rd., Kenmore, N.Y.
Lindahl, Eric Jean ('37; '42) (HEFS), Asst. Prof. Mech. Engrg., Ohio State Univ., Columbus 10, Ohio.
Lindahl, Robert Leslie ('46) (ABM), Ch. Engr., Div. of War Research, Columbia Univ., 64th Fl., 350—5th Ave., New York; *for mail,* 303 E. 37th St., New York 16, N.Y.
Lindau, Jules Washington, III ('44) (AGL), Prof. Mech. Engrg., Univ. of S.C., 906-08 Main St., Columbia 10; *for mail,* 625 Henderson St., Columbia 21, S.C.
Lindberg, Miss Anne Katherine (J'45) (JKN BRT), Engr., Metallurgical Research, Gen. Elec. Co., 1 River Rd.; *for mail,* 1007 Congress St., Schenectady, N.Y.
Lindberg, B. A. (J'38) (BDH), Personnel Dir., H. & B. Am. Mch. Co., Pawtucket; *for mail,* 163 Bartlett Ave., Edgewood 5, Providence, R.I.
Linde, Gustave Frederick ('28; '35) (MOLJGC), c/o Linde Tool & Engrg. Co., 9401 Schiller Blvd., Franklin Park, Ill.
Linde, Leonard J. ('37) (CNWBMJ), Ch. Engr., A. B. Chance Co., 210 N. Allen St., Centralia, Mo.
Lindell, W. Francis ('38) (SEFKAB), Asst. Prof. Mech. Engrg., Univ. of Del., Newark, Del.
Lindemann, Walter Carl ('13; '21) (BCJ), *Manager, '35-'38;* V.P., Secy., A. J. Lindemann & Hoverson Co., 601 Cleveland Ave., Milwaukee 7, Wis.
Lindemuth, F. L. ('14; '23) (BJM), Ch. Engr., W. B. Pollock Co., Andrews Ave.; *for mail,* 2267—5th Ave., Youngstown, Ohio.
Lindemuth, Ralph Lewis (J'45), Plant Engr., Design, L. B. Smith, Inc., Camp Hill; *for mail,* 315½ Market St., Lykens, Pa.
Linden, Gosta Roland (J'45) (BLONMS), Mech. Engr., Mch. Designer, J. M. Lehmann Co., Inc., 550 New York Ave., Lyndhurst; *for mail,* 13 Lloyd Court, Nutley 10, N.J.
Lindenauer, Martin Mathews (J'46), 75 W. Mosholu Pkwy., New York, N.Y.
Lindenmeyer, Ray Sherman (J'41) (JMC), Asst. Prof. Indus. Engrg., Northwestern Univ., Evanston, Ill.
Lindenmeyer, Robert E. (J'37), Lt. Comdr., U.S.N.R., 116 Scarsdale Rd., Crestwood, N.Y.
Linder, Clarence H. ('45) (C), Asst. Works Mgr., Gen. Elec. Co., Schenectady 5, N.Y.
Linder, Thomas ('24; '34; '35) (ABCEFS), Res. Engr., Accident Prevention, Travelers Ins. Co., Marx & Bensdorf Bldg., 2nd & Monroe, Memphis 3; *for mail,* Apt. 64, 1899 Poplar Ave., Memphis 4, Tenn.
Linderman, Harold C. ('43) (CFSKBO), Cent. Dist. Stoker Supt., Engrg. & Serv. Dept., Westinghouse Elec. Corp., 306—4th Ave., Box 1017, Pittsburgh 30, Pa.
Linderoth, L. Sigfred, Jr. ('45) (ONHBCJ), Mech. Engr., Research Div., United Shoe Mchy. Corp., Balch St., Beverly; *for mail,* Main St., Wenham, Mass.
Lindhagen, Manne T. ('33), Kungsgatan-32, Stockholm, Sweden.
Lindhard, Sven Norman (J'30) (KFZBD), Field Engr., F. L. Smidth & Co., 11 W. 42nd St., New York 18, N.Y.
Lindkvist, Gustav Adolf ('31) (DBKHLN), Engr., Link-Belt Co., 300 W. Pershing Rd., Chicago 9; *for mail,* 6720 N. Lakewood Ave., Chicago 26, Ill.
Lindley, Keith (J'38) (BH), Capt., Corps of Engrs., Hdq., Army Serv. Forces Trng. Center, Ft. Leonard Wood, Mo.; *for mail,* 1217 S. Elwood St., Tulsa, Okla.
Lindley, Roy W. ('22; '25; '35) (JLM), Prof. Engrg. Shop Practice, Purdue Univ., Lafayette; *for mail,* 114 De Hart St., West Lafayette, Ind.
Lindner, Berthold Alfred ('44) (ACMOJB), Liaison Engrg. Gp. Leader, Fairchild Aircraft Div., Fairchild Eng. & Airplane Corp., Hagerstown, Md.; *for mail,* 215 W. Beech St., Long Beach, L.I., N.Y.
Lindquist, Lawrence L. (J'42), Ch. Inspr., Permutit Co., 5401—1st Ave.; *for mail,* 7205 Shore Rd., Brooklyn, N.Y.
Lindquist, Willard E. (J'44) (LOZKSJ), Design Engr., Stand. Oil Co., Whiting, Ind.; *for mail,* 7622 S. Merrill Ave., Chicago 49, Ill.
Lindsay, Charles Hardie ('43) (AHFBM), Ch. Engr., Am.-LaFrance-Foamite Corp.; *for mail,* 315 Larchmont Rd., Elmira, N.Y.
Lindsay, David ('45) (CHL), Stands. Engr., Ohio Injector Co., Wadsworth, Ohio.
Lindsay, Donald Lorne (J'43) (CLMTD), Indus. Engr., G. Lorne Wiggs & Co., 1411 Crescent St., Montreal, Que., Can.

Lindsay, George Luther ('25; '38) (BENSJR), Ch. Engr., Skinner Eng. Co., 337 W. 12th St.; *for mail*, 4808 Reed Rd., Erie, Pa.
Lindsay, John Oliver ('40) (TLDSNO), Plant Engr., Pacific Mills, Lyman, S.C.
Lindseth, Elmer Lincoln ('26; '37), Pres., Cleveland Elec. Illum. Co., 75 Public Sq., Cleveland 1, Ohio.
Lindsey, Joseph Tyler (A'26) (BMW), Owner, Lindsey Products, 3—4th St.; *for mail*, 65 Queens St., Rochester 9, N.Y.
Lindsley, Allen Moore ('45) (CMDJ), Head Planning & Methods Engr., Wilson & Co., Inc., 4100 S. Ashland Ave., Chicago 9; *for mail*, 2147 W. 114th Pl., Chicago 43, Ill.
Lindstrom, Alvin Lawrence ('41) (BCFHKO), Cons. Mech. Engr., 1001 Mortgage Guarantee Bldg., Atlanta 3, Ga.
Lindstrom, Arthur William ('20; '35) (CDS LM), Plant Engr., Pabst Brewing Co., 917 W. Juneau Ave., Milwaukee 1; *for mail*, 901 W. Green Tree Rd., Milwaukee 9, Wis.
Lindstrom, Gustaf Thorsten ('36) (CJLNT), Mech. Engr., E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Lindstrom, Robert John (J'40) (ACKMB), Engr., Analyst, Boeing Aircraft Co., 7755 E. Marginal Way, Seattle 8; *for mail*, 911 E. Lynn, Seattle 2, Wash.
Linford, John William ('25) (CER), Sales Mgr., Diesel Eng. Div., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Lingner, George Leopold (J'36) (CM), Prod. Mgr., Jabez Burns & Sons, Inc., 11th Ave. & 43rd St., New York 18, N.Y.
Liniger, Richard B. (J'42) (DHLMOS), Mech. Engr., Liniger Co., Inc., Marion, Ind.
Lininger, Bryon A. ('43) (SFCMO), Power Consultant, E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Link, Charles Theodore, Jr. ('31; '42) (HNMJ CD), Asst. Ch. Engr., Hyd. Div., Racine Tool & Mch. Co., 1760 State St., Racine, Wis.; *for mail*, 3219 Balmoral Ave., Chicago 25, Ill.
Linker, John I. ('21; '25; '35) (BEFKSY), Engr., Gibbs & Cox, Inc., 21 West St., New York 6, N.Y.; *for mail*, 96 Oxford Pl., Glen Rock, N.J.
Linker, John Joseph (J'45) (BSAKHE), Project Engr., H. H. Buggie Co., 22nd & Madison Sts., Toledo 1; *for mail*, 462 Norwood Ave., Toledo 2, Ohio.
Linkletter, Richard L. (J'41) (ZBMSHY), Elec. Engr., Prod. Office, Puget Sound Naval Shipyard, Bremerton; *for mail*, Apt. 305, 1101—17th Ave., Seattle 22, Wash.
Linn, Frank Chester ('37) (FP), Engrg. Staff Member, Charge Mech. Design & Devel. Work, Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Linnell, Clinton W. ('23; '35) (SEK), Mgr., Trane Co., 31 St. James Ave., Boston 16; *for mail*, 14 Chesterford Rd., Winchester, Mass.
Linnenbruegge, Hans Herman Georg ('28) (ER), Supvg. Engr. Designer, Worthington Pump & Mchy. Corp., Roberts & Clinton Sts., Buffalo 6; *for mail*, 40 Oschawa Ave., Buffalo 10, N.Y.
Linscott, Leroy Nelson ('42) (TBO), Mech. Engr., Alex. Smith & Sons Carpet Co., Saw Mill River Rd., Yonkers; *for mail*, 607 Bellevue Ave., Yonkers 3, N.Y.
Linsenmeyer, Francis J. ('28; '30) (FSZOCK), Pub. Relations, Hoffman Combustion Engrg. Co., 710 Marquette Bldg., Detroit 26, Mich.
Linsenmeyer, John Z. (J'43), Application Engr., Industry Engrg. Dept. 10L, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.
Linsker, Eugene (J'45), 326 Oxford Ave., Dayton 7, Ohio.
Linsley, Leonard Noel ('25) (CGM), Capt., U.S.N., Asst. Naval Inspr. Gen., Navy Dept., Washington 25; *for mail*, Apt. 1, 2346—40th St., N.W., Washington 7, D.C.
Linten, Leonard (J'43) (EJAFRC), Mar. Engr., Code 643, Bur. of Ships, Navy Dept., Washington 25; *for mail*, 1518 Whittier St., N.W., Washington 12, D.C.
Linton, Wallace Ross (A'43) (TCL), Pres., Horace Linton & Bros., Inc., 3061 Ruth St., Philadelphia 34, Pa.
Lintz, Edgar Jules (J'42) (BHNZC), Mech. Engr., M. W. Kellogg Co., 225 Broadway, New York, N.Y.; *for mail*, 73 S. Walnut St., East Orange, N.J.
Linville, Thomas Merriam ('39) (NJCSML), Elec. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 1131 Waverly Pl., Schenectady 8, N.Y.
Lione, Leonard Victor (J'46), 6945 Groton St., Forest Hills, N.Y.
Lipetz, Alphonse I. ('19; F'38) (RBEO), *Melville Medallist*, '38; Ch. Cons. Engr., Am. Loco. Co., Jay St., Schenectady; *for mail*, 66 Willett St., Albany 6, N.Y.
Lipka, George S. (J'43) (MJA), c/o Metrol Co., 5538 Cass Ave., Detroit 2, Mich.
Lipke, Leopold H. ('19; '35) (ACM), Owner, Lipke Tool Works, 91 Lindsley Ave., Newark 6, N.J.
Lipkis, Robert Philip (J'44), 410 N. Rossmore Ave., Los Angeles 4, Calif.
Lipman, Robert Newman (J'41) (NMSHKE), Student Mech. Engr., RCA Victor Div., Radio Corp. of Am.; *for mail*, 1388 Kenwood Ave., Camden, N.J.
Lippenholz, Joseph S. (J'38) (LJDMYZ), Asst. Mech. Engr., Indus. Dept., Naval Shipyard, Philadelphia 12; *for mail*, 5423 Hadfield St., Philadelphia 43, Pa.
Lippincott, J. Gordon ('44), J. Gordon Lippincott & Co., Indus. Designers, 500—5th Ave., New York 18; *for mail*, 5 Reynal Crossing, Scarsdale, N.Y.
Lippitt, Henry F., II (J'43) (CTSM), 4495 Hortensia St., San Diego 3, Calif.
Lippmann, Edmund Eugene (J'34) (CDLMON), Gen. Mgr., Lippmann Engrg. Works, 4603 W. Mitchell St., Milwaukee 14; *for mail*, 2872 N. 74th St., Milwaukee 10, Wis.
Lipschultz, Harold L. (J'40) (BCDM), Ensign, U.S.N.R., U.S.S. *Amsterdam*, F.P.O., San Francisco, Calif.; *for mail*, 329 Lyons Ave., Newark, N.J.
Lipsett, Harry (J'41) (CH), 616—32nd Ave., Seattle 22, Wash.
Liptay, John M. ('29), Pres., Lab. Furniture Co., Inc., 37-18 Northern Blvd., Long Island City, N.Y.
Lipton, Martin H. (J'42), Tool Designer, George Scher Engrg. Co., 800 McCarter Highway; *for mail*, 41 Rose Terrace, Newark, N.J.
Lish, Kenneth Colman (J'41) (KJC), 6 Forest Lane, Lawrence, N.Y.
Liskow, Bernard H. ('14) (CBJM), Plant Engr., Chevrolet Grey Iron Fdy., 10th & Washington Sts.; *for mail*, 514 Atwater St., Saginaw, Mich.
Lister, Alfred ('13), Mech. Supt., Clark Thread Co., 260 Ogden St., Newark, N.J.
Litchfield, Norman ('23) (RSCO), V.P., Treas., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.
Litchfield, Paul Weeks (Non-Member), *Spirit of St. Louis Medallist*, '32; Chmn. of Bd., Goodyear Tire & Rubber Co. & Pres., Goodyear Aircraft Corp., Akron, Ohio.
Lite, Milton (J'44) (SOKCFE), Proposition Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Littell, Frederick Moore ('44), Detroit Rep., F. J. Littell Mch. Co., 6432 Cass Ave., Detroit 2; *for mail*, 19946 Canterbury Rd., Detroit 21, Mich.
Littig, John Stansfield ('45) (CZM), Lt., U.S.N., Facilities Officer, Office of Asst. to Indus. Mgr., Naval Shipyard, Terminal Island; *for mail*, 701 Los Altos Ave., Long Beach 4, Calif.
Little, Chas. H. (A'04), Mgr., Universal Drafting Mch. Co., 407 Blackstone Bldg., *for mail*, 2507 Stratford Rd., Cleveland, Ohio.
Little, David Harold ('44), Engr., Boston Edison Co., 39 Boylston St., Boston; *for mail*, 27 Rangley St., Dorchester, Mass.
Little, Edward W., II (J'41) (CDM), Pres., Treas., Crown Products Co., 3401 Newton, Indianapolis; *for mail*, 972 N. Layman, Indianapolis 1, Ind.
Little, Edwin Robert ('16; '21) (FOS), Pres., E. R. Little Co., Inc., 1807 Kales Bldg., Detroit 26, Mich.
Little, Frederick Alden (J'40) (NHBJCM), Design Engr., Pneumatic Drop Hammer Co., 200 Adams St., Braintree; *for mail*, 9 Granger St., Wollaston, Mass.
Little, Harry ('42), 25 Leland Ave., Pleasantville, N.Y.
Little, Richard Pangborn (J'42) (LRJ), Mem., Maint. Engrg. Dept., Carbide & Carbon Chemicals Corp., 437 MacCorkle Ave., South Charleston 3; *for mail*, 1536 Lee St., Charleston 1, W.Va.
Littlefield, Waldemar Vickery (J'42) (HWRS), Test Engr., Research & Devel. Dept., Worthington Pump & Mchy. Corp., Harrison; *for mail*, Apt. 407, 28 Gates Ave., Montclair, N.J.
Littler, Carl W. ('25) (EJS), Ch. Engr., Jones & Laughlin Steel Corp., 3rd Ave. & Ross St., Pittsburgh 30, Pa.
Littleton, Norman Gunnison ('44) (BCD), Factory Engr., Corning Glass Works; *for mail*, 142 E. 3rd St., Corning, N.Y.
Litty, Francis Edward (J'36) (MNBJAS), Estimating Engr., Stearns-Roger Mfg. Co., 1718-22 California St., Denver 2; *for mail*, 653 Olive St., Denver 7, Colo.
Litzinger, Lewis Peter, Jr. (J'42) (AEMRKB), Layout Draftsman, Aircraft Gas Turbine Div., De Laval Steam Turbine Co., Trenton 2, N.J.; *for mail*, 72-A Sayers Ave., Lansdowne, Pa.
Liu, Dennis Hsush-Hsi (J'42) (EFCM), 3136—19th St., N.W., Washington 10, D.C.
Livacich, John Thomas (J'44) (BDM), Tool Designer, Am. Can Co., 499 Alabama St., San Francisco 10; *for mail*, 595—3rd Ave., San Francisco 18, Calif.
Livengood, James Cooper (J'41) (AEKB), Research Asst., Mass. Inst. of Tech., Cambridge; *for mail*, Page Rd., South Lincoln, Mass.
Liversidge, Horace P. ('17; F'37), *Manager*, '21-'24; Pres., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Livingston, Beryl Edward ('46), Ch. Draftsman, Research & Devel. Dept., Elliott Co.; *for mail*, Woodlawn, Jeannette, Pa.
Livingston, Howard Stanley (J'41) (SKMF), Supvr., E. I. du Pont de Nemours & Co.; *for mail*, 503 Benham, Richland, Wash.
Livingston, Robert Teviot ('45), Assoc. Prof. Indus. Engrg., Columbia Univ., New York; Dir. of Education & Commercial Research, L.I. Lighting Co., Mineola, N.Y. (*Use former address for mail.*)
Livingston, William Hale (J'42) (CDLMZO), Branch Mgr., No. Calif. Mgmt. & Sales, E. O. Livingston Co., 2009 Shattuck Ave., Berkeley 4; *for mail*, 1349 Arch St., Berkeley 8, Calif.
Livingstone, Edward A. ('28) (JSP), Gen. Sales Mgr., Babcock & Wilcox Tube Co., Beaver Falls, Pa.
Livingstone, John Edward ('45) (HMNYJS), Mgr., Pres., John E. Livingstone Co., 2921 E. Strand Blvd., Detroit 2, Mich.
Livitski, William John (J'41), 916 Beaver Rd., Ambridge, Pa.
Llansó, Joaquin Jornet (J'29) (HELR), Casilla de Correo 3590, Buenos Aires, Argentina, S.A.
Llewellyn, Harold Z. (J'41) (JMS), Mech. Engr., Natl. Tube Co., 28th & Pearl Ave., Lorain; *for mail*, 139 Stanford Ave., Elyria, Ohio.
Llewellyn, John Earl (J'41) (J), Mech. Engr., Carnegie-Ill. Steel Corp., 1 Library Pl., Duquesne; *for mail*, 1251 Wisconsin Ave., Pittsburgh 16, Pa.
Llewellyn, Walter E. (J'41) (NYL), Mech. Engr., Chem. Div., E. I. du Pont de Nemours & Co., 626 Schuyler Ave., Arlington, N.J.
Lloyd, Bernard Hugh (J'44) (CMZ), Research Mech. Engr., Ont. Research Foundation, 43 Queens Park; *for mail*, 5 Devonshire Pl., Toronto 5, Ont., Can.
Lloyd, Frank H. (J'41) (ACG), Btry. H, 209th Coast Arty. Corps (AA); Camp Stewart, Ga.; *for mail*, 183 Vassar St., Rochester, N.Y.
Lloyd, George Arthur (J'45) (JBNESF), Instr., Univ. of B.C.; *for mail*, 4119 Georgia St. E., Vancouver, B.C., Can.
Lloyd, John Armon ('36; '44) (AFS), Mech. Engr., John Lloyd & Sons, Bennett Bldg., Wilkes-Barre, Pa.
Lloyd, Joseph Robert (J'46), 2355 Hilgard Ave., Berkeley 4, Calif.
Lloyd, Robt. S. (J'39), Cadet Engr., Sales Engr., Trng. Course, Buffalo Forge Co., 191 Mortimer St., Buffalo, N.Y.
Lloyd, Russell G. (J'39) (BEK), Stress Analyst, Babcock & Wilcox Co.; *for mail*, 74 Fernwood Ave., Barberton, Ohio.
Lloyd, Wm. ('36), Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Lobben, Peder ('95), Skotbu P.O., Norway.
Lobisser, Raymond John (J'41) (JLS), Mech. Designer, Radiation Lab., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39; *for mail*, 138 Pleasant St., East Walpole, Mass.
Lobley, Fred Askew ('37) (LCJDMN), Ch. Engr., Miles Labs., Inc., 1127 Myrtle St., Elkhart, Ind.
Lobo, Walter Eder ('44) (KBEF), Dir., Chem. Engrg. Dept., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.; *for mail*, 2 Stoneleigh Park, Westfield, N.J.
Lock, Thomas (J'40) (CMNBAD), Devel. Engr., Thompson Products Co., 2196 Clarkwood Rd., Cleveland 3; *for mail*, 4102 Maplewood Ave., Cleveland 9, Ohio.
Locke, David H. (J'38), Lt., U.S.N.; 817 S. 51st St., Philadelphia, Pa.
Locke, Emanuel David ('29; '35) (MW), Examiner, Mch. Tool Sec., War Assets Corp., 256 Montgomery St., San Francisco 4; *for mail*, 45 Lincoln Way, San Francisco 22, Calif.
Locke, Melvin Wardle ('44) (CKS), Supt., Super-Power Dept., Ch. Engr., Commonwealth Edison Co., Box 149, Pekin, Ill.
Locke, R. A. (A'36) (KPS), Mgr., Steel Heating Boiler Inst., Middletown, Pa.
Locke, Walter ('30; '35) (SFK), Sales Mgr., Edge Moor Iron Works, Edge Moor; *for mail*, 101 Perkins Ave., Wilmington 278, Del.
Lockeman, Geo F. ('22; '28; '32) (BJL), Mech. Engr., Procter & Gamble Co., Ivorydale; *for mail*, 45 Forest Ave., Wyoming, Cincinnati, Ohio.
Lockett, Kenneth ('04; '07; '13) (CO), War Assets Corp., 222 W. North Bank Dr., Chicago 54; *for mail*, 4440 Beacon St., Chicago 40, Ill.
Lockett, Robert Paine ('31; '35) (SECHKL), V.P., A. M. Lockett & Co., Ltd., 308 Whitney Bldg., New Orleans 7, La.
Lockett, Robert Paine, Jr. (J'37) (SEFC), A. M. Lockett & Co., Ltd., New Orleans; *for mail*, 2130 Palmer Ave., New Orleans 15, La.
Lockhart, Harold Alburn ('42) (HKS), Ch. Engr., Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove, Ill.
Lockhart, James D. (J'41) (BCD), 2 Perkins Manor, Jamaica Plain 30, Mass.
Lockhart, Robert William (J'44) (HKBA), Jr. Engr., Cabin Heater Research Project, Natl. Advis. Com. for Aero., Univ. of Calif., Berkeley 4; *for mail*, 835 Santa Barbara, Berkeley 7, Calif.
Lockwood, Frank A. ('21) (DMA), Design Engr., Silver Engrg. Works, Inc., 3309 Blake St., Denver 5; *for mail*, 815 Monroe St., Denver 6, Colo.
Lockwood, George Bachus ('42) (CFS), 105 Mountain View Ave., Port Richmond, S.I., N.Y.
Lodewyks, Peter Frederick (J'44) (ABH), Engrg. Draftsman, Broderick & Bascom Rope Co., Airport Way & Edmunds St., Seattle 8; *for mail*, 8447 Delridge Way, Seattle 6, Wash.
Lodge, H. M. (J'40) (FSA), Dist. Mgr., Am. Engrg. Co., 75 West St., New York 6, N.Y.; *for mail*, 122 Valley View Rd., Ridgewood, N.J.
Loeb, Leo ('11; '21) (CES), Pres., Supvg. & Cons. Engr., Pub. Utilities, Loeb & Eames, Inc., 57 William St., New York, N.Y.
Loebel, Fred Arthur (J'45), Project Engr., Cleaver-Brooks Co., 5100 N. 33rd St.; *for mail*, 3868 N. 52nd St., Milwaukee, Wis.

Loeber, Charles ('18; '23) (OMC), Dir. of Research (Biological), 26 Central Ave.; *for mail,* Box 295, Staunton, Va.
Loeffler, Alfons Georg ('45), Ch. Engr., Allen Billmyre Co., 437 Fayette Ave., Mamaroneck, N.Y.; *for mail,* Comstock Hill Ave., Norwalk, Conn.
Loeffler, Fritz ('24), Pres., Loeffler, Inc., 8 W. 47th St., New York 19, N.Y.
Loffler, John Edward ('42) (EHJKMN), Ch. Mech. Engr., Thornhill Craver Co., Inc., 1700 Lockwood Dr., Houston 1; *for mail,* 8417 Leander, Houston 12, Tex.
Loeffler, John Edward, Jr. (J'41) (LRF), Asst. Ch. Engr., Maint., Tin Processing Corp., Texas City; *for mail,* 8119 Niles St., Houston 12, Tex.
Loeffler, Walter Barnes (J'41) (AKNO), Capt., Army Air Forces, Dept. TSEPR-S, Air Tech. Serv. Comd., Wright Field, Dayton, Ohio.
Loepsinger, Albert John ('08; '13) (BHTKSZ), Dir. of Research, Grinnell Corp., 260 W. Exchange St., Providence 1, R.I.
Loercher, Herbert Willy (J'42), Warrant Officer (j.g.), U.S.A., 51 Sage Hall, Cornell Univ., Ithaca; *for mail,* 121 W. Quaker St., Orchard Park, N.Y.
Loewe, Peter L. (J'37) (OCNM), Pres., Midwest Engrg. Associates, 30 N. La Salle St., Chicago 2; *for mail,* 5801 Dorchester Ave., Chicago 37, Ill.
Loewen, Erwin G. (J'41), Pvt., Co. A, 1298th Engr. Combat Bn., Camp Shelby, Miss.
Loewen, Wilbert L. (J'41) (AKCE), Engr., Field Engrg. Dept., Gen. Elec. Co., 5 Lawrence St., Bloomfield; *for mail,* 437 N. Walnut St., East Orange, N.J.
Loewy, Raymond Fernand (A'44), Owner, Firm of Raymond F. Loewy, 580—5th Ave., New York; *for mail,* Sands House, Sands Point Rd., Sands Point, N.Y.
Lofft, William Arthur ('32) (TSFKLB), Ch. Engr., Eavenson & Levering Co., 3rd & Jackson Sts., Camden; *for mail,* 515 Bettlewood Ave., Collingswood, N.J.
Lofgren, Gustaf Einar ('43) (CBJY), Dir. of Research, Electrolux Corp., 51 Forest Ave.; *for mail,* Box 13, Old Greenwich, Conn.
Lofgren, Kenneth E. ('36) (BCM), Instr., Mch. Design, Cooper Union, Cooper Sq., New York, N.Y.
Lofstedt, Carl J. ('25; '29), 260 Valentine Lane, Yonkers, N.Y.
Loftheim, Kaare J. (J'40) (CERMJS), 2517—27th St. S., Arlington, Va.
Lofthouse, Fred Richard (J'46), 1819 N. 9th, Springfield, Ill.
Lofts, David ('01), Retired; Shore Crest Hotel, Wrightwood at Pine Grove Ave., Chicago 14, Ill.
Logan, Frederick Dowell (J'44) (BKM), Co-Partner, Genn & Logan, 3959 Lincoln Ave., Chicago 13; *for mail,* 216 S. Clinton Ave., Oak Park, Ill.
Logan, George Henry (J'37) (MCNBJA), Exec. V.P., Logan Engrg. Co., Inc., Chicago; *for mail,* 6107 Newburg Ave., Chicago 31, Ill.
Logan, Jack Donald (J'42) (MJB), Asst. Engr., Cemco Elec. Mfg. Co., Ltd., 22 E. 5th Ave.; *for mail,* 1996 W. 41st Ave., Vancouver, B.C., Can.
Logan, John W. ('94; '99; '04) (LJFC), Secy., Treas., Alan Wood Steel Co., Conshohocken, Pa.
Logan, Orwell ('17; '35) (CAWMER), Retired; 842 The Alameda, Berkeley 7, Calif.
Logue, Charles Hayes ('22) (ABN), Cons. Engr., 123 Clarke St., Syracuse 10, N.Y.
Logus, Anthony Theodore ('42) (BCDHM), Ch. Engr., Pettibone Mulliken Corp., 4710 W. Division St., Chicago 51; *for mail,* 5242 W. 63rd Pl., Chicago 38, Ill.
Lohmann, Conrad G. ('27; '30; '35) (JLDGB), Tool Engr., Hart & Cooley Mfg. Co., E. 8th St.; *for mail,* 56 E. 21st St., Holland, Mich.
Lohse, Frederick E. ('31; '35) (CMJNHS), Works Mgr., V.P., Crown Fastener Corp., 30 Cutler St., Warren, R.I.
Loidl, Joseph Michael (J'34) (CMSHD), 1847 S. Clinton Ave., Trenton 10, N.J.
Loman, John Kohl ('35; '45) (CAMDJ), Engr., Haynes Stellite Co., Kokomo, Ind.
Lomax, Burt, Jr. (J'40) (OKCL), Sales Engr., Brandeis Mchy. & Supply Co., 2225 S. Brook St., Louisville, Ky.
Lombard, Nathaniel Alvah ('45) (ACMNHJ), Supt. Tooling, Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 634 Arden Ave., Glendale 2, Calif.
Lombardi, Eli Faust (J'45), Engr., Robins Conveyors, Inc., 270 Passaic Ave., Passaic; *for mail,* 448—21st Ave., Paterson 3, N.J.
Lombardo, Joseph Michel (J'44), Pvt., Co. A, 380th Engr. Combat Bn., A.P.O. 14561, c/o Postmaster, San Francisco, Calif.
Lommel, Robert George (J'42) (ABESFH), Sr. Test Engr., Wright Aero. Corp., Paterson; *for mail,* Box 441, Packanack Lake, N.J.
Lomonossoff, Boris Marcovitch (J'44) (EKDF TZ), Mech. Engr., Engrg. & Research Corp., Riverdale, Md.; *for mail,* 2605 Connecticut Ave., N.W., Washington, D.C.
London, Alexander Louis ('35; '45) (KBZLEH), Lt. Comdr., U.S.N.R., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* 3348 Martha Custis Dr., Alexandria, Va.
London, George (J'37) (SLE), Cons. Engr., 2390 Tiebout Ave., New York 57, N.Y.
London, Woodrow Parker ('45) (FLS), Designing Draftsman, H. G. Acres & Co., Niagara Falls, Ont., Can.
Loney, Neil McIntyre ('41) (DMF), Ch. Works Engr., Fisher Body Div., Gen. Motors Corp., 11-148 Gen. Motors Bldg., Detroit 2, Mich.
Long, Clarence Hardy (J'41) (SFRKEZ), Asst. Prof. Mech. Engrg., Va. Poly. Inst.; *for mail,* Box 504, 905 Preston Ave., Blacksburg, Va.
Long, David Raymond ('20) (GDNCML), Pres., Tagcraft Corp., 142 S. Christian St., Lancaster, Pa.
Long, Geo. Alex. ('14), 500 Simsbury Rd., Bloomfield, Conn.
Long, Herman (J'42), Test Engr., Ford Instrument Co., Rawson St., Long Island City; *for mail,* 31-44—95th St., Jackson Heights, N.Y.
Long, James (J'39) (RABYH), Loco. Designer, Whitcomb Loco. Co.; *for mail,* 115 S. Washington St., Rochelle, Ill.
Long, John Ewing ('42) (RMSFCE), Gen. Sales Mgr., Franklin Ry. Supply Co., Inc., 60 E. 42nd St., New York 17, N.Y.
Long, John J. ('19; '25) (CMW), The Arcady Hotel, 2619 Wilshire Blvd., Los Angeles, Calif.
Long, John Richard (J'42) (ABJ), Stress Analysis Engr., Aluminum Co. of Am., 2210 Harvard Ave., Cleveland 5, Ohio.
Long, Joseph Dorge (J'45) (CMTSLD), Draftsman, West Coast Shipbuilders, Ltd., 205 W. 1st Ave.; *for mail,* 520 E. 7th Ave., Vancouver, B.C., Can.
Long, Paul John, Jr. (J'41), Ensign, 1931—16th St., N.W., Washington, D.C.
Long, Ralph Henry, Jr. (J'43) (ACJN), U.S.N.R., PCE-868, F.P.O., San Francisco, Calif.; *for mail,* 504 Smith St., Vienna, W.Va.
Long, Richard Hubert ('35; '42) (MCNBS), Engr., 403 Montgomery Bldg., Spartanburg, S.C.
Long, Robert Irving (J'42), 2504—29th St. W., Seattle, Wash.
Long, Wilber Owen ('44) (CDHKLZ), Asst. Ch. Draftsman, Fluor Corp., Ltd., 1012 Baltimore Ave., Kansas City; *for mail,* 3733 Gillham Rd., Kansas City 2, Mo.
Longabaugh, George P. ('45) (BCDJMN), Dir., Mexican Project, Westinghouse Elec. Corp., Clark-Wright Bldg., East Pittsburgh, Pa.
Longacre, George Washington (J'44) (HKN), Supvr., Mech. Equip. Design Gp., RCA Victor Div., Radio Corp. of Am., 501 N. LaSalle St., Indianapolis, Ind.
Longbucco, James Richard (J'44) (EBKSFN), Merchant Mar.; 428 Bellevue Ave., Yonkers, N.Y.
Longcoy, Grant B. ('24) (SFLD), Mech. Engr., with Joseph Breslove, Cons. Engr., Leader Bldg., Cleveland 14; *for mail,* 1215 Ramona Ave., Lakewood 7, Ohio.
Longenecker, Charles ('92), Commercial Trust Bldg., Philadelphia, Pa.
Longfield, William Francis ('44), Ch. Engr., Cleveland Punch & Shear Works Co., 3917 St. Clair Ave., Cleveland; *for mail,* 3467 Woodridge Rd., Cleveland Heights, Ohio.
Longley, Rodger E. (J'39) (CJM), 104 Nahant St., Lynn, Mass.
Longmaid, Sydney E. (J'28) (CMBJ), Exec. V.P., Esterbrook Pen Co., Camden, N.J.
Longmire, Harry Earl ('45) (CSMLEO), Sales Engr., Gen. Elec. Co., 700 Antoinette St., Detroit 2, Mich.
Longstreth, Charles ('99), Retired; 611 A Ave., Coronado, Calif.
Lonngren, Bruno Leonard ('44) (BNARHJ), Mech. Engr., Bucyrus-Erie Co.; *for mail,* 702 Michigan Ave., South Milwaukee, Wis.
Lontz, Dudley M. ('40) (CGM), Box 54X, R.R. 1, Indian Hill, Cincinnati, Ohio.
Loomis, Allen ('20) (HW), 915 E. Beardsley Ave., Elkhart, Ind.
Loomis, Burdett, Jr. ('04), 812 Longmeadow St., Longmeadow 6, Mass.
Loomis, Evarts Greene ('43) (NYLH), Pres., Evarts G. Loomis Co., 126 S. 14th St., Newark 7; *for mail,* 275 Montclair Ave., Newark 4, N.J.
Loomis, Wayland E. (J'37) (BCS), Planning & Control Engr., Hartford-Empire Co., Drawer 1620, Hartford 1, Conn.
Loonin, H. Howard (J'44) (JMY), Design Engr., Lewyt Corp., 157 Chambers St., New York; *for mail,* 3001 W. 1st St., Brooklyn, N.Y.
Lootens, Edward D. (J'44) (CJM), Time Study Engr., Monarch Aluminum Co., 9301 Detroit Ave.; *for mail,* 10507 Lake Ave., Cleveland 2, Ohio.
Lopez, Luis Mascott ('41) (BEHKNS), Regional Engr., Westinghouse Elec. Internatl. Co., Juarez 4; *for mail,* Box 2568, Mexico, D.F., Mex.
Lord, Albert E. ('23; '35), Ch. Engr., Jos. H. Meyer Bros., 212—25th St., Brooklyn, N.Y.; *for mail,* 24 Highland Ave., Kearny, N.J.
Lord, David G. (J'41) (DJM), 66 Victory St., Cranston, R.I.
Lord, G. Ross ('32; '45) (HBA), Assoc. Prof. Mech. Engrg., Univ. of Toronto, Toronto, Ont., Can.
Lord Harry Chester ('30) (O), Pres., John A. Stevens, Inc., 16 Shattuck St., Lowell, Mass.
Lord, Kenneth Milton (J'37) (DMLCH), Engr., Maint. & Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn 3; *for mail,* 85 Jenness St., Lynn, Mass.
Lord, Thomas Biddle (J'46) (CLS), 1004 Merced St., Berkeley 6, Calif.
Lorenz, George Joseph (J'44) (AMNCH), Lt., U.S.N.R., 3110 Fanshawe St., Philadelphia 24, Pa.
Lorenz, Joseph George, Jr. (J'43) (KLNHJB), Mech. Engr., Design, F. J. Stokes Mch. Co., Tabor Rd. & Adams Ave., Philadelphia 20; *for mail,* 3221 Unruh Ave., Philadelphia 24, Pa.
Lorenzini, Robert A. (J'40) (HKS), Serv. Engr., Serv. Dept., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Lorig, Marvin B. (J'43) (ZHCBY), Mech. Engr., Carbide & Carbon Chemicals Corp., South Charleston 3; *for mail,* 1506 Virginia St. E., Charleston 1, W.Va.
Lorimer, Gordon Alexander (J'46), Engr. & Designer, with R. A. Hanright, Cons. Engr., 14 Strachan Ave.; *for mail,* 17 Nanton Ave., Toronto, Ont., Can.
Loring, Samuel Jasper ('46) (ABO), Professional Engr., 29 Stiles St., Stratford, Conn.
Lormor, Henry Webb ('30) (DBMOJY), Mch. Devel. Engr., Willard Storage Btry. Co., 246 E. 131st St., Cleveland 1, Ohio.
Lorne, John Clifton (J'38), 658 W. Euclid, Detroit, Mich.
Losh, Clarence A. ('44), So. Pac. Co., 65 Market St., San Francisco, Calif.
Loskot, Bartholomew C. (J'45) (CDM), Maj., U.S.A., 155 W. 80th St., New York 24, N.Y.
Loss, Isidor Robert ('24; '42) (LCSEHN), Gen. Sales Dept., Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.; *for mail,* R.F.D. 1, South Norwalk, Conn.
Losse, Paul L. (J'40) (LZCDME), Process Engr., Baldwin-Hill Co., 501 Klagg Ave., Trenton 2; *for mail,* 334 Brinton Ave., Trenton 8, N.J.
Losse, Robert Henry (J'41) (BGSN), 1908 N. 31st St., Milwaukee 8, Wis.
Losson, Wesley L. ('28; '37) (AEBCMN), Asst. Mgr., License Div., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* R.F.D. 1, Pines Lake, Paterson, N.J.
Lott, Howard P. (J'33) (ZMC), M.M., Utah-Idaho Sugar Co.; *for mail,* Box 232, Chinook, Mont.
Lottes, Richard Frederick (J'43) Lt. (j.g.), U.S.N.R., GRO.PAC. 13, Navy 3934, F.P.O., San Francisco, Calif.; *for mail,* 604 M Ave., La Grande, Ore.
Lotz, Leonard Charles ('39) (DJ), Mgr., B. E. Jarvis, Inc., 74-80 Malvern St.; *for mail,* 16 Park Pl., Newark, N.J.
Lotz, Ralph Walter (J'41) (ZSHCM), 6242 N. Hoyne Ave., Chicago 45, Ill.
Lotz, Ralph William ('19; '23; '35) (CMNB), Dir. of Mfg., Apex Elec. Mfg. Co., 1070 E. 152nd St., Cleveland 10, Ohio.
Louden, James Keith ('41) (CMDYLJ), Prod. Mgr., Glass & Closure Div., Armstrong Cork Co., Liberty St.; *for mail,* 419 Atkins Ave., Lancaster, Pa.
Louderback, Page Gordon (J'39) (CMYD), Prod. Engr., O'Sullivan Rubber Co., Box 603; *for mail,* 424 W. Cecil St., Winchester, Va.
Loudon, David S. (A'38) (OMC), V.P., Treas., Geo. S. Armstrong & Co., Inc., 52 Wall St., New York 5, N.Y.
Louer, Harold Francis (J'44) (BHKS), Design Engr., Sanderson & Porter, Engrs., 52 William St., New York; *for mail,* 30 College Pl., Brooklyn 2, N.Y.
Loughney, Charles Edward, Jr. (J'38) (ABM JYN), Sr. Layout Designer, Glenn L. Martin Co., Middle River; *for mail,* 253 Linden Ave., Towson 4, Md.
Loughran, John J. ('20; '31), East. Dist. Sales Mgr., Doehler Die Casting Co.; *for mail,* 364 Walnut St., Pottstown, Pa.
Lourie, George Edward (J'35) (RSF), Maj., Trans. Corps, Instr. in Trans., Comd. & Gen. Staff Sch.; *for mail,* 627-1 McClellan, Ft. Leavenworth, Kan.
Loutrel, Cyrus Henry ('12; '16; '35), (J), Chmn. of Bd., Natl. Lock Washer Co., 40 Herman St., Newark 5; for mail, 270 Irving Ave., South Orange, N.J.
Louzecky, Paul J. (J'32) (BNERAM), Supvr., Tech. Dept., Cleveland Diesel Eng. Div., Gen. Motors Corp., 2160 W. 106th St., Cleveland 11; *for mail,* 1342 Brockley Ave., Lakewood 7, Ohio.
Love, Clyde Peter ('32; '46), Mine M.M., Andes Copper Min. Co., Potrerillos, Chile, S.A.
Love, Richard O. (J'41) (EKS), 16 Lexington Ave., Buffalo, N.Y.
Love, Russell LeRoy (J'42) (NSCJLM), Lt. (j.g.), U.S.N.R., Engr. Officer, U.S.S. *Bremerton,* F.P.O., San Francisco, Calif.; *for mail,* 726 Brooklyn Ave., New Castle, Pa.
Lovejoy, Elijah P. (A'35) (CS), News Dir., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Lovejoy, Stanley Williams, Jr. (J'46) (KOS), 133 Park Ave., Swarthmore, Pa.
Lovekin, Raymond E. (A'27) (BEGJKS), Pres., R. E. Lovekin Corp., 200 N. 15th St., Philadelphia; *for mail,* 8 Chester Pike, Ridley Park, Pa.
Lovelace, Richard Shriver (J'40) (BEWYHJ), Asst. Prof. Naval Constr., Mass. Inst. of Tech., 55 Massachusetts Ave., Cambridge 39, Mass.
Lovell, George H., Jr. (J'41) (CKS), Min. Dept., Lehigh Navigation Coal Co., Inc., Lansford, Pa.
Lovell, Kenneth Roscoe (J'41) (BJ), Assoc. Engr., Bur. of Ships, Navy Dept., Navy Bldg. T4-2912, Washington, D.C.; *for mail,* Apt. 8, 2205—2nd St. S., Arlington, Va.

Lovell, Thos. S. ('36), 951 Kitchener Ave., Detroit, Mich.
Lovely, John Emerson ('16; '25) (MNAC), *Vice-President,* '44-'45, *Director-at-Large,* '45-'46; V.P., Ch. Engr., Jones & Lamson Mch. Co., 160 Clinton St., Springfield, Vt.
Lovercheck, Charles Lester (J'42) (COS), Engr., Aero. & Mar. Eng., Gen. Elec. Co., River Rd., Schenectady 5; *for mail,* 207 Sunnyside Rd., Schenectady, N.Y.
Lovett, Louis E. ('28) (CLT), Firm of Louis E. Lovett, Engrs., Madison, Ohio.
Lovette, Stanislaus A. ('42) (ACERSW), Mech. Engr., Norfolk & West. Ry. Co.; *for mail,* 405 Mountain View Terrace, Roanoke 15, Va.
Low, Herbert Melvin ('44) (EFHS), Mgr., Power Div., J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Mo.
Low, John Judson ('45) (BHN), Ch. Engr., Kay-Brunner Steel Products Inc., 999 Meridian Ave., Alhambra; *for mail,* 841 S. Orange Dr., Los Angeles 36, Calif.
Low, John Routh, Jr. ('45) (JBA), Head, Div. of Metal., Pa. State College, State College, Pa.
Low, Sidney (J'41) (JBS), Research Engr., Chapman Valve Mfg. Co., 203 Pinevale St., Indian Orchard; *for mail,* 144 Buckingham St., Springfield 9, Mass.
Lowe, H. Leland ('03; '14) (CS), Cons. Engr., Stone & Webster Engrg. Corp., 90 Broad St., New York, N.Y.
Lowe, Stuart S. ('23; '25; '35) (CM), Prod. Scheduling Div., War Prod. Bd.; *for mail,* 1731 Irving St., N.W., Washington, D.C.
Loweke, George Paul ('45) (BHJAKG), Asst. Prof., Mech. Engrg. Dept., Wayne Univ., Detroit 1, Mich.
Lowell, Chapin McKinney (J'40) (BR), 505 Oakwood St., East Aurora, N.Y.
Lowen, Walter (J'43) (ESAKFR), Instr., N.C. State College, 105 Page Hall; *for mail,* 2707 Bedford Ave., Raleigh, N.C.
Lowenstein, Herbert M. ('19; '25; '30) (SFE), Proposition Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 865 West End Ave., New York 25, N.Y.
Lowenstein, Seymour (J'41) (CMNZ), Gage Research & Design Engr., Inspc. Gage Sub-Office, Ord. Dept., U.S.A., 238 E. Wyoming Ave., Philadelphia 20, Pa.; *for mail,* 33—5th Ave., New York 3, N.Y.
Lower, Nathan M. ('13), Pres., Stoker Mfg. & Engrg. Co., 2500 E. 12th St.; *for mail,* 4802 Clinton Dr., Dorchester Heights, Erie, Pa.
Lowman, Albert Hurd (A'30) (CJS), Pres., Treas., Phoenix Specialty Mfg. Co., Inc., 155 Wooster St., New Work 12; *for mail,* 60 Jarvis Pl., Lynbrook, N.Y.
Lowrey, John Christopher (J'44) (BCDJMS), Gen. Mgr., Salco Corp. Ltd., Long Branch; *for mail,* 207 Ellis Ave., Toronto, Ont., Can.
Lowry, Curtis Milton ('31; '35) (BDES), Prof. & Head, Engrg. Dept., Stetson Univ.; *for mail,* 510 W. Minnesota Ave., De Land, Fla.
Lowry, Paul Miller (J'44) (CF), Sales Engr., Estimator, Butler Mfg. Co., 13 Eastern Ave., Kansas City 3; *for mail,* 703 N. Noland St., Independence, Mo.
Lowther, Geo. Wesley ('29; '38) (SZCF), Mech. Engr., Tex. Gulf Sulphur Co., Newgulf, Tex.
Lowther, Joseph Gibson ('39) (BJ), Lt., U.S.N.R., Naval Supply Depot; *for mail,* 5 E. Maplewood, Mechanicsburg, Pa.
Lowther, William Glynn (J'34) (JKY), Inspc. Engr., Neches Butane Products Co., Port Neches; *for mail,* 2224 Wignall Ave., Port Arthur, Tex.
Lowy, Mortimer J. (J'45) (BZAHM), Stress Analyst, Consltd. Vultee Aircraft Corp., Vulteefield, Downey, Calif.
Lowy, Robert ('40) (BHS), Engr., Baldwin Loco. Works, Eddystone; *for mail,* 108 Glencoe Rd., Upper Darby, Del. County, Pa.
Loyd, Albert E. ('31; '35), Sales Engr., Roots-Connersville Blower Corp., 21 West St., New York 6, N.Y.; *for mail,* 1141 Stilford Ave., Plainfield, N.J.
Loyd, Robert Rolland (J'40) (FEKS), Engr., Mch. Design, "Packomatic" or J. L. Ferguson Co., Route 52, Joliet; *for mail,* R.F.D. 1, Frankfort, Ill.
Lubbert, Geo. L. ('16; '35), Elec. & Mech. Engr., Manhattan & Merville Aves.; *for mail,* 5609 Merville Ave., Baltimore, Md.
Lucarelle, Joseph Marseglia ('29) (CMNBJD), Gen. Supt., Dictaphone Corp., 375 Howard Ave., Bridgeport 5, Conn.
Lucas, John Lewis (J'42) (MCLZDO), M.M., Gen. Chem. Co., Box 2429, Baton Rouge 2, La.
Lucas, John W. ('37; '40) (CF), Ch. Fuel Engr., Old Ben Coal Corp., 230 S. Clark St., Chicago 4, Ill.
Lucas, Joseph A. ('21) (G), Retired; 111th St., Richmond Hill, L.I., N.Y.
Lucas, Merle Ward (J'41) (SBKFEA), Field Engr., Gen. Elec. Co., 1801 N. Lamar St., Dallas 2, Tex.
Lucas, Thomas Elza (J'44), U.S.N.; 1728 Washington Blvd., Ogden, Utah.
Lucas, William Francis (J'38) (LCM), Ch. Engr., Brown-Forman Distillers Corp., 1908 Howard St., Louisville 1; *for mail,* 3003 Sunnyside Dr., Louisville, Ky.
Lucca, Frank Paul (J'44) (SHKELC), Field Serv. Engr., Westinghouse Elec. Corp., 420 S. San Pedro St., Los Angeles 13; *for mail,* Apt. 5, 2460 Cedar Ave., Long Beach 6, Calif.
Luce, Alexander W. ('27; '34) (BCM), Box 212, R.F.D. 1, Springfield, Vt.
Luce, George Griggs (J'42), 1828 Campbell Ave., Schenectady 6, N.Y.
Luce, Richard Sanborn ('19; '35) (EFBJNS), Sales & Serv. Engr., Double Seal Ring Co., 157 Chambers St., New York 7; *for mail,* 168—73rd St., Brooklyn 9, N.Y.
Lucht, Frederick William, Jr. ('15; '20; '35) (MJNBCL), Devel. Engr., Carboloy Co., Inc., Box 237, R.P.A., Detroit 32; *for mail,* Route 3, 17605 Glenwood Blvd., Birmingham, Mich.
Lucia, Robert Elliot (J'44), Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland, Ohio.
Luck, Harold M. (J'43) (AZCEBM), Mech. Engr., Raytheon Mfg. Co., Willow St., Waltham; *for mail,* 350 Seaver St., Roxbury, Mass.
Lucke, Charles Edward ('30; '08; F'38) (KESFAL), Prof. Emeritus, Mech. Engr., Spec. Lecturer, Pupin Bldg., Columbia Univ., New York 27, N.Y.
Lucke, Francis Charles (J'43) (KSFENO), Serv. Engr., Steam Div., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4, Pa.; *for mail,* 658—68th St., Brooklyn 20, N.Y.
Lucken, Ernest George (J'42) (SEH), Research & Devel. Engr., Aerojet Engrg. Corp., Box 631, Pasadena 19; *for mail,* 831 N. Oakland Ave., Pasadena 6, Calif.
Luckes, Robert Frank (J'42) (CMD), Prod. Engr., Chain Belt Co., 45th & Greenfield, Milwaukee; *for mail,* 1307 N. Franklin Pl., Milwaukee 2, Wis.
Luckie, George O. ('28; '35) (DLW), Ch. Engr., Casein Co. of America Inc., Div. of Borden Co., 350 Madison Ave., New York 17; *for mail,* 135 N. Chatsworth Ave., Larchmont, N.Y.
Lucking, Walter Theodore ('43; '46) (S), 1 Haddalve Ave, Wheeling, W.Va.
Luckner, Leo B. (J'36) (BJM), Mem. Tech. Staff, Bell Tel. Labs. Inc., 463 West St., New York 14; *for mail,* 8713—55th Ave., Elmhurst, L.I., N.Y.
Lucky, Murray Elbert (J'45) (SNKCZH), Sr. Mech. Engr., Commonwealth & So. Corp., 600 N. 18th St., Birmingham 3, Ala.
Lucy, Sam G. (J'33) (CDK), Maj.; 104-12 Continental Ave., Forest Hills, N.Y.
Ludlum, W. J. (J'37) (L), Engr., Am. Sugar Refinery Co., Key Highway E.; *for mail,* 600 W. 37th St., Baltimore 11, Md.
Ludolph, Karl Horst (J'45) (CEFKOS), Ch. Engr. & Bldg. Mgr., Am. Woman's Realty Co., Inc., 353 W. 57th St., New York 19, N.Y.
Ludwick, William Leslie ('42) (OLZH), Tech. Dir., Peck, Stow & Wilcox Co., Center St., Southington; *for mail,* 4463 Whitney Ave., Mt. Carmel, Conn.
Ludwickson, James Kenneth ('39; '45) (BEFGKS), Asst. Prof. Mech. Engrg., Univ. of Neb., Lincoln; *for mail,* 4302 Sheridan Blvd., Lincoln 6, Neb.
Ludy, Llewellyn V. ('05) (SEKFBZ), Retired; 600 Russell St., West Lafayette, Ind.
Luebs, A. A. ('19; '35) (SEFA), Assoc. Prof. Mech. Engrg., Univ. of Neb., Lincoln 8, Neb.
Lueckel, William John ('28) (DMY), Partner, Dunning Lueckel Engrg. Co., 17 John St., New York 7, N.Y.
Luedicke, Alex Henry, Sr. ('20; '35) (BKLOS), Cons. Engr., 2379 N. 46th St., Milwaukee 10, Wis.
Luehrmann, Hugh ('31) (SKYELF), Mem., Firm of Walter Cooke, 816 Howard Ave., New Orleans 13; *for mail,* 1213 Nashville Ave., New Orleans 15, La.
Luehrs, Daniel Michael ('44) (CJKMNS), Indus. Engr., Naval Arch., 1901 Green St., Philadelphia 30, Pa.
Luehrs, Hans J. ('32) (GBNYJM), Devel. Engr., C. B. Cottrell & Sons Co.; *for mail,* Avondale, R.F.D. 1, Westerly, R.I.
Luepker, Frank Nicholas, III (J'46), 937 Dover Pl., St. Louis 11, Mo.
Luer, George (J'43) (ABME), Aviation Electronics Technician's Mate, 3rd Class, U.S.N.R.; *for mail,* William St., Hicksville, N.Y.
Lufkin, Clarence Rawson (J'44) (AC), Serv. Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.; *for mail,* 127 Overhill Rd., Salina, Kan.
Lufkin, Garland ('23; '26; '29) (DMYNJZ), V.P., Owens-Ill. Glass Co., Box 1035, Toledo 1, Ohio.
Luiggi, Mario L., ('21; '35) (SRFA), Sales Engr., Gen. Motors Overseas, 1775 Broadway, New York 17, N.Y.
Luke, Chester P. (J'42), Capt., Sqd. D, Combat Crew Trng. Sch. (LB), Army Air Field, Florence, S.C.
Lukens, Arch M. ('41) (NACBKM), Prof. & Head, Dept. of Mech. Engrg., New Mex. College of Agric. & Mech. Engrg., State College, New Mex.
Lukens, William L. ('17; '21) (CJMNYZ), Ch. Ord. Engr., Picatinny Arsenal; *for mail,* 6 Mt. Prospect Ave., R.D. 1, Dover, N.J.
Lukey, Robert Kenneth (J'42) (CJM), Ensign O-V(P), U.S.N.R.; 396 Uvedale Rd., Riverside, Ill.
Lumbers, Clifton Glen (J'37) (CMWDA), Plant Supt., Aristocrat Mfg. Co. Ltd., 7 Fraser Ave., Toronto, Ont., Can.
Lumsden, Walter Branham, Jr. (J'41) (GJB), Naval Arch., P-2, Naval Shipyard, Charleston; *for mail,* 2 Atlantic St., Charleston 1, S.C.
Lund, Bjorn (J'41) (NBAZH), Engr., Rectifier Design & Devel., Allis-Chalmers Mfg. Co., Milwaukee; *for mail,* 825 N. 25th St., Milwaukee 3, Wis.
Lund, Charles Richard, Jr. (J'44), U.S.N.R., 209 Park Dr., Catonsville, Md.
Lund, Sidney C. (J'37) (HBZM), Engr., Control Instrument Co., 67—35th St., Brooklyn 31; *for mail,* 669—2nd Ave., New York 16, N.Y.
Lundberg, Gosta Alexander (J'39) (BEH), Asst. Mech. Engr., Magnolia Pipe Line Co., Box 900, Dallas; *for mail,* 5626 Bell Ave., Dallas 6, Tex.
Lundbye, Axel E. ('30; '31) (GJLNY), Crowell-Collier Research Lab., 254 W. 54th St., New York 19, N.Y.
Lundelius, John Fredrik (J'42) (CJBHM), Mech. Engr., Superior Tube Co., Collegeville; *for mail,* R.D. 1, Malvern, Pa.
Lundgren, Iver Herbert ('34; '44) (NBJ), Engr., Republic Steel Corp., 8th St., N.E., Canton; *for mail,* 1519 Ridgeway Pl., N.W., Canton 3, Ohio.
Lundin, Bruce Theodore (J'42) (KABHE), Mech. Engr., Sec. Head, Jet Propulsion Research, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 1595 Riverside Dr., Lakewood 7, Ohio.
Lundin, Myrin I. (J'44) (CLOEMA), Tech. Engr., Supvr., Carbide & Carbon Chemicals Corp., Box P; *for mail,* 105 E. Holston Lane, Oak Ridge, Tenn.
Lundquist, Arvid ('20; '35) (BMNJGY), Project Engr., Farrand Optical Co., Inc., 4401 Bronx Blvd., New York 66; *for mail,* 33 Riverside Dr., New York 23, N.Y.
Lundquist, Elmer Clinton (J'38) (ABKH), Asst. Prof. Aero., Engrg. Bldg., State Univ. of Iowa, Iowa City, Iowa.
Lundquist, George Teofile (J'42) (DMCN), Mgmt. & Estimating, Bay City Iron Works, 4th & Washington Sts., Oakland 4; *for mail,* 24 Estrella Ave., Piedmont 11, Calif.
Lundsted, James Herbert (J'40) (SZKAFM), Asst. Test Engr., Kansas City Power & Light Co., 1300 Baltimore; *for mail,* 2234 E. 67th St. Terrace, Kansas City 5, Mo.
Lundstrom, Carl Brynolf ('25; '34; '35) (WYCMD), Factory Mgr. & Dir., Stand. Furniture Co., Herkimer; *for mail,* 38 Waverly Pl., Little Falls, N.Y.
Lundstrom, Ernest A. ('42) (KFS), Asst. Ch. Engr., Air Preheater Corp., 60 E. 42nd St., New York 17, N.Y.
Lundy, William L. ('20; '25; '35) (EFKSWZ), Staff Steam Engr., Kimberly-Clark Corp., 128 N. Commercial St., Neenah, Wis.
Luney, Edward Ross ('37) (CKLMO), Process Engr., Hiram Walker & Sons, Inc., Ft. Edmund St., Peoria; *for mail,* Sherman Ave., Peoria 5, Ill.
Luney, Frank Solon ('36) (ADFMNY), Mech. Engr., Barnes & Reinecke, 230 E. Ohio St., Chicago, Ill; *for mail,* 6347 Van Buren Ave., Hammond, Ind.
Lunn, John A. ('17; '25; '29) (CDKLYM), V.P., Dewey & Almy Chem. Co., 62 Whittemore Ave., Cambridge 40; *for mail,* 37 Larch Rd., Cambridge 38, Mass.
Lupke, Paul, Jr. ('22; '35) (CLS), Ch. Chemist, Cons. Engr., Essex Rubber Co., Inc., Beakes & May Sts., Trenton; *for mail,* 771 E. State St., Trenton 9, N.J.
Luppold, George Ernest, Jr. (J'42) (ZSFE), Sales-Serv. Engr., Mech. Engr., Bailey Meter Co., 1709 W. 8th St., Los Angeles 14, Calif.
Lusink, Clarence Irving ('40) (RJB), Ch. Mech. Engr., Symington-Gould Corp., 25 Lincoln Ave., Rochester 3; *for mail,* 90 Margaret St., Rochester 11, N.Y.
Lusk, Edward Albert (J'41) (BCJMYG), Lt., U.S.N.R., Res. Naval Inspr. of Ord., Bur. of Ord., Navy Dept., 1620 Central, N.E., Minneapolis 13, Minn.
Lusk, James B. (J'36) (BKS), c/o S.D. State College, Brookings, S.D.
Lusk, Sherman W. ('41) (SCEF), Asst. Engr., Bldg. Serv. Dept., Prudential Ins. Co., 763 Broad St., Newark 2, N.J.
Luster, Donald Raymond (J'40) (BJY), Engr., Charge of Gun Design, Remington Arms Co., Ilion; *for mail,* 402 Prospect St., Herkimer, N.Y.
Lustwerk, Ferdinand (J'42) (BKM), 125 Baker St., West Roxbury, Mass.
Luth, Harold John ('42) (CWY), Pres., Harsyd Chemicals Inc., 397 W. 21st St., Holland, Mich.
Luthe, Henry P. (J'37) (CDK), Treas., Luthe Hardware Co., 100 Court Ave., Des Moines 5, Iowa.
Lutz, George ('24; '35) (CYNJ), Ch. Engr., Carl Ford, Inc., 311 W. 66th St., New York 23; *for mail,* 64-41 Madison St., Brooklyn 27, N.Y.
Luukkonen, Voitto Arnold (J'39) (NBJMAC), Ord. Engr., Springfield Armory, Springfield; *for mail,* 114 Williams St., Longmeadow, Mass.
Luzzatto, Alvin S. (J'42) (BCDHMO), Plant Layout & Method Engr., R. R. Donnelley & Sons, Lakeside Press, 350 E. 22nd St., Chicago, Ill.
Luzzatto, Giovanni Welding (J'40) (AMB), Asst. Prod. Engr., Sperry Gyroscope Co., Inc., Manhattan Bridge Plaza, Brooklyn 1; *for mail,* 47 Plaza St., Brooklyn 17, N.Y.
Lyerly, Ray Lentz (J'43) (AER), Lt. (j.g.), U.S. Maritime Serv.; *for mail,* R.F.D. 1, Newton, N.C.

Lyford, Frederic Eugene ('36) (COR), Asst. to Chmn. of Bd., Merritt-Chapman & Scott Corp., 17 Battery Pl., New York 4, N.Y.
Lyle, James McKee ('44) (FKLSYZ), St. Louis Mgr., Henry Vogt Mch. Co., 1343 Syndicate Trust Bldg., 915 Olive St., St. Louis 1; *for mail,* 6838 Julian Ave., St. Louis 14, Mo.
Lyman, Hugh Marion ('45) (CDLOS), Pres., Lyman Bros., 512 McIntyre Bldg., Salt Lake City, Utah.
Lyman, John Robert (J'46) (CFP), Ensign, U.S.N.; 17 Silver St., Middletown, Conn.
Lyman, Oliver Bridgman ('28) (EKS), Owner, Firm of Oliver B. Lyman, Sales Rep. for Mech. Equip., 116 New Montgomery St., San Francisco 5, Calif.
Lyman, Theodore Benedict (J'40) (CM), Dir. of Labor Relations Research, Matson Navigation Co., 215 Market St., San Francisco 5, Calif.
Lynah, James ('22), V.P., Gen Mgr., Brunswick Mar. Constr. Corp., Brunswick; *for mail,* 330 E. 46th St., Savannah, Ga.
Lynam, John Wesley (J'41) (AER), 303 Pennsylvania Ave., Wilmington Manor, New Castle, Del.
Lynam, William A. (J'36) (CJLM), Asst. Prof., Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6, Ohio.
Lynch, Daniel Gail ('40) (SKEF), V.P., J. T. Thorpe & Son, Inc., 941—16th St., San Francisco 7, Calif.
Lynch, Frank W., Jr. (J'41) (JYCD), Shop Supvr., Precision Heat Treating Co., 216 William St., New York 7; *for mail,* 3627 Corporal Stone St., Bayside, N.Y.
Lynch, John Edward, Jr. ('21; '35) (FLSYZ), Mech. Designer, Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 168 Bartlett Rd., Winthrop, Mass.
Lynch, Thomas Francis (J'45) (DJNLCB), M.M., Andes Copper Min. Co., Potrerillos, Chile, S.A.
Lyne, Lewis F., Jr. ('45) (C), Examiner, Plant Clearance, Reconstruction Finance Corp., 70 Pine St., New York 5, N.Y.; *for mail,* 3 Sunset Dr., Asbury Park, N.J.
Lyness, Arthur Alois, Jr. (J'46), 606 Highland Ave., Towson, Md.
Lyngstad, Anders E. (J'37) (CDJ), U.S. Navy; 514 Union Ave., Steubenville, Ohio.
Lynn, Eric William ('41) (NMCWYZ), Project Engr., Conmar Products Corp., 140 Thomas St., Newark 1, N.J.; *for mail,* 220 W. 71st St., New York 23, N.Y.
Lynn, Richard Harold (J'38) (ACM), 1st Lt., Army Air Forces, Sqd. Engrg. Officer, Box 61, Army Air Field, Dodge City, Kan.
Lynn, Solomon Stephen (J'44) (ABMDHN), Prod. Control, Rek-O-Kut Co., 146 Grand St., New York 13; *for mail,* 730 E. 93rd St., Brooklyn 12, N.Y.
Lynn, Stanley ('45) (ASNJOL), Mgr., Gas Turbine & Gear Sales, Joshua Hendy Iron Works, Sunnyvale; *for mail,* 340 Churchill Ave., Palo Alto, Calif.
Lyon, Alan George ('45) (CMO), 11 St. James St., Gosforth, Newcastle-on-Tyne 3, England.
Lyon, Chas. S. ('39) (ACE), V.P., Charge Operas., Motor Haulage Co., Inc., 18 Amity St., Brooklyn; *for mail,* 15 Davis Rd., Port Washington, L.I., N.Y.
Lyon, Claude Lealon, Jr. (J'42) (ESK), 513 S. Amherst, Albuquerque, New Mex.
Lyon, Daniel R. ('45) (MCNOBG), Indus. Mech. Engr., Internatl. Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Lyon, James Andrew (J'45) (BCO), 280 Scarsdale Rd., Crestwood 7, N.Y.
Lyon, Lewis E., Jr. (J'40) (JBT), E. I. du Pont de Nemours & Co.; *for mail,* 44 N. Pearl St., Buffalo, N.Y.
Lyon, Percy Shepardson ('15; '19; '30) (CFK LMS), Pres., Gen. Mgr., C. H. Wheeler Mfg. Co., Lehigh & Sedgley Ave., Philadelphia 32, Pa.
Lyons, Bartholomew J. ('29; '35) (CLT), Pur. Agent, Celanese Corp. of Am., Box 444, Cumberland, Md.
Lyons, Daniel A. ('20; '28) (BCDJM), 4727 Roberts Way, Seattle 99, Wash.
Lyons, Francis Thomas (A'45) (JCMNB), Pres., Gen. Mgr., DoAll Hudson Co., Inc., 618 S. Salina St., Syracuse 4; *for mail,* 235 Wadsworth St., Syracuse 8, N.Y.
Lyons, Herbert Rodgers ('40) (SWLCD), Asst. to Ch. Engr., Diamond Match Co., Oswego, N.Y.
Lyons, Robert Edward (J'39) (BJCMSK), Matls. Engr., Bur. of Ships, Navy Dept., Washington 25, D.C.; *for mail,* 615 Ave. E, District Heights, Md.
Lype, Eric Fred ('44) (BKES), Instr. & Sec. Supvr., Ill. Inst. of Tech. & Inst. of Gas Tech., 3300 Federal St., Chicago 16; *for mail,* 1715 E. 67th St., Chicago 49, Ill.
Lyster, Thomas L. B. ('15) (FKLOYZ), Ch. Engr., Hooker Electrochemical Co., Buffalo Ave. & 47th St., Niagara Falls, N.Y.
Lytle, Charles Walter ('16; '21; '23) (CMLJ DN), Prof. Indus. Engrg., N.Y. Univ., University Heights, New York 53, N.Y.
Lytle, John E. ('36; '37) (EJ), c/o O. G. Pratt, Stone House, Bishopsgate, London, E.C. 2, England.
Lytle, William Orland ('30) (LCOM), Research Engr., Pittsburgh Plate Glass Co., Creighton, Pa.

M

Maack, Walter F. W. (J'40) (AKS), Lt., U.S.N.R., Air U.S.S. *Randolph,* CV-15, F.P.O., San Francisco, Calif.
Maag, Joseph Mark ('44), Dist. Mgr., Elliott Co., Shell Bldg., St. Louis 3, Mo.
Maahs, Carl Ernest (J'38) (CRS), Inspr., Appliances & Matl., Erie R.R. Co.; *for mail,* 832 Water St., Meadville, Pa.
Maak, Chas. ('16; '35) (FJS), Supt. of Erection, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Mabb, William Smith (J'35) (HSBANE), Mech. Engr., Chapman Valve Mfg. Co., 203 Hampshire St., Indian Orchard; *for mail,* 90 Monroe St., Agawam, Mass.
Mabee, Ernest W. ('26; '35) (CJM), Tool Engr., Dodge Div., Chrysler Corp., 7900 Jos. Campau Ave., Detroit 11; *for mail,* 4065 Field Ave., Detroit, Mich.
Mabey, Arthur R. ('16) (ZNB), Stands. Engr., Bristol Co., Waterbury 91; *for mail,* 22 Farnham Ave., Waterbury 37, Conn.
Mabie, Hamilton Horth (J'45) (BJM), Asst. Prof. Mech. Engrg., College of Engrg., Cornell Univ., Ithaca, N.Y.
Mabley, Carlton Ray, Jr. ('25; '32; '35) (FR SE), Gen. Mgr. of Indus. Sales, Island Creek Coal Sales, Co., 700 Guaranty Bank Bldg., Huntington 1, W.Va.
MacAfee, Ralph Evans ('24) (CFS), Mgr., East. Branch, Babcock Wilcox—Goldie-McCulloch Co., 312 Can. Cement Bldg., Montreal 2, Que., Can.
Macalister, Robert Norman ('13) (MRS), Retired; 1622 Cullom Ave., Chicago 13, Ill.
Macaluso, Francis Louis (J'42) (MNCWYZ), Design Engr., Indus. Electronics Div., Westinghouse Elec. Corp., 2519 Wilkens Ave., Baltimore 3; *for mail,* 14 S. Augusta Ave., Baltimore 29, Md.
Macan, William A., III (J'37) (ZSL), Field Engr., Leeds & Northrup Co., 1310 Am. Bldg., Cincinnati 2, Ohio.
MacArthur, Charles J. ('32) (CGJY), Ptg. Consultant, Reynolds Metals Co., 3rd & Grace St., Richmond 19, Va.; *for mail,* 296 Ridgewood Ave., Glen Ridge, N.J.
MacArthur, Robert (A'04) (CEFK), Htg. Engr., New Haven Gas Light Co., 80 Crown St., New Haven 6; *for mail,* 26 Maplewood Rd., New Haven 15, Conn.
MacArthur, Stanley Herbert ('44), N.Y. Air Brake Co., 420 Lexington Ave., New York 17, N.Y.
Macaulay, Donald Stuart ('30; '35) (DSCLMN), So. Calif. Mgr. & Assoc., Peter D. Bowley & Associates, Rm. 204, 1807 E. Olympic Blvd., Los Angeles 21; *for mail,* 2064 Watsonia Terrace, Hollywood 28, Calif.
Macauley, John Blair, Jr. ('37) (EAFBJ), Dir., Engrg. Research, Ethyl Corp., 1600 W. 8 Mile Rd., Detroit 20, Mich.
MacBriar, Wallace N. ('16) (CFS), Asst. V.P., Carnation Co., 111 W. Massachusetts St., Seattle, Wash.
MacCallum, George A. (J'40), Draftsman, Firestone Rubber & Latex Products Co., Ferry St., Fall River; *for mail,* 82 County St., Taunton, Mass.
Maccaronio, Vincent Santo (J'43) (SAHZK), 34-33 Crescent St., Astoria 1, L.I., N.Y.
Macconochie, Arthur F. ('40) (BCJMO), Prof. Mech. Engrg., Univ. of Va.; *for mail,* Box 1362, University, Charlottesville, Va.
MacCracken, Calvin Dodd (J'46) (AEK), Waldorf & Kerns, 103 Lafayette St., New York; *for mail,* 100 Overlook Rd., New Rochelle, N.Y.
MacCullough, Gleason Harvey ('21; '26; '30) (BAN), Prof. Engrg. Mechanics, Worcester Poly. Inst., Worcester 2, Mass.
MacDonald, Edward T. ('27; '35), Mech. Engr., Charge of Operas., Maint. & Pur., St. Marys Hospital, Buffalo & St. Marks Aves., Brooklyn, N.Y.
MacDonald, Frederick Malcolm (J'44) (AEN), Student, Mass. Inst. of Tech., Cambridge; *for mail,* 7 Winthrop St., Stoneham 80, Mass.
Macdonald, George Allo (J'40) (COSEBF), Indus. Engr., Westinghouse Elec. Corp., 40 Wall St., New York, N.Y.; *for mail,* 930 Summit Ave., River Edge, N.J.
MacDonald, John W. (J'39) (AFM), 1709 Jackson St., Omaha, Neb.
MacDonald, Karl ('15; '23) (SBNKJ), Cons. Engr., Worthington Pump & Mchy. Corp., Coats St.; *for mail,* 27 Nelson Ave., Wellsville, N.Y.
MacDonald, Kenneth Wallace ('42) (BJL), Ch. Design Engr., Anderson-Nichols & Co., 53 State St., Boston 6; *for mail,* 27 Wolcott Rd., Watertown, Mass.
MacDonald, Murray Jay ('23; '35) (WCDLMZ), Mgr., Natl. Casket Co., Inc., 29-76 Northern Blvd., Long Island City 1, N.Y.
MacDonald, Robert Francis (J'43) (SEKH), Mech. Engr., Northrop-Hendy Co., Sunnyvale; *for mail,* 434 Summit Dr., Redwood City, Calif.
Macdonald, Ronald Gordon (J'28) (GLZYCK), Secy.-Treas., Tech. Assn. of the Pulp & Paper Industry, 122 E. 42nd St., New York 17, N.Y.
MacDonnell, V. E. ('41), c/o Reynolds Metals Co., Box 191, Sheffield, Ala.
MacDougall, Ernest A. (J'40) (MBJ), Machinist, Phelps Dodge Corp., Bisbee; *for mail,* Box 1954, Warren, Ariz.
MacDougall, Irvine (J'37) (NCBH), Phelps Dodge Corp., Junction Ave., Lowell; *for mail,* 506 Vista, Warren, Ariz.
MacFadyen, Frank R. (J'43) (BJ), 1332 Logan Ave., N.W., Canton, Ohio.
MacFarlane, James ('08) (CM), Edifico Gomez Mena 805, Apartado 109, Havana, Cuba.
MacFarlane, Warren Courtland ('19) (CMDE), Pres., Gen. Mgr., Minneapolis-Moline Power Implement Co., Box 1050, Minneapolis 1, Minn.
MacGillis, Dan J. (J'32) (MCKHB), M.M., Long Mfg. Div., Borg-Warner Corp., 12501 DeQuindre St., Detroit 12; *for mail,* 16252 Cruse St., Detroit 27, Mich.
Macgillivray, Alfred Hanson (J'39) (YBC), Asst. Mech. Engr., Collyer Insulated Wire Co., 249 Roosevelt Ave., Pawtucket; *for mail,* 207 Doyle Ave., Providence, R.I.
MacGowan, George Fredericks (J'42) (EJR), Am. Loco. Co., Schenectady 5; *for mail,* 248 Union St., Schenectady, N.Y.
MacGowan, James F. ('41), Crawford Clothes, 34-02 Queens Blvd., Long Island City; *for mail,* 44 Hawkins Ave., Baldwin, L.I., N.Y.
MacGrath, Kenneth ('27; '35) (CAMNDH), Pres., Air Associates, Inc., Teterboro, N.J.
MacGregor, Charles Winters ('31; '41) (BJN RSY), Prof. Applied Mechanics, Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
MacGregor, David ('45) (SMBNCH), Ch. Engr., Edward Valves Inc., 1200 W. 145th St., East Chicago; *for mail,* 319 Beverly Pl., Munster, Ind.
MacGregor, Raymond Emile (J'39) (JMC), Engr., Harold W. Baker Co., 1030 Lancaster Ave., Bryn Mawr, Pa.; *for mail,* Box 1976, Trenton, N.J.
MacGregor, Robert McQueen ('44) (MCOY), Supt., Lobdell Co., P.O. Drawer 1108, Wilmington 99; *for mail,* 2307 W. 17th St., Wilmington 73, Del.
Macha, Edward Andrew (J'43) (BJNKMA), Design Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* Box 615, Wilmerding, Pa.
MacHenry Richard ('37; '46) (TYWL), Mech. Engr., Indus. Div., Textile Research Dept., Am. Viscose Corp., Marcus Hook, Pa.
Machold, Chas. Emmet ('01), Retired; 40 Hillcrest Ave., Philadelphia 18, Pa.
Macias, Carlos ('28; '35) (EFG), Engr., Gen. Mgr., Electromotor, S.A., Ave I, La Católica 43; *for mail,* 3a Calle de Zarco 59, Mexico, D.F., Mex.
Macintyre, Mackenzie ('43) (MCD), Pres., Dir., Alba Mfg. Co., Box 189; *for mail,* 74 S. View St., Aurora, Ill.
MacIntyre, Victor S. (J'40), Lt. (j.g.), U.S.N.R., Box 41, Naval Mine Depot, Yorktown, Va.
Mack, Albert J. ('22; '35) (EFS), Prof. Mech. Engrg., Kan. State College, Manhattan, Kan.
Mack, Harold Dwight, Jr. (J'45) (AKS), Eagle Picher Sales Co., 420 Lexington Ave., New York, N.Y.
Mack, John J., Jr. ('26; '32; '35) (FS), Design Supt., Tide Water Associated Oil Co., E. 22nd St., Bayonne; *for mail,* 111 Makatom Dr., Cranford, N.J.
Mack, Thomas Robert ('45) (BKLDT), Devel. Engr., Titanium Div., Natl. Lead Co., Sayreville; *for mail,* 96 MacArthur, Fords, N.J.
Mackal, Henry H. (J'33) (ACHBLJ), Pres., Halkey Roberts Co., E-37 Spring Valley Ave., Paramus, N.J.
MacKamey, Rhodus ('30; '36) (CKHDYB), Sr., Engr., Shell Oil Co., Inc., Box 711; *for mail,* Box 323, Martinez, Calif.
Mackay, Geo. W. ('26; '34; '35) (CJM), Am. Type Founders, Inc., 200 Elmora Ave., Elizabeth; *for mail,* 8 Park Terrace, Montclair, N.J.
MacKay, James (J'40), 3051 Warren Rd., Cleveland 11, Ohio.
Mackay, Ralph Bawden (J'43) (HAKS), Army Trng. Gp., A.P.O. 300, San Francisco, Calif.; *for mail,* 2177 W. 47th S., Murray, Utah.
MacKay, Simon ('17), V.P., Charge of Mfg. & Maint., Union Twist Drill Co.; *for mail,* 387 School St., Athol, Mass.
Mackay, Thomas R. (J'35) (ACH), Engrg. Supvr., Fairbanks, Morse & Co., 206 Commercial St.; *for mail,* 345 N. Towne Ave., Pomona, Calif.
MacKendrick, John N. ('38), Ch. Engr., Clark Bros. Co.; *for mail,* 305 Madison Ave., Olean, N.Y.
MacKenzie, Frank Cook (J'36) (WHCD), Engrg. Dept., Can. Paper Co.; *for mail,* Box 291, Windsor Mills, Que., Can.
Mackenzie, Henry Arnold ('29) (FERLA), Pres., Mackenzie Coal Co., 1056 Hanna Bldg., Cleveland 15, Ohio.
MacKenzie, John Alexander ('31) (CLSDH), Ch. Project Engr., Kaiser Engrs., Inc., 1924 Broadway, Oakland 12, Calif.
MacKenzie, Joseph Butz ('44) (RFNBJK), Mech. Engr., Stand. Stoker Co., Inc., Erie, Pa.
Mackenzie, Joseph Wesley ('22; '26) (SFH), Mech. Supvr., Consumers Power Co., 212 W. Michigan Ave., Jackson, Mich.
Mackenzie, Kenneth Gerard (A'17) (F), Asst. to V.P., Ch. Technologist, Tex. Co., 135 E. 42nd St., New York 17, N.Y.
Mackenzie, Sidney Thompson ('38) (FSRKJC), Sales Engr., Babcock & Wilcox Co., 1120 Packard Bldg., Philadelphia 2, Pa.

Mackie, Daniel Maries ('41) (BDHJN), Tool & Methods Engr., Chain Belt Co., 1600 W. Bruce St., Milwaukee 4; *for mail,* 6009 W. Wells St., Wauwatosa 13, Wis.
Mackie, John W., Jr. (J'39), Lt., Reception Center, Jefferson Barracks, Mo.
Mackintosh, D. ('29; '35), Chicago Nipple Mfg. Co., 1966 Southport Ave., Chicago 14; *for mail,* 1416 W. Winnemac Ave., Chicago 40, Ill.
Macklin, Leo (J'42) (KHE), 5701—15th Ave., Brooklyn 19, N.Y.
Macklin, Ralph W. ('38) (BJM), M.M., Wickwire Spencer Steel Co., 1 New Bond St., Worcester; *for mail,* 87 Whitmarsh Ave., Worcester 6, Mass.
MacLaren, Jas. E. ('30; '35), Branch Mgr., Buck & Hickman Ltd., 80 Whittall St., Birmingham, England.
MacLaren, Malcolm N. ('99), Retired; 4034 Ventura Canyon Ave., Van Nuys, Calif.
Maclay, William Richard (J'42) (NYHBEZ), Design Engr., Ingersoll-Rand Co., Athens, Pa.
MacLean, J. A. (J'46) (ABM), Bendix Prod. Div., Bendix Aviation Corp., South Bend 20, Ind.
MacLean, James Herbert (J'41) (EFNSMH), Design Engr., Ottawa Car & Aircraft Ltd., Slater St., Ottawa; *for mail,* 111 Wallace Ave., Toronto, Ont., Can.
Maclean, Wallace Emerson (J'39) (LDES), Ensign, U.S.N.R., Box 5482, State College Sta., Raleigh, N.C.
MacLehose, Maitland ('27; '35) (FSEKR), Sales Engr., Berwind-White Coal Min. Co., 1 Broadway, New York 4, N.Y.; *for mail,* 233 Mountain Ave., Summit, N.J.
MacLeod, Alan Stewart (J'29) (YTNHCD), Project Engr., U.S. Rubber Co., Eagle St., Providence; *for mail,* 233 Bowen St., Providence 6, R.I.
MacLeod, Charles Knox (J'46), Ensign, Civil Engr. Corps, U.S.N.R., 35th Naval Constr. Bn., Manila, P.I., F.P.O., San Francisco, Calif.
MacLeod, Daniel Tate ('01; '07) (CM), Propr., D. T. MacLeod Agency, 735 N. Water St.; *for mail,* 2138 E. LaFayette Pl., Milwaukee, Wis.
MacLeod, Lester Rounds ('30) (KFDLE), Office Engr., Anaconda Copper Min. Co., 25 Broadway, New York 4; *for mail,* 18 King St., Ardsley, N.Y.
MacLeod, Norman Douglas ('34) (NCMJ), Pres., Gen. Mgr., Abrasive Mch. Tool Co., Dexter Rd., East Providence 14, R.I.
Macmann, Edward Norman (J'41) (JKS), Pvt., 2307 Kimbal Court, Akron 14, Ohio.
MacMaster, Ronald Kellogg ('43) (C), Indus. Mech. Engr., Coverdale & Colpitts, 120 Wall St., New York, N.Y.; *for mail,* 30 Kempshall Pl., Elizabeth 3, N.J.
MacMillan, George Duncan (J'44) (NKRLJS), Maint. Engr., Carbide & Carbon Chemicals Corp., South Charleston; *for mail,* 425 Rosemont Ave., South Charleston 3, W.Va.
MacMillin, Howard Francis ('39) (CNH), Arthur D. Little, Inc., 30 Memorial Dr., Cambridge 42; *for mail,* 16 Garden St., Cambridge 38, Mass.
MacMurphy, William Clancy, Jr. (J'43), 176 Broad St., Charleston, S.C.
Macnab, Archibald Cameron ('45) (CJNOST), Mech. Engr., Dominion Woollens & Worsteds Ltd.; *for mail,* Church St., Hespeler, Ont., Can.
MacNair, Donald John (J'44) (CHS), 159 Maplewood Ave., Philadelphia 44, Pa.
MacNamara, M. J. ('21; '35), "Clifden," North St. & Highland Rd., Rye, N.Y.
MacNamee, William R. ('41) (CHJ), Engr., Baldwin Loco. Works, Paschall P.O., Philadelphia 42; *for mail,* 58 S. Clifton Ave., Clifton Heights, Pa.
MacNaughton, Edgar ('15; '21) (CMSNFE), Prof. Mech. Engrg., Engrg. Sch., Tufts College, Medford 55; *for mail,* 61 Ravine Rd., West Medford 55, Mass.
MacNaughton, Ernest John (J'45) (BR), Mich. Div., N.Y. Cent. R.R., Detroit; *for mail,* 658 Alter Rd., Detroit 15, Mich.
Macneale, Neil ('35; '45) (NBDMZJ), Mech. Engr., Procter & Gamble Mfg. Co., Ivorydale; *for mail,* 3421 Paxton Rd., Cincinnati 8, Ohio.
Macnee, Charles Maitland ('26) (FST), Ch. Engr., Patons & Baldwins Ltd., Alloa; *for mail,* Hardriggs, Clackmannan, Scotland.
MacNeill, John Harmon (J'42) (BHKYZ), 8 Alida Court, Oakland 2, Calif.
MacNeille, Martin Benjamin ('21) (ACEHMR), Mgr., Pump Div., A. Y. McDonald Mfg. Co., 12th & Pine Sts.; *for mail,* 1095 Mt. Pleasant St., Dubuque, Iowa.
Macomber, Henry E. ('37) (FKS), Engr., Prod. Dept., Detroit Edison Co., 2000—2nd Ave., Detroit, Mich.
Macomber, James Keith ('22; '30; '35) (NZAC), Head, Design Engrg. Dept., Arma Corp., 254—36th St., Brooklyn 32; *for mail,* 85 Andover Rd., Rockville Centre, L.I., N.Y.
Macomber, Mark Morris (J'45), Apprentice Seaman, U.S.N.; 1690 N. Garfield Ave., Pasadena 6, Calif.
Macomber, Thomas W. ('37; '45) (BKNAH), Devel. Engr., Joshua Hendy Iron Works, Sunnyvale; *for mail,* 327 Emerson St., Palo Alto, Calif.
MacPherson, David Avery ('36; '44) (ERCNAS), Mech. Engr., Design, Electro-Motive Div., Gen. Motors Corp., La Grange; *for mail,* 156 N. Oak Park. Ave., Oak Park, Ill.
MacPherson, John William (J'44) (SFOEKA), Sales Engr., C. C. Moore & Co., Inc., 555 S. Flower St., Los Angeles 13; *for mail,* 1444 El Miradero Ave., Glendale 1, Calif.
MacVannel, Duncan Pine (J'45) (COA), Engr. in Trng., Otis-Fensom Elev. Co. Ltd., Hamilton, Ont., Can.
Macwatty, Frank L. (J'45), 345 Mountain Ave., Ridgewood, N.J.
MacWilliam, Edwin Warren (J'42) (F), Lt. (j.g.), U.S.N.R., Skiff St., North Haven, Conn.
Macy, Ralph Gerard ('16; '19; F'44), 7 Goodron St., East Norwalk, Conn.
Madden, Dean Ennis (J'43) (CZMH), Secy., Treas., A. W. Cash Valve Mfg. Corp., 666 E. Wabash Ave., Decatur 80, Ill.
Maddison, Robert James (J'34) (CJDY), Co. Rep., Whitehead Bros. Co., 537 W. 27th St., New York 1, N.Y.; *for mail,* 53 Arlington Ave., Caldwell, N.J.
Maddock, Bill (J'45) (CHOO), Asst. Editor, Maujer Publ. Co., St. Joseph; *for mail,* 815 Pipestone St., Benton Harbor, Mich.
Maddock, John Thomas (J'37) (NCMA), Head, Gear Engrg. Dept., Joshua Hendy Iron Works, Sunnyvale; *for mail,* Box 900, Route 2, Los Altos, Calif.
Maddox, Adolph Walter ('46), Capt., U.S.S. *Europa*, AP-177, F.P.O., New York, N.Y.; *for mail,* Apt. 305, 2615—42nd St. N.W., Washington 7, D.C.
Madea, Frederick John (J'42) (BCSMKH), Mech. Engr., Gibbs & Cox, Inc., 21 West St., New York 6, N.Y.; *for mail,* 208 Walthery Ave., Ridgewood, N.J.
Madeheim, Huxley ('27; '33; '35) (CEK), Cons. Mech. & Elec. Engr., 235 E. 42nd St., New York 17, N.Y.
Madey, Eugene (J'43) (FKS), Layout Man, Westinghouse Elec. Corp., 2519 Wilkens Ave., Baltimore; *for mail,* Clifmar Rd., Randallstown, Md.
Madgett, Hugh Herbert (J'42) (ZSHF), Sales & Serv. Engr., Bailey Meter Co. Ltd., 980 St. Antoine St., Montreal 3; *for mail,* Apt. 17, 5551 Queen Mary Rd., Montreal 29, Que., Can.
Madison, Richard Durham ('18; '35) (KSHBAF), Research Engr., Buffalo Forge Co., 490 Broadway, Buffalo 5, N.Y.
Madsen, Dean James (J'40) (ABC), Test Engr., Northrop Aircraft Inc., Hawthorne; *for mail,* 640-A W. Queen St., Inglewood, Calif.
Madsen, Herman P. (J'40) (CNLWD), Engr., E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington 98, Del.
Maertin, Harvey Arthur, Jr. (J'41) (CNBMHJ), Asst. Supt., Taylor Forge & Pipe Works, Box 485, Chicago; *for mail,* 6320 S. Kenneth Ave., Chicago 29, Ill.
Magalhaes, Willard S. (J'33) (CFS), Maj., Army Air Forces, The Pentagon, Washington 25; *for mail,* 3605—20th St., N.E., Washington 18, D.C.
Magdeburger, Edward C. ('17; '20) (ESFB), Head Engr., Tech. Adviser, Diesel Engs., Bur. of Ships, Code 643, Navy Dept., 17th & Constitution Ave., Washington 25; *for mail,* 1612 Concord Ave., Washington 11, D.C.
Magee, Frederick M. (J'39) (DMOCTY), Ch. Engr., Bellamy-Robie, Inc., 45 Main St., Cambridge; *for mail,* 6 Daniels St., Lexington 73, Mass.
Magee, George Hugh ('21; '35) (FLBK), Gas Engr., L. I. Ltg. Co., 250 Old Country Rd., Mineola; *for mail,* 91 Poplar St., Garden City, N.Y.
Magill, Franklin Robert ('17; '23) (J), Owner, F. R. Magill Co., 1022 Empire Bldg., Pittsburgh 22, Pa.
Maglathlin, Sidney A. ('21; '35), Supt., Engrg. Dept., New England Safety Div., London Guarantee & Accident Co. Ltd., 141 Milk St., Boston 9; *for mail,* 16 Howard St., West Bridgewater, Mass.
Magos, John Peter ('38) (BCSKJ), Dir. of Research, Crane Co., 836 S. Michigan Ave., Chicago 5; *for mail,* 2142 Chestnut Ave., Wilmette, Ill.
Magrath, Howard A. (J'33) (ABNO), Aero. Engr., Army Air Forces, Air Tech. Serv. Comd., TSEAC-5, Wright Field, Dayton; *for mail,* 550 Forest Ave., Dayton 5, Ohio.
Magraw, Lester Andrew ('20) (CMSFKZ), Pres., Dir., Cent. Ill. Pub. Serv. Co., 1228 Ill. Bldg., Springfield, Ill.
Maguire, Edward L. (J'38) (BJM), Mech. Engr., Charge of Estimate & Design Dept., Alloy Fabricators, Inc., 500 Market St., Perth Amboy; *for mail,* Orchard Terrace, Clark Township, N.J.
Maguire, James (J'39) (CDL), Engrg. Dept., Wyeth, Inc., Mason, Ingham Co., Mich.
Maguire, James Henry ('22) (CMJ), Retired; 1739 W. Taylor St., Kokomo, Ind.
Maguire, Jeremiah D. ('11), Pres., Federation Bank & Trust Co., 461—8th Ave., New York, N.Y.
Maguire, Norman Lunny (J'44) (ACMSFE), Aero. Engr., Research Dept., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 101 Hope St., Stamford, Conn.
Magura, Milton John (J'41) (KBFS), Assoc. Mar. Engr., U.S. Maritime Comm., 14th & Franklin Sts., Oakland, Calif.
Maguth, Frank J. (J'42), M. J. Maguth & Son, 508 Jefferson St., Carlstadt, N.J.
Magyar, Ernest Alexander (J'40) (EAFKOC), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., Main St., East Hartford; *for mail,* 46 Congress St., Hartford, Conn.
Maha, Otto Joseph ('44) (NHBJC), V.P., Engrg., Hannifin Mfg. Co., 621 S. Kolmar Ave., Chicago 24, Ill.
Mahaffy, Hugh Morris (J'43) (AMNYOC), Sr. Draftsman, Glenn L. Martin Co., Baltimore, Md.; *for mail,* 1700 Riverview Ave., Wilmington, Del.
Mahaffy, Reid Alexander (J'41) (NJDML), Packaging Engrg. Dept., Union Bag & Paper Corp., 233 Broadway, New York; *for mail,* 209 Clinton Ave., Brooklyn 5, N.Y.
Mahan, Frank Titus, Jr. (J'44) (AMJ), Engr., Aeronca Aircraft Corp., Mun. Airport; *for mail,* 411 Shafor Ave., Middletown, Ohio.
Mahen, Kenneth W. (J'41) (HJL), c/o Hercules Powder Co., Parlin, N.J.
Maher, Robert Lee (J'40) (KLS), Engr., Chase Brass & Copper Co., Waterbury, Conn.; *for mail,* 1418 W. State St., Trenton 8, N.J.
Mahin, Robert Leedy (J'44), Engrg. Dept. Jr. Officer, U.S.N.R., U.S.S. *Shangri-La*, CV-38 F.P.O., San Francisco; *for mail,* Box 137, Route 1, Arvin, Calif.
Mahlab, Salim Saleh (J'42) (DKP), Staff Sgt., Hdq. XXII Corps, G-2 Sec., Ft. DuPont, Del.; *for mail,* 66-01 Saunders St., Forest Hills, L.I., N.Y.
Mahle, Herbert Niedringhaus (J'40) (MJCO), Prod. Engr., Mahle Auto Body Co. Inc., 370 E. 8th St., St. Paul 1, Minn.
Mahon, Buford Milburn (J'37) (HBNJML), Devel. & Design Engr., Simmons Mfg. Co., Ohio St.; *for mail,* 411 E. 4th St., Ashland, Ohio.
Mahon, William Joseph ('24; '34; '35) (CM), Gen. Mgr., Johnson & Johnson de Mex., Apartado Postal 1133, Mexico, D.F., Mex.
Maienknecht, Gaile Edward (J'44) (NKJ), Research Engr., Linde Air Products Co., Tonawanda; *for mail,* 102 Fowler Pl., Kenmore 17, N.Y.
Maier, August R. ('30; '35) (BHECMY), Ch. Engr., Oil Well Supply Co., Colbert Ave., Oil City, Pa.
Maier, Hanns John (J'45), U.S.Navy; 513 Front St., Hempstead, N.Y.
Maier, Harry J. ('45) (BJCSAH), Engrg. Dept., Lunkenheimer Co., Beekman St. & Waverly Ave., Cincinnati 14; *for mail,* 3740 Marydell Pl., Cheviot, Cincinnati 11, Ohio.
Maier, Harry Ludwig, Jr. (J'33) (AKMNTY), Devel. & Design Engr., Natl. Vulcanized Fibre Co., Box 311, Wilmington, Del.
Maier, Michael Walter (J'45), Test Engr., Shock & Vibration Div., Naval Research Lab., Washington; *for mail,* 161—35th St., N.E., Washington; 19, D.C.
Maiersperger, Walter P. (J'39), Maj., Hdq., Air Tech. Serv. Command, Box 1041, Area A, Wright Field, Dayton, Ohio.
Maillard, Albert L. ('25) (CSFKOE), Managing Dir., Elec. League of Indianapolis, Inc., K. of P. Bldg., Indianapolis, Ind.
Mailler, John Preston ('19; '35) (SEHFCD), Ch. Engr., N.J. Power & Light Co., McFarlan St., Dover, N.J.
Main, James M. ('25; '35) (BDHKLS), Mech. Engr., H. K. Ferguson Co., 1650 Hanna Bldg., Cleveland 15; *for mail,* 2347 Valleyview Dr., Rocky River 16, Ohio.
Maine, William Charles (J'36) (FLSTZ), Power Engr., Mohawk Carpet Mills Inc., 57 Lyon St.; *for mail,* 2 McGibbon Ave., Amsterdam, N.Y.
Mainhardt, Robert (J'43) (JYDNA), Engr., U.S. Engrs., Los Alamos; *for mail,* Box 1663, Santa Fe, New Mex.
Mains, William D. (J'38) (KLEY), Ch. Engr., J. C. Mahoney, Inc., 21 West St., New York 6, N.Y.
Mair, Kenneth T. ('30; '35) (CDM), Engr., Am. Terry Derrick Co., South Kearny; *for mail,* 419 E. 7th St., Plainfield, N.J.
Maissian, Edward Daniel (J'38) (HSKA), Lt. Comdr., U.S.N.R., Hull Planning Officer, Naval Shipyard, Terminal Island, San Pedro, Calif.
Majcher, Walter Stanislaus (J'44) (ONEMAJ), Lt. (j.g.), U.S.N., Spec. Projects Officer, U.S.S. *Greyhound*, Pudget Sound Naval Shipyard, Bremerton, Wash.; *for mail,* 1548 S. 5th St., Milwaukee 4, Wis.
Majercik, Anthony S. (J'36) (LBD), Devel. & Mfg. Engr., Teletype Corp., 1400 W. Wrightwood Ave., Chicago 14; *for mail,* 4728½ N. Beacon St., Chicago 40, Ill.
Majneri, Ludwig Anton ('41) (ABHFKC), V.P., Charge of Engrg., Warner Aircraft Corp., 20263 Hoover Ave., Detroit 5, Mich.
Major, Alfred Robert (J'38), Fla. Pulp & Paper Co., Cantonment, Fla.
Major, William S. ('40) (FSEK), Sales Mgr., Power Plant Div., Am. Engrg. Co., Aramingo St., Philadelphia 25; *for mail,* 1415 Edgevale Rd., Philadelphia 31, Pa.
Majors, Harry, Jr. (J'39) (BAHKNY), Instr. in Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Makasiar, Vincente Valencia ('40) (HBC), Owner, Firm of Vincente V. Makasiar, 205 N. Greene St.; *for mail,* Box 2289, 1004 Jefferson Stand. Bldg., Greensboro, N.C.

Maker, Frank L. ('30) (BK), Designing Engr., Refinery Plants, Stand. Oil Co. of Calif., 225 Bush St., San Francisco; *for mail,* 950 Galindo St., Concord, Calif.

Makhno, Vladimir Victor ('44) (BCKMNO), Mech. Engr., Commonwealth of Australia, 82 Pitt St.; *for mail,* 348 Bronte Rd., Waverly, Sydney, N.S.W., Australia.

Maki, Eino Evert (J'41 (ACMNYB), Mch. Designer, The Celotex Corp., 205 W. Monroe St., Chicago 6; *for mail,* 527 Helmholz Ave., Waukegan, Ill.

Malakoff, Norman H. (J'40) (CKS), 2225½ Linden Ave., Baltimore 17, Md.

Malamut, Martin (J'42) (HSJ), Mech. Engr., Bur. of Ships, Navy Dept., Washington; *for mail,* 247 Delaware Ave., S.W., Washington 4, D.C.

Malany, Herbert Smith ('44) (CSFMK), Steam Serv. Supvr., Westinghouse Elec. Corp., 307 W. 3rd St.; *for mail,* 3855 Hyde Park Ave., Cincinnati 2, Ohio.

Malcolm, Jos. F. (J'37), Chapman Valve Mfg. Co.; *for mail,* 151 Oak St., Indian Orchard, Mass.

Maldari, Carmine Daniel (J'40) (CDM), Lt., Army Air Forces, 120th Base Unit, Sec. K-1, Army Air Base, Richmond, Va.

Male, Reginald LeFleming (J'44) (BHK), Mech. Engr., Gen. Elec. Co., 5 Lawrence St., Bloomfield; *for mail,* Waxberg Lane, Pine Brook, N.J.

Malecki, Myron Benedict (J'43) (CABZ), 4000 E. 74th St., Cleveland 5, Ohio.

Maledy, J. Edward (J'39) (HBE), Maint. Engr., Stand. Oil Co. of Calif., Binxx; *for mail,* 615 D St., Taft, Calif.

Maleev, Vladimir Leonidas ('21) (ENAB), Cons., Research Engr., Harvey Mch. Co., Inc., 6200 Avalon Blvd., Los Angeles 3; *for mail,* 1232 N. Orange Dr., Hollywood 38, Calif.

Malin, Chester George ('40) (MHCO), Cons. Engr., 447 Riverdale Dr., Glendale 4, Calif.

Malischke, Carl (J'42) (CJMN), Pres., Treas., Charles Malischke Co., 2518 N. 30th St., Milwaukee 10; *for mail,* 6466 Upper Pkwy. N., Wauwatosa 13, Wis.

Mallay, Paul David ('45) (RJMC), Mgr., R.R. Div., Lukens Steel Co., Coatesville, Pa.

Maller, Melvin A. (J'41) (NMBG), Mech. Engr., Watson Labs., Army Air Forces, Red Bank; *for mail,* 70 N. Bath Ave., Long Branch, N.J.

Mallick, Randolph William ('43) (DCMO), Sec. Engr., Hdq. Plant, Layout & Matls. Handling Sec., Westinghouse Elec. Corp.; *for mail,* 325 North Ave., East Pittsburgh, Pa.

Mallon, Charles Joseph ('44) (CEFHOS), Mech. Engr., Design, Dept. of Pub. Works, City of Philadelphia, 223 Chestnut St. Pier, Philadelphia 6; *for mail,* 1224 Harrison St., Philadelphia 24, Pa.

Mallon, David J. (J'46) (SKJL), Steam Power Div., Worthington Pump. & Mchy Corp., Harrison; *for mail,* 171 Sterling Dr., Orange, N.J.

Mallon, Frederick J. (J'36), (ELS), Ch. Engr., Fairmont Creamery Co.; *for mail,* 507 A Ave., Lawton, Okla.

Mallory, Burton Cowles ('23; '34) (SFEKD), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.

Mallory, Henry Reed ('33) (CT), Exec. V.P., Treas., Cheney Bros.; Pres., Treas., Pioneer Parachute Co.; *for mail,* 68 Prospect St., Manchester, Conn.

Mallory, Walter Franklin ('18; '30) (EFJRSA), Prof. Mech. Engrg., Univ. of Colo.; *for mail,* 915—15th St., Boulder, Colo.

Malloy, John F. ('24; '33) (OHYSNL), Mech. & Struc. Engr., Carbide & Carbon Chemicals Corp., South Charleston; *for mail,* 3714 Staunton Ave., Charleston 4, W.Va.

Mally, George Gregory (J'42) (NBCMJL), Mech. Engr., Bundy Tubing Co., 10951 Hern Ave., Detroit 13; *for mail,* 19316 Teppert St., Detroit 5, Mich.

Malmberg, Philip Otto (J'39) (HBS), Hyd. & Mech. Engr., Tampa Shipbldg. Co., Inc., Tampa; *for mail,* 8109—36th Ave. N., Box 128, Route 2, St. Petersburg 6, Fla.

Malone, James L. (J'44) (CJS), Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13, Pa.

Malone, John F. ('37) (CFKS), Asst. Mgr., Dist. Steam Dept., Consltd. Gas, Elec. Light & Power Co. of Baltimore, 503 Lexington Bldg., Baltimore 3, Md.

Maloney, James Dodson, Jr. (J'43) (AFM), Route 14, Lyons Bend Rd., Knoxville, Tenn.

Maloney, Michael Joseph ('21; '35) (CDFKNR), Supt., Poor & Co., Burden Ave., Troy, N.Y.

Malzahn, Gus Edwin (J'45) (BJM), Engr., Charlie's Mch. Works, Inc., Box 66; *for mail,* 1103 N. 6th, Perry, Okla.

Manas, Vincent T. ('44) (FHO), Chief, Mech. & Elec. Engrg. Sec., Fed. Pub. Housing Authority, Longfellow Bldg., Washington 25; *for mail,* 926 Massachusetts Ave., N.W., Washington 1, D.C.

Mandel, Lawrence (J'44) (CJM), Exec. Engr., Masell Mfg. Corp., 195 Front St., Brooklyn, N.Y.

Manderfield, Robert Earl (J'46) (BHM), Project Engr., Fisher Body Div., Gen. Motors Corp., 465 Milwaukee Ave., Detroit; *for mail,* 8717 Morley, Detroit 4, Mich.

Mangan, John R., Jr. (J'42) (AC), Engr., Goodyear Aircraft Corp., Akron; *for mail,* c/o John Ritzman, Box 145, Ellet, Ohio.

Mangan, William Edward (J'42) (CHL), Tech. Asst., Carbide & Carbon Chemicals Corp., South Charleston; *for mail,* 1632 Quarrier St., Charleston 1, W.Va.

Manger, Clarence Philip (J'39), Capt., 604th Bomb Sqd., March Field, Calif.

Manger, Paul A. (J'21) (NBJYHS), Ch. Engr., Buffalo Plant, Farrel-Birmingham Co. Inc., 344 Vulcan St., Buffalo 7; *for mail,* 117 South Dr., Eggertsville 21, N.Y.

Mangler, William Eugene (J'44) (BLN), Mch. & Tool Designer, Bridge Tool & Die Works, 106 S. 16th St., Philadelphia 3; *for mail,* 1201 E. Shelmire Ave., Philadelphia 11, Pa.

Maniates, Peter George (J'41) (ABMCLZ), Bowman Dairy Co., 140 W. Ontario St., Chicago; *for mail,* 4737 W. North Ave., Chicago 39, Ill.

Manierre, George ('17) (D), Propr., Manierre Engrg. & Mchy. Co., 622 Colby Abbot Bldg., Milwaukee, Wis.

Manifold, George O. (J'38) (CMDNOG), Ch. Engr., Universal Lub. Systems, Inc., 743 Allegheny Ave., Oakmont, Pa.

Manilych, Sam (J'43) (MJCD), Mech. Engr., N.Y. Naval Shipyard, Brooklyn 1; *for mail,* 160 Hewes St., Brooklyn 11, N.Y.

Manjoine, Michael Joseph (J'37) (BJNZOA), Research Engr., Westinghouse Research Labs., Ardmore Blvd., East Pittsburgh; *for mail,* 1000 Braddock Rd., Pittsburgh 21, Pa.

Manke, Gustave Leonard (J'41) (ZTMB), Ch. Draftsman, Veeder-Root, Inc., 20 Sargeant St., Hartford 2, Conn.

Mankiewicz, Victor John (J'40) (ANJBG), Stress Analyst, Lockheed Aircraft Corp., Burbank; *for mail,* 1424 S. Mariposa Ave., Los Angeles 6, Calif.

Mankin, Guy ('30) (CFS), 1404—1st Natl. Bank Bldg., Atlanta 3, Ga.

Mankus, Roman Theodore (J'43), Lt. (j.g.), U.S.N.R., Aviation Engr., Specialist, Assembly & Repair Dept., Naval Air Sta., Seattle 5, Wash.; *for mail,* 6030 S. Rockwell, Chicago 29, Ill.

Manley, Humes Baker ('45) (LKZCD), Project Engr., J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6; *for mail,* 1008 E. 84th Terrace, Kansas City 5, Mo.

Manley, Robert F. (J'37), Field Engr., Bryant Chucking Grinder Co., Springfield, Vt.; *for mail,* 3798 W. 136th St., Cleveland 11, Ohio.

Manley, Robert William (J'42) (CBMYFN), Pvt., 1st Class, Div. of New Devels., U.S.A., Antiaircraft Serv. Test Sec., Army Ground Forces, Board 1, Ft. Bliss, Tex.; *for mail,* 57-22—81st St., Elmhurst, N.Y.

Manley, Sumner M. ('07; '17), Cons. Engr., Procter & Gamble Co., Ivorydale, Cincinnati, Ohio.

Mann, Carl P. ('16; '23), Secy., Treas., A. Kristoferson, Inc., 1300 Rainier Ave., Seattle, Wash.; *for mail,* 403 Main St., Riverton, N.J.

Mann, Frederick L., Jr. (J'45) (ABFK), Test Engr., Ranger Aircraft Engines Div., Fairchild Eng. & Airplane Corp., 712 Fulton St., Farmingdale, L.I., N.Y.

Mann, George Lee (J'42) (BACKSM), Stress Analyst, Douglas Aircraft Co., Inc., 5651 W. Manchester St., Los Angeles; *for mail,* 225-D San Vicente, Santa Monica, Calif.

Mann, Harvey Blaine ('11; '18) (DESZ), Owner, Mann Engrg. Co., 429 Pennsylvania Ave.; *for mail,* 7300 Brighton Rd., Ben Avon, Pittsburgh 2, Pa.

Mann, Isham W., Jr. (J'39) (JLP), 728 State St., New Orleans 15, La.

Mann, John ('30) (HJBY), Ch. Engr., Goulds Pumps, Inc., 240 Fall St., Seneca Falls, N.Y.

Mann, John J. (J'45) (KOS), Designing & Supvg. Engr., Jaros, Baum & Bolles, 415 Lexington Ave., New York; *for mail,* 1421 Mamaroneck Ave., Mamaroneck, N.Y.

Mann, John Winfield (J'36) (ECFKZH), Gen. Oper. Foreman, Socony-Vacuum Oil Co., Inc., 3821 Indianapolis Blvd., East Chicago, Ind.

Mann, Robert M. (J'36) (AOM), Gp. Engr., Glenn L. Martin-Neb. Co., Omaha; *for mail,* 426 Ave. A, Plattsmouth, Neb.

Mannes, Robert Lynn (J'43) (ABHKY), Chief Specialist (X), N. Div., Naval Air Sta., Moffett Field, Calif.

Manney, Charles Joseph ('34; '41) (DJB), Mech. Engr., Columbus-McKinnon Chain Corp., Box 72, Tonawanda, N.Y.

Manning, Edward Vincent (J'39) (AZOJSF), Supvr., Altitude Test Lab., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 640 Albert Pl., Ridgewood, N.J.

Manning, Ralph Melville (J'42) (ZNC), Factory Engr., West Lynn Works, Gen. Elec. Co., 40 Federal St., West Lynn; *for mail,* 27 Winthrop St., Winchester, Mass.

Manning, Thomas Joseph, Jr. (J'43), Ensign, Naval Air Exper. Sta., Naval Shipyard, Philadelphia, Pa.

Manning, William Thomas ('35) (SBJEKO), Sec. Engr., Indus. Turbine Sec., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* 1104 Lindale Ave., Drexel Hill, Pa.

Manny, Erwin Harry (J'39) (CSZEFK), Ch. Serv. Engr., Manning, Maxwell & Moore, Inc., 450 Communipaw Ave., Jersey City 4, N.J.

Mans, Frederick J. ('40) (CL), Major, c/o B.P.M., Willemslaan 2, Batavia-Centrum, N.E.I.

Mansfield, Ernest B. ('36; '44) (YBLR), Sr. Engr., Truck & Bus Tire Devel., Firestone Tire & Rubber Co., Akron; *for mail,* 127 Birdwood Rd., Akron 13, Ohio.

Mansfield, Howard Vaughn ('44) (FRSCK), Ch. Engr., So. Coal Co. Inc., 81 Madison Bldg., Memphis 1; *for mail,* 19 Mosby Rd., Whitehaven, Tenn.

Mansfield, Judson H. ('27) (BMW), Ch. Engr., Greenlee Bros. & Co., 12th St., Rockford, Ill.

Mansfield, William Muir (J'41) (SFKEC), Asst. Ch. Engr., Power Plant, Nev. Mines Div., Kennecott Copper Corp.; *for mail,* Box 1384, McGill, Nev.

Mansky, Clarence Bernard (J'44) (BHNA), Asst. Prof. Mech. Engrg., Cornell Univ., Ithaca, N.Y.

Mantey, Louis Henry (J'41) (NBMZJC), Designer, Research & Devel. Div., Addressograph-Multigraph Corp., 1200 Babbitt Rd., Cleveland 17; *for mail,* 70 Glenwood Ave., Willoughby, Ohio.

Manthei, Edward Charles (J'38) (OKS), Mech. Engr., Univ. of Ill., Urbana; *for mail,* 605 S. Edwin St., Champaign, Ill.

Manuel, Harold Eugene (J'38) (MLDN), Mech. Engr., Buick, Oldsmobile, Pontiac Assembly Div., Gen. Motors Corp., Fairfax & Kindleberger Sts., Kansas City, Kan.; *for mail,* 2834 Askew, Kansas City 3, Mo.

Manuele, Joseph, Jr. (A'42) (CDLMOY), Dir., Quality Control, Westinghouse Elec. Corp., East Pittsburgh; *for mail,* Box 169, R.D. 4, Irwin, Pa.

Manz, Louis C. ('13) (ABFM), Inspr., Engrg. Matl., Philadelphia Ord. Dist., Philadelphia; *for mail,* 3322 Friendship St., Philadelphia 24, Pa.

Mao, Easin ('46), Supt. of Shops, Chinese Natl. Rys. Ministry of Communications, Chungking, China; *for mail,* c/o Chinese Supply Comm., 2224 R St., N.W., Washington 8, D.C.

Mapes, Joe Milton (J'39) (ACM), 311 Delaware St., Corry, Pa.

Marantette, Louis Edward (J'43) (ALM), U.S. Navy; Box 324, Columbia Falls, Mont.

Marazzini, Gilbert John (J'44) (DCL), Lt. (j.g.), U.S.N.R., Navy 926, F.P.O., San Francisco, Calif.; *for mail,* 500 W. 114th St., New York 25, N.Y.

Marcero, Aloysius ('23) (BFS), Plant Engr., Detroit Edison Co., 2000—2nd Ave., Detroit, Mich.

March, John Parrish (J'42), 1st Lt., Army Air Forces, Air Trans. Comd.; 7302 River Rd., Cincinnati 33, Ohio.

March, Perrin George, III (J'39) (CMN), Pres., Cincinnati Shaper Co., Hopple, Elam & Garrard Sts., Cincinnati 25; *for mail,* 7302 River Rd., Cincinnati 33, Ohio.

Marchant, John Henry ('36) (EAKBF), U.S. N.R., Bunker Hill Rd., Andover, Conn.

Marchant, Richard Douglas ('36; '45) (CBJ MND), Designing Engr., Capacitors, Gen. Elec. Co., 100 Woodlawn Ave.; *for mail,* 40 Buel St., Pittsfield, Mass.

Marco, Salvatore Michael (J'38) (NBKJZ), Assoc. Prof. Mech. Engrg., Ohio State Univ., Columbus 10, Ohio.

Marcotte, Robert Lee (J'43) (NMZD), Lt. (j.g.), U.S.N.R., Gunnery, Watch & Div. Officer, U.S.S. *Cortland,* APA-75, F.P.O., San Francisco, Calif.; *for mail,* 5100 E St., Lincoln 2, Neb.

Marder, Ira K. (J'40) (CMYJAD), Prod. Dept., Astor, Ramel Mfg. Co., 249 Brook Ave., Bronx; *for mail,* 245 Amherst St., Brooklyn 29, N.Y.

Marek, Clarence Thomas (J'43) (JKYM), Instr. in Engrg. Shop Practice, Purdue Univ.; *for mail,* 209 De Hart, West Lafayette, Ind.

Margrave, Wilton (J'37) (ACEMHN), Mech. Engr., Parker Appliance Co., 17325 Euclid Ave.; *for mail,* 3028 E. 132nd St., Cleveland 20, Ohio.

Marich, Frederick ('37; '45) (AJMNOR), Design Engr., Farrand Optical Co., 4401 Bronx Blvd., New York; *for mail,* 55 W. 55th St., New York 19, N.Y.

Marin, Joseph ('36) (BAOYJ), Prof. Engrg. Mechanics, Pa. State College, State College, Pa.

Marinelli, Gino J. (J'37) (SFLEHB), *Student Award,* '37; Engrg. Dept., Anaconda Copper Min. Co., 25 Broadway, New York 4, N.Y.; *for mail,* 25—2nd St., Dumont, N.J.

Marinoff, George (J'44) (GYJM), Instr., Dept. of Engrg. Drawing, Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland, Ohio.

Marion, Frank Ivichievich ('26; '34) (CESF), Lt., Fuel & Lub. Engr., Indus. Dept., U.S.N., Box 6093, Sta. A; *for mail,* 2601 Octavia St., New Orleans 15, La.

Mark, Clayton, III (J'38), Engr., Clayton Mark & Co., 1900 Dempster St., Evanston; *for mail,* Sheridan Rd., Wilmette, Ill.

Mark, Melvin (J'45) (NKBAJH), Instr. in Mch. Design, Inst. of Tech., Univ. of Minn., Minneapolis 14; *for mail,* 1484 Summit Ave., St. Paul 5, Minn.

Mark, Walter J. ('26) (DGWLSB), Mech. Engr., Natl. Housing Agency, Fed. Pub. Housing Authority, Washington, D.C.; *for mail,* 317 S. Exchange St., St. Paul 2, Minn.

Markell, John, Jr. (J'41) (AKOE), Devel. Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester; *for mail,* 714 Colwell Rd., Grace Park, Chester, Pa.

Marker, Roland Howard ('21; '38) (OSEFD), Partner, Froehlich & Emery Engrg. Co., 410 Toledo Trust Bldg., Toledo 4, Ohio.

Markey, Harold Irving ('14; '20; '35) (DEHN), East. Dist. Mgr., Diamond Chain & Mfg. Co., 1011 Chestnut St., Philadelphia 7, Pa.

Markfelder, Charles F. ('24; '31) (CDLMOY), Inspc. Indus. Surveys, Fidelity & Casualty Co. of N.Y., 80 Maiden Lane, New York 7; *for mail*, 8039—88th Rd., Woodhaven 21, L.I., N.Y.
Markham, Alfred Emile, Jr. (J'43) (CMO), Supvg. Engr., Defense Plant Corp., 302 Pacific Mutual Bldg., Los Angeles; *for mail*, 4955 Arcola Ave., North Hollywood, Calif.
Marks, Arthur Adolph, Jr. (J'41) (CADMZ), Inspr. of Ord. Matl., Twin Cities Regional Office, Chicago Ord. Dist., Army Serv. Forces, 609—2nd Ave. S., Minneapolis 2, Minn.; *for mail*, 744 Wrightwood, Chicago 14, Ill.
Marks, Gilbert Leonard (J'44) (CKZ), Student Engr., York Corp.; *for mail*, Y.M.C.A., York, Pa.
Marks, Harry J. ('07) (FS), 28 W. 85th St., New York 24, N.Y.
Marks, James H. ('40), 622 Rivard Blvd., Grosse Pointe Village, Mich.
Marks, Lionel S. ('97; '04; F'39) (BES), Prof. Emeritus, Harvard Univ., Pierce Hall, Cambridge, Mass.
Marks, Malcolm J. (J'41) (AFM), U.S. Naval Air Corps; *for mail*, 5626 Darlington Rd., Pittsburgh, Pa.
Markson, Alfred A. ('25; '40) (BKS), Mech. Engr., Hagan Corp., 323—4th Ave., Pittsburgh 30; *for mail*, 815 Parkside Ave., Pittsburgh 16, Pa.
Markson, Francis I. (A'45) (SKFZ), East. Mgr., Bird-Archer Co., 400 Madison Ave., New York 17, N.Y.
Marlatt, Charles Mills (J'46), Sales Engr., Firestone Indus. Products Co., Akron; *for mail*, 1621 Evergreen Ave., Akron 1, Ohio.
Marley, George E. (J'39) (CEFLS), Project Engr., Dravo Corp., 300 Pennsylvania Ave., Pittsburgh 22, Pa.
Marlow, Alfred Stanley, Jr. (J'41) (HMCL), Partner, Marlow Pumps, Ridgewood; *for mail*, Wiley Pl., Wyckoff, N.J.
Marmont, E. Leonard (J'34) (LBCMSJ), 7355 S. Yates Ave., Chicago 49, Ill.
Marmorek, Ernest Figdor (J'39) (ACM), 72-38—113th St., Forest Hills, N.Y.
Marny, Robt. G. (J'38), c/o Aeroil Burner Co., Inc., Park Ave. & 57th St., West New York, N.J.
Marquis, Donald H. (J'39) (NSABE), Gen. Engrg. & Cons. Lab., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Marquis, Franklin Parr ('35; '46) (CMSJ), Asst. Mgr., Parr Instrument Co., 222-52nd St., Moline, Ill.
Marquis, Franklin W. ('08; '14) (SFEA), Prof. Mech. Engrg., Chmn. of Dept., Ohio State Univ., Columbus 10, Ohio.
Marquit, Carl Haessler (J'40) (BKAT), Mech. Engr., Gen. Elec Co., 1285 Boston Ave., Bridgeport 2; *for mail*, 78 Adams St., Bridgeport 7, Conn.
Marr, John Hendry (J'46) (BHZ), Sr. Draftsman, Bendix Aviation Corp., Route 1, Norwood; *for mail*, 22 Berlin Ave., Milton, Mass.
Marriner, John M. S. ('39) (EFKLS), V.P., Taylor Engrg. & Constr. Co., Ltd., 80 Richmond St. W., Toronto 1, Ont., Can.
Marrow, Charles Edward (J'43) (NLHK), Engr., Plant Opera. & Equip. Design, Schwarz Engrg. Co., 2312 Middlefield Rd., Redwood City; *for mail*, Box 243, Fremont Ave., Cupertino, Calif.
Marschallinger, Francis L. (J'37) (BDJ), Designer, McNally-Pittsburg Mfg. Corp., 315 W. 3rd St.; *for mail*, 216 E. 10th St., Pittsburg, Kan.
Marsh, Arthur B. ('20; '35) (JBO), Mem. of Firm, Wright, Brown, Quinby & May, 53 State St., Boston 9, Mass.
Marsh, Charles Graham (J'40) (BJM), U.S. Navy; 4817 Baring Ave., East Chicago, Ind.
Marsh, Fred Gaffett (J'42) (HKL), Radio Technician, 2nd Class, U.S.N., Treasure Island; *for mail*, 1105 N. Howard, Glendale 7, Calif.
Marsh, Harry B. ('18) (CLM), Dir. Market Research, Perfect Circle Co., Hagerstown, Ind.
Marsh, Herbert Warren (J'44), Lt., U.S.N.R.; *for mail*, 1543 E. 61st St., Chicago 37, Ill.
Marsh, Robert Henry ('46) (BJM), Ch. Engr., Cal-Cutter Co. & Super Tool Co., 21650 Hoover Rd., Detroit 13; *for mail*, 1844 Yorkshire Rd., Birmingham, Mich.
Marsh, Thomas Alfred ('05; '12) (FSCK), Natl. Indus. Engr., Iron Fireman Mfg. Co., 3107 W. 106th St., Cleveland 11, Ohio.
Marsh, Wilbur Deane (J'42) (ACM), 218 Waldron St., West Lafayette, Ind.
Marshack, Eugene ('45) (BDL), Exec. Engr., Bunge Corp., 80 Broad St., New York, N.Y.
Marshall, Alfred George ('33) (EAFR), Consultant, Shell Oil Co., Inc., 100 Bush St., San Francisco 6, Calif.
Marshall, Arnold Raymond (J'42) (LNYZ), Research Dept., Union Bag & Paper Corp.; *for mail*, 31 Catherine St., Hudson Falls, N.Y.
Marshall, Carl Alexander ('45), Engrs. Club, 32 W. 40th St., New York, N.Y.
Marshall, Clifford W. (J'37) (CMS), 4116 Locust Court, Rainier Vista, Seattle 8, Wash.
Marshall, Daniel Quayle (J'38) (MNJ), Engr., Gen. Elec. Co., Schenectady; *for mail*, c/o F. I. Degen, Leonardsville, N.Y.
Marshall, Donald Eddy ('44) (LOBZMN), Cons. Engr., 16 Greenbriar Dr., Summit, N.J.
Marshall, Donald McMurray (J'41) (JMN), Job Engr., M. W. Kellogg Co., Foot of Danforth Ave., Jersey City 3; *for mail*, 131 Mohawk Dr., Cranford, N.J.
Marshall, Elbert Hall (J'45), Ensign, Civil Engr. Corps, U.S.N.R., Ship's Co., Seabees, Box 4, R/S Navy 3149, F.P.O., San Francisco, Calif.; *for mail*, Box 7, Farmers Branch, Tex.
Marshall, Emerson Sterling (J'43) (RWKJSE), Maint. Engr., Am. Creosoting Co., 401 W. Main St., Louisville, Ky.; *for mail*, 908 Corrine St., Hattiesburg, Miss.
Marshall, Ernest W. ('04), Counselor at Law, 535—5th Ave., New York, N.Y.
Marshall, Harold Benjamin ('40; '44) (CMJ), Staff Engr., Trundle Engrg. Co., 1501 Euclid Ave., Cleveland 15, Ohio; *for mail*, 233 Manor Ave., Elkhart, Ind.
Marshall, Harold Francis ('17; '23; '35) (ACEK), Adv., Sales Exec., Warren Webster & Co., 17th & Federal Sts., Camden; *for mail*, 103 Morgan Ave., Palmyra, N.J.
Marshall, Harold Plunkett (J'46), 9905 River Rd., Hilton Village, Va.
Marshall, Harry Dwight, Jr. (J'43) (MJCBHN), Prod. Engr., Gallmeyer & Livingston Co., 336 Straight Ave., S.W., Grand Rapids 4, Mich.
Marshall, Hugh A. ('46), Asst. to V.P., Natl. Aluminate Corp., 155 E. 44th St., New York 17, N.Y.
Marshall, J. D. (J'41) (BKR), Lt., Canadian Armed Forces; 474 Albert St., Kingston, Ont., Can.
Marshall, James Monroe (J'45) (MBKS), Test Engr., Gen. Elec. Co., 1 River Rd.; *for mail*, 13 State St., Schenectady 5, N.Y.
Marshall, James Thomas (J'42) (HKS), 251 E. Morgan, Spencer, Ind.
Marshall, James West ('23; '31; '35) (SANBHD), Draftsman, Designer, M. H. Treadwell Co. Inc., 140 Cedar St., New York 6, N.Y.; *for mail*, Conklin Ave. & Prospect Pl., Basking Ridge, N.J.
Marshall, Jay Clayton ('29; 42) (CMDL), Plant Engr., Procter & Gamble Mfg. Co., 1232 W. North Ave., Chicago 22; *for mail*, 914 Linden Ave., Oak Park, Ill.
Marshall, Leonard Joseph (J'41) (SFKEJB), Design Research, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Marshall, Paul William (J'42) (CJM), Oper. Engr., Divisional, Aluminum Co. of Am., New Kensington; *for mail*, 308 Highland Rd., Pottstown, Pa.
Marshall, Richard C., Jr. ('19) (CDS), Cons. Engr., Shoreham Bldg., Washington, D.C.
Marshall, Richard Walker (J'43) (SKMCJH), Indus. Sales Engr., Crane Co., 4730—29th St., Long Island City, N.Y.; *for mail*, 98 Lincoln Ave., New Brunswick, N.J.
Marshall, S. W., Jr. ('41), 1245 Shoreham Bldg., Washington, D.C.
Marshall, Stewart M. ('07; '10; F'38) (CFJMOS), Cons. Engr., 68 Post St., San Francisco 4, Calif.
Marshall, Vivus O. ('41) (LKFB), Chem. Engr., Tenn. Eastman Corp.; *for mail*, 1138 Cherry St., Kingsport, Tenn.
Marshall, William ('29; '30; '35) (AMC), Ch. Inspr., Norma-Hoffman Bearings Corp., Stamford; *for mail*, 27 Shore Rd., Old Greenwich, Conn.
Marshall, William ('41) (CDL), Mech. Engr., Charge Design & Specification, Lawrence Peterson & Associates, 312 E. Wisconsin Ave., Milwaukee 2, Wis.
Marsilius, Newman Marius, Jr. (J'41) (CM), Producto Mch. Co., 990 Housatonic Ave., Bridgeport; *for mail*, Daniel's Farm Rd., Trumbull, Conn.
Marsland, Edward H. (J'40), Ernst & Ernst, 2000 Buhl Bldg., Detroit, Mich.
Marslek, Emil Jacob (J'43) (CMNJBH), Asst. to V.P., Acme Indus. Co., 200 N. Laflin St., Chicago 7; *for mail*, 3253 N. Newland Ave., Chicago 34, Ill.
Marston, Roland Gilbert (J'41) (RSEJKA), 1043 Park Ave., Schenectady 8, N.Y.
Mart, Leon Thomas ('42) (CEKMW), Pres., Gen. Mgr., Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan.
Martel, F. Antonio (J'45) (BHS), Compania Cubana de Electricidad, Apartado 390, Santiago de Cuba, Oriente, Cuba.
Martellotti, M. E. ('26; '35) (BAM), Research Engr., Cincinnati Milling Mch. Co., Oakley, Cincinnati 9; *for mail*, 6309 Lisbon Ave., Cincinnati 13, Ohio.
Martens, Howard E. (J'42) (JBH), Research Engr., Calif. Inst. of Tech., 1201 E. California St., Pasadena; *for mail*, 855 Morada Pl., Altadena, Calif.
Martens, Richard Wesley (J'45) (CM), 447 E. 17th St., Brooklyn 26, N.Y.
Martensson, Marten ('24; '35) (FSBK), Sr. Mar. Engr., Bur. of Ships, Navy Dept., Washington; *for mail*, 2641 Connecticut Ave., N.W., Washington 8, D.C.
Marthens, Richard S. ('37) (RCBJMD), Mgr., Diesel Dept., Baldwin Loco. Works, Paschall P.O., Philadelphia, Pa.
Martin, Aaron Jay (J'45) (LHOKES), Jr. Engr., Wood River Refinery, Shell Oil Co., Inc.; *for mail*, 801 Lorena Ave., Wood River, Ill.
Martin, Charles Ralph ('21) (HSB), Engr., Hyd. Dept., Allis-Chalmers Mfg. Co., 1126 S. 70th St., Milwaukee 1; *for mail*, 7122 Hillcrest Dr., Wauwatosa 13, Wis.
Martin, Dean Earl (J'43), 1st Lt., Army Air Forces, A.A.F.T.D., Calif. Inst. of Tech., Pasadena, Calif; *for mail*, R.R. 2, Princeton, Kan.
Martin, Edward J. ('17; '26) (EFS), Mech. Engr., Procter & Gamble Co., Ivorydale; *for mail*, 6122 Robison Rd., Cincinnati, Ohio.
Martin, Elmer C. ('17; '25; '35) (CFLY), 423 Scarritt Bldg., Kansas City, Mo.
Martin, Evan Search ('20) (DCB), Constr. Engr., 16 Saulter St., Toronto 8, Ont., Can.
Martin, Everett Vance ('43) (YDNBJM), Staff Engr., Firestone Tire & Rubber Co., Akron; *for mail*, 1721—7th St., Cuyahoga Falls, Ohio.
Martin, Frank Whitcher ('43) (SEFOKC), Mech. Engr., Stanley Engrg. Co., Hershey Bldg.; *for mail*, 108 W. 7th St., Muscatine, Iowa.
Martin, George H., Jr. (J'44) (NBJA), Instr. in Mech. Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago 16; *for mail*, 2538 W. 66th St., Chicago 29, Ill.
Martin, George W. ('11), 340 Prospect St., Ridgewood, N.J.
Martin, Harold Edward ('29; '30) (SFCK), Dist. Mgr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, Lawrence Farms, Chappaqua, N.Y.
Martin, Harry D. (J'38) (CHM), Engr., Charge of Test Lab., New Departure Div., Gen. Motors Corp., N. Main St.; *for mail*, 33 Melville St., Bristol, Conn.
Martin, Henry Bailey ('27; '36) (CM), Asst. to Controller, York Corp.; *for mail*, Box 363, York, Pa.
Martin, Henry H., Jr. ('26; '34; '35) (EHS), Maj., Corps of Engrs., Pub. Works & Utilities Officer, Military Govt. Det. E-6, Frankfurt, A.P.O. 757, c/o Postmaster, New York, N.Y.
Martin, Jesse C., Jr. ('19), Cons. Engr., Combustion & Refractories, 1325 Miller Dr., Los Angeles, Calif.
Martin, John Gregory ('38) (CEFS), Asst. Dist. Sales Mgr., Babcock & Wilcox Co., 2511 Carew Tower, Cincinnati 2, Ohio.
Martin, John L. ('39; '42) (BHM), Design Engr., Kearney & Trecker Corp.; *for mail*, 1570 S. 75th St., West Allis, Wis.
Martin, Joseph Alfred ('45) (BHM), Mfg. Engr., Asst. V.P. for Prod., Vickers Inc., 1400 Oakman Blvd., Detroit 32; *for mail*, 15394 Ohio Ave., Detroit 21, Mich.
Martin, Kingsley Leverich ('19; F'38) (FS), Pres., Engr. Co., 75 West St., New York 6, N.Y.
Martin, Lee (J'43) (CJM), Snug Harbor, Route 1, Edwardsburg, Mich.
Martin, Lemuel Hillie (J'43) (CMAB), Dist. Rep., Landis Tool Co., Waynesboro, Pa.
Martin, Leo M. ('39) (BCN), Plant Mgr., Pacific Gear & Tool Works, 1035 Folsom St., San Francisco 3; *for mail*, 2200 Leavenworth St., San Francisco 11, Calif.
Martin, Marvin DeWitt (J'42) (BKLN), Engr., Radiation Lab., Univ. of Calif., Berkeley 4; *for mail*, 2854 Shasta Rd., Berkeley 8, Calif.
Martin, Maurice (J'38), (KABYH), Sr. Research Engr., Republic Aviation Corp., Farmingdale, L.I.; *for mail*, 1705 Purdy St., New York 62, N.Y.
Martin, Oscar John (J'42) (EOS), Serv. Engr., Gen. Elec. Co., 140 Federal St., Boston 1; *for mail*, Beagle Manor, Peaches Point, Marblehead, Mass.
Martin, Raymond Anthony ('25; '31; '35) (CSRM), Asst. Ch., Serv. & Maint., N.J. Zinc Co. of Pa.; *for mail*, Residence Park, Palmerton, Pa.
Martin, Roger Ayres ('44) (KLSZ), Dist. Mgr., Yarnall-Waring Co., 408 Bona Allen Bldg., Atlanta 3, Ga.
Martin, Russel W. (31; '35) (CJHASE), Asst. to Maint. Supt., Middletown Div., Am. Rolling Mill Co., Middletown; *for mail*, R.R. 1, Franklin, Ohio.
Martin, Wallace Hope ('13; '19; '28) (EFKS), Prof. Mech. Engrg., Ore. State College; *for mail*, 327 N. 29th St., Corvallis, Ore.
Martin, William Theodore ('43) (LCMKDN), Ch. Engr., Beech-Nut Packing Co., Church St., Canajoharie 1; *for mail*, 87 Cliff St., Canajoharie 3, N.Y.
Martindale, Robert Henry (J'44) (CBM), Plant Engr., Martindale Elec. Co., 1367 Hird Ave.; *for mail*, 1055 Erie Cliff Dr., Cleveland 7, Ohio.
Martindale, Robert M. (J'39) (DLOZRG), Owner, R. M. Martindale & Co., Box 148, Franklin, Ohio.
Martinelli, Raymond C. (J'38) (KHBE), Asst. Prof. Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.
Martinez, Felipe S. (J'42), Central Mercedes, Mercedes, Matanzas, Cuba.
Martinez, Henry Green (J'43), Commercial Engr., Internatl. Gen. Elec. Co. Inc., 1 River Rd., Schenectady, N.Y.
Martinez, Juan Jose ('45) (JLM), Ch., Research & Control Dept., Mexican Light & Power Co., Ltd., Gante 20; *for mail*, Unsumacinta 14, Mexico, D.F., Mex.
Martinson, Earl B. (J'37) (JMC), Fdy. Foreman, Aluminum Co. of Am., 2210 Harvard, Cleveland 9, Ohio.
Martinson, George C. (J'39) (BEWHCD), Ord. Engr., Puget Sound Naval Shipyard; *for mail*, 92-J McNeal Ave., Bremerton, Wash.

Martinson, Raymond Arvid (J'45) (MKRCJD), Operas. Spec. Apprentice, Brake Shoe & Castings Div., Am. Brake Shoe Co., Mahwah, N.J.
Martinto, Pedro (A'01), Pres., Pedro Martinto, Inc., 90 West St., New York, N.Y.
Martis, Jerome Michael (J'45) (COP), Draftsman, Am. Car & Fdy. Co., Berwick; *for mail,* 6 Stewart St., Wilkes-Barre, Pa.
Martus, Martin Lorenz ('12) (CMNRGB), Pres., Treas., Gen. Mgr., Waterbury Battery Co., 1036 S. Main St., Waterbury 86, Conn.
Marty, Edgar Omar ('42) (OSDFHL), Mech., Elec. Engr., Parsons, Brinckerhoff, Hogan & Macdonald, 142 Maiden Lane, New York 7; *for mail,* 84 Long Ridge Rd., Plandome, L.I., N.Y.
Martz, Lawrence S. ('42) (BCM), Asst. to Pres., Micromatic Hone Corp., 8100 Schoolcraft Ave., Detroit 4; *for mail,* Box 255, R.F.D. 3, Pontiac 3, Mich.
Marvin, Philip Roger (J'45) (ABMNYZ), Dir., Research Engrg., Milwaukee Gas Specialty Co., Box 461, 722 N. Jackson St., Milwaukee 1, Wis.
Marwood, Robert Merritt, Jr. (J'45) (FEKMS), Ensign, U.S.N.R., U.S.S. *Vella Gulf,* CVE-111, F.P.O., San Francisco, Calif.; *for mail,* 558 Ashland, River Forest, Ill.
Marx, Erich ('36; '41) (YONMC), Cons. Engr., 3424 W. 8th St., Los Angeles 5, Calif.
Marx, Harry Joseph ('46) (AHO), Comdr., Head, Hyds. & Pneumatics, Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail,* 4818 N. 22nd Rd., Arlington, Va.
Marx, Henry ('80; '92) (CN), Pres., G. A. Gray Co., 3611 Woodburn Ave., Cincinnati 7, Ohio.
Masek, Cyrill A. M. ('43), "Torglen," Grand-View, Nyack, N.Y.
Masino, Frank Donald (J'34) (JMNOHB), Fdy. Product Engr., Eclipse Aviation Div., Bendix Aviation Corp., Williams Ave., Teterboro; *for mail,* 99 Wayne Ave., River Edge, N.J.
Masland, Charles Henry, II ('31) (TCMYB), V.P., C. H. Masland & Sons, Inc., Carlisle, Pa.
Maslow, Ralph (J'42), 2063—77th St., Brooklyn, N.Y.
Mason, C. Keith (J'38) (KEHZ), Engr., Process Engrg., Shell Oil Co., Inc., Box 728, Wilmington; *for mail,* 1354 S. Tanner Ave., Compton, Calif.
Mason, Frank Cleveland (J'39) (CDJKMZ), Quality Control Supvr., Gen. Elec. Co., 1605 Winter St.; *for mail,* 220 E. Jefferson St., Ft. Wayne 2, Ind.
Mason, Harold L. (A'39) (CM), Comdr., U.S.N.R., Employment Supt., Indus. Relations Div., Puget Sound Naval Shipyard, Bremerton, Wash.
Mason, Henry Lea ('25; '31; '35) (ZBKL), Research Prof. Mech. Engrg., Iowa State College, Ames, Iowa.
Mason, Howard Ward ('25) (SEKHFJ), Prof. Mech. Engrg., Ga. Sch. of Tech., Atlanta, Ga.
Mason, John E. ('30) (CFKLOS), Mgr., Sugar Mchy. Dept., Stearns-Roger Mfg. Co., 1720 California St., Denver 2, Colo.
Mason, John Roberts ('41) (TCM), V.P., Millville Mfg. Co., Millville, N.J.
Mason, Martin A. ('32; '40) (HBCO), Prin. Engr., Asst. to Res. Mem., Beach Erosion Bd., War Dept., Little Falls Rd., N.W., Washington 16, D.C.; *for mail,* 205 Raymond St., Chevy Chase 15, Md.
Mason, Richard Nowlin (J'44), 1418 Laurel Ave., Knoxville 16, Tenn.
Mason, Wendell Earle ('21; '39) (SKBMJZ), Assoc. Prof. Engrg., Univ. of Calif. at Los Angeles, 405 Hilgard Ave., Los Angeles 24; *for mail,* 220 S. Swall Dr., Beverly Hills, Calif.
Mason, William A. ('42) (NBFHJG), *Student Award,* '32; Acting Asst. Prof., Mech. Engrg. Dept., Stanford Univ., Stanford University; *for mail,* 836 Forest Ave., Palo Alto, Calif.
Massa, Robert Falconer ('04) (WJ), Secy., Treas., Robert Falconer Massa, Inc., 484 Park Ave.; *for mail,* 510 Park Ave., New York 22, N.Y.
Massare, Quido Pete (J'43) (BASM), Engr., Charge of Design, Warren City Mfg. Co., Warren; *for mail,* Box 166, Windham, Ohio.
Masson, James Ross, Jr. (J'42) (KEFMD), 630 Amherst St., Palo Alto, Calif.
Master, J. N. ('24; '35), Ch. Boiler Inspr., Seervai Block A, Wembridge Compound, Bombay 7, India.
Masters, Romney Wolfe (J'38) (AH), Box 97, Avenal, Calif.
Masters, Wm. C. ('43) (RJOM), Engr., East. Sales Mgr., Flannery Bolt Co., Bridgeville; *for mail,* R.D. 2, Canonsburg, Pa.
Matchett, James C. ('12; '21) (ZSJCFM), V.P., Gen. Mgr., Ill. Engrg. Co., Racine Ave. & 21st St., Chicago 8, Ill.
Matena, Philip (J'42) (AFS), Seaman, 1st Class, Btry 2, Platoon 2, N.T.S. (EE & RM), Gulfport, Miss.; *for mail,* 712—10th St., Union City, N.J.
Mater, Milton Herbert (J'43) (CNWEMD), Owner, Mater Mch. Works, 520 S. 1st St., Corvallis, Ore.
Matheisel, Rudolph Adolph, Jr. (J'45) (ABE KN), Mech. Engr., Application, Davis Engrg. Co., E. Grand St., Elizabeth, N.J.; *for mail,* 555 Ovington Ave., Brooklyn 9, N.Y.
Mather, Archie Joseph ('21) (BJDCM), V.P., Walworth Co., Inc., 313 N. Elm St.; *for mail,* 519 S. Tremont St., Kewanee, Ill.
Mather, Don W. (J'40), 1st Lt., 51st Armd. Inf., Pine Camp, N.Y.
Mather, Robt. Harrison ('22), 51 Elm St., Windsor Locks, Conn.
Mather, Robert Henry (J'41) (CMDLZ), **Jr.** Mgmt. Engr., McClure, Hadden & Ortman, Inc., 111 W. Washington St., Chicago 2, Ill.; *for mail,* Hotel Roberts, Muncie, Ind.
Mather, Thomas H. ('18; '24; '35) (BSFJMC), Ch. Supvr., Application Design, Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail,* 1229 E. 125th St., East Cleveland 12, Ohio.
Matherne, Russell Adam (J'43), 425 S. 21st St., Baton Rouge, La.
Mathes, Earl (J'39) (MCDAEL), Prod. Planner, Wright Aero. Corp., Beckwith Ave., Paterson; *for mail,* 27 Mullen Ave., Midvale, N.J.
Mathes, Stanley Fred (J'39) (CBKM), Sales Engr., Spec. Products, Grinnell Co., Inc., 260 W. Exchange St., Providence 1; *for mail,* 381 Wayland Ave., Providence 6, R.I.
Matheson, Edmund Ernest ('44) (LHCJEM), Sales Mgr., Nordstrom Valve Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Mathews, Henry Mobbett (J'34) (EFS), Supt., City of Thomasville Water & Light Dept., Thomasville, Ga.
Mathews, Ralph Thompson ('30; '35; '38) (SOE FLK), Power Design Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail,* 36 Providence Rd., Chester, Pa.
Mathews, Raymond Clifford (J'39) (CMON), Designing Engr., Humble Oil & Refining Co., Houston 1; *for mail,* 6673 Sylvan Rd., Houston 12, Texas.
Mathews, William Edwin ('95) (BJKERS), Ch. Engr., Air & Gas Compressor Div., Hardie-Tynes Mfg. Co., 800—28th St. N., Birmingham 1; *for mail,* 4328—5th Ave. S., Birmingham 6, Ala.
Mathews, William LeRoy ('30; '37) (CLFSKH), Asst. Ch. Engr., Stand. Oil Co. of Ind., Sugar Creek; *for mail,* 1826 Spruce Ave., Kansas City 1, Mo.
Mathewson, James Stanley ('15; '25) (WDF), Engr., Research, U.S. Forest Products Lab., N. Walnut St., Madison 5, Wis.
Mathewson, Robert Colville ('44) (EFABRH), Asst. Mgr., Am. Bosch Corp., 310 S. Michigan Ave., Chicago 5, Ill.
Mathez, Edmond C. (J'37) (SJOHK), Asst. Engr., Myers, Fuller & Addington, 21 E. 40th St., New York, N.Y.; *for mail,* Mountain Ave., Pompton Lakes, N.J.
Mathis, Eugene ('45) (COMG), Pres., Mathis Ill. Corp., 1935 W. 95th St., Chicago 43, Ill.
Mathis, Harold Fletcher (J'42) (BJCMS), Student, Univ. of Chicago; *for mail,* 1117 E. 60th St., Chicago 37, Ill.
Matiuk, Alexander (J'38) (SFKH), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 6; *for mail,* 597 E. 53rd St., Brooklyn 3, N.Y.
Matlat, Gustave William ('45) (AJMC), Matls. Design Engr., Glenn L. Martin Co., Middle River, Baltimore 3; *for mail,* 816 Ridgeleigh Rd., Baltimore 12, Md.
Matlin, Charles (J'46) (BEJ), Stress Analyst, Oliver Corp., Cleveland; *for mail,* Box 613, Edgewater Branch, Cleveland 7, Ohio.
Matlock, Chauncey ('13) (OSKEFL), Cons. Engr., 70 E. 45th St., New York 17, N.Y.
Matlow, George (J'40) (CJM), Mech. Engr., Colonial Iron Works, 17643 St. Clair Ave.; *for mail,* 3547 E. 154th St., Cleveland 20, Ohio.
Matson, Elmer A. (J'41), Ensign, U.S.N.R., 126 Dinwiddie St., Portsmouth, Va.
Matson, Ray McKinley ('29) (NBMJEH), Head & Prof. of Mech. Engrg., So. Methodist Univ., Dallas 5, Tex.
Matte, Hubert Peter ('40) (HZYCJ), Cons. Engr., Worthington-Gamon Meter Co., 296 South St., Newark 5, N.J.
Matter, Albert John ('45) (MNY), Ch. Engr., Oscar W. Hedstrom Corp., 4836 W. Division St., Chicago 51; *for mail,* 1026 N. Hamlin Ave., Park Ridge, Ill.
Matter, Gustave Oswald ('20; '28) (MYCH), Mech. Cons. Engr., 3112 N.E. 46th Ave., Portland 13, Ore.
Matter, Roger E. (J'37) (CZN), Asst. Engr., Fed. Products Corp., 1144 Eddy St., Providence; *for mail,* 39 Denver Ave., Cranston 5, R.I.
Mattern, John Frederick ('22) (CMDS), Dir., Mch. Tools & Facilities, Elliott Co., P.O. Drawer 800; *for mail,* R.D. 2, Pleasant Valley, Jeannette, Pa.
Mattern, Roy Edwin, Jr. (J'42), Lt., U.S.N.R., 1626 Burlingame Ave., Detroit, Mich.
Matthes, Geo. F. ('39) (EKS), Assoc. Prof. Mech. Engrg., La. State Univ., Baton Rouge 3, La.
Matthes, Max H. (A'22), V.P., Charge of Sales, Firemen's Mutual Ins. Co., 1062 Union Trust Bldg., Cleveland, Ohio.
Matthew, Edward G. (J'41) (CDE), Ensign, U.S.C.G.R., C.G.C. *Cuyahoga,* F.P.O., New York, N.Y.; *for mail,* 440 Jackson St., Gary, Ind.
Matthew, Robert Theodore (J'38) (TYMSDE), Asst. Plant Engr., Gen. Asbestos & Rubber Div., Raybestos-Manhattan, Inc., North Charleston, S.C.
Matthews, Benj. H. (J'36) (BNMHDC), Plant Engr., Lincoln Elec. Co., 12818 Coit Rd., Cleveland 1, Ohio.
Matthews, Burgess Stanley (J'38) (CEMW), Prin. Inspr. (Engrg. Matl.), c/o Inspr. of Naval Matl., U.S.N., Allentown; *for mail,* 79 W. Market St., Bethlehem, Pa.
Matthews, Edward (J'45) (HJS), Instr. in Engrg. Drawing, Univ. of Toronto; *for mail,* 536 Euclid Ave., Toronto, Ont., Can.
Matthews, Noel H., Jr. (J'39) (BCM), Mech. Engr., Gleason Works, 1000 University Ave., Rochester; *for mail,* 34 South Goodman St., Rochester 7, N.Y.
Matthews, Seth Hayden, Jr. (J'41), **Lt.,** 31st Engr. Bn., Ft. Belvoir, Va.
Matthews, Vincent, Jr. (J'42) (NBKDMH), Ch. Draftsman, Fabrication Div., Tenn. Operas., Aluminum Co. of Am., Alcoa; *for mail,* Anderson Ave., Maryville, Tenn.
Matthews, William Edward ('28) (CMNE), Gen. Foreman, Ingersoll-Rand Co., Centre St., Phillipsburg; *for mail,* 85 W. Stewart St., Washington, N.J.
Mattick, Nicholas J. ('39) (BM), Plant Engr., Kellogg Co.; *for mail,* 117 Vale St., Battle Creek, Mich.
Mattimore, John Dalton ('44) (BHJSN), Dir., Research & Devel., Tube Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Matting, Fred Walter (J'40) (EYKMO), Capt., Hdq., Eastern Base Sec., Ord. Sec., A.P.O. 517, New York, N.Y.; *for mail,* 830 Louisiana St., Vallejo, Calif.
Mattingly, Everett A. (J'44) (CKS), Apt. 5-A, 240 W. 104th St., New York, N.Y.
Mattingly, Robert Douglas (J'41) (BHK), 7026 Reedland St., Philadelphia, Pa.
Mattis, Robert Joseph (J'39) (MJCD), Factory Mech. Engr., Philco Corp., G & Tioga, Philadelphia 34; *for mail,* 7783 Hasbrook Ave., Philadelphia 11, Pa.
Mattison, Alan C. (J'30) (CMW), Supt., Secy., Mattison Mch. Works, 545 Blackhawk Park Ave., Rockford, Ill.
Mattke, Chas. F. (J'31), Engr., Charge of Design, Bell Tel. Labs., Inc., 463 West St., New York; *for mail,* 31-33—69th St., Jackson Heights, L.I., N.Y.
Mattlage, Rudolph F. L. ('23; '30; '35) (FKOS), Sales Engr., H. B. Smith Co., Inc., Westfield, Mass.; *for mail,* 12 Archer Rd., Hempstead, L.I., N.Y.
Mattson, Henry (J'38) (EHSBKC), Dist. Petroleum Engr., Tex. Co., Box 2100, Denver 1, Colo.
Mattson, Irwin Ferdinand ('36) (CDKNS), Supt. of Engrg., United Fruit Co., Banes, Oriente, Cuba.
Mau, George Albert ('38) (BDFHMS), V.P., Johnson-March Corp., 52 Vanderbilt Ave., New York 17, N.Y.; *for mail,* 12506 Edgewater Dr., Lakewood, Ohio.
Mauger, David Napier ('37) (SFKM), Exec. Asst., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* 61 Templar Way, Summit, N.J.
Maulbetsch, John L. ('35; '40) (BCMOZL), V.P., Gen. Mgr., Kollmorgan Optical Corp., 2 Franklin Ave., Brooklyn 11, N.Y.
Mauldin, Earle ('40) (TLYZC), So. Editor, McGraw-Hill Publ. Co., Inc., 1011 Rhodes-Haverty Bldg., Atlanta 3, Ga.
Maull, William R. ('21; '28), Lt. Col., Chem. Warfare Serv., 2nd Prov. Regt., Unit Trng. Center, Camp Sibert, Ala.
Maunz, Harold William (J'42) (MC), 169 Kenview Ave., Kenmore, N.Y.
Maurer, Edward Rose ('30) (BASHG), Prof. of Mechanics, Emeritus, Univ. of Wis., Madison; *for mail,* 167 N. Prospect Ave., Madison 5, Wis.
Maurer, Wm. R. ('28; '35), 311 Demott Ave., Teaneck, N.J.
Mäussnest, Frederick K. (J'43) (BAJHM), Mech. Engr., J. I. Bernitz, Inc., 321 Broadway, New York 7; *for mail,* 85-36—212th St., Queens Village 8, N.Y.
Mavis, Frederic Theodore ('33; '34) (HBOC), Prof. & Head, Civil Engrg. Dept., Carnegie Inst. of Tech., Schenley Park, Pittsburgh 13, Pa.
Mawhinney, Matthew Holmes ('22; '30; '35) (FJKO), Cons. Engr., 882 S. Lincoln Ave., Salem, Ohio.
Mawson, Robert ('13) (CLMYDJ), Indus. Engr., 58 Stewart St., Providence 3, R.I.
Maxfield, Edwin David (J'38) ((AFC), Dist. Mgr., Shell Oil Co., Inc., Box 515, La Guardia Field, Jackson Heights; *for mail,* 5 Fairview Ave., East Williston, L.I., N.Y.
Maxfield, H. H. ('04) (CMR), Supt. M.P., Pa. R.R. Co.; *for mail,* 2307 Saymoure Rd., Wilmington, Del.
Maxfield, Horatio Winfred ('42) (CLM), Cleaver-Brooks Co., 50 Church St., New York 7, N.Y.
Maxim, Walter Sam (J'45) (LOSHC), Mech. Engr., Abbott Labs., North Chicago; *for mail,* 1000 Grove St., Evanston, Ill.
Maximon, Louis Bernard ('43) (JFKLZM), Owner, Engr., Thermal Equip. Co., 401 N. Broad St., Philadelphia 8, Pa.
Maxson, Donald John (J'44) (CJM), Technician, 4th Grade, Hdq. Det., 550th QM. Gp., A.P.O. 169, c/o Postmaster, New York, N.Y.
Maxson, John Edgar (J'42), Lt. (j.g.), U.S.S. *Value,* ARS-28, F.P.O., San Francisco, Calif.; *for mail,* 220 Sterling St., Watertown, N.Y.
Maxwell, Bryce (J'43) (YJZGOC), Research Fellow, Plastics Lab., Princeton Univ., Charlton Rd., Princeton, N.J.; *for mail,* Red Spring Lane, Glen Cove, L.I., N.Y.
Maxwell, Carl Allen ('40) (OCSFMB), Supt., Erection Dept., Babcock & Wilcox Co., Barberton; *for mail,* 637 Orlando Ave., Akron 2, Ohio.
Maxwell, George Lorenzo ('27) (CEF), Sr. Engr., Gas Transmission & Distribution, Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia; *for mail,* 220 W. Westwood Pk. Dr., Llanerch, Pa.

Maxwell, Glenn Darryl (J'45) (MCNBSH), Lt. (j.g.), Engr. Officer, U.S.N.R.; 330 N. Whittier Ave., Whittier, Calif.
Maxwell, Harland S. (J'38), 125 Indiana St., Maplewood, N.J.
Maxwell, John Wallace (J'39) (RHOD), Engr. P-2, U.S. Bur. of Reclamation, Golden Eagle Bldg., Denver 2; *for mail,* 1416 Pennsylvania St., Denver 3, Colo.
Maxwell, Maxwell Carson ('08; F'43) (DMCJ), Asst. to Pres., Yale & Towne Mfg. Co., Chrysler Bldg., New York, N.Y.
Maxwell, Richard B. (J'39) (BJDMN), Prod. Engr., Premier Products Co., 124 Stuyvesant Ave., Brooklyn; *for mail,* 436 Crown St., Brooklyn 25, N.Y.
Maxwell, Robert L. (J'44), 405 Huntingdon St., Lexington, Tenn.
May, Bertram (J'39) (NCOJMB), Co-Owner, Charge of Engrg., May Mch. Co., 195 Chrystie St., New York 2, N.Y.
May, Don, Jr. (J'42) (ENABHM), Research Design Engr., Am. Fdy. Equip Co., Mishawaka; *for mail,* 2638 Birchway, South Bend 17, Ind.
May, Edward Anton (J'43) (FSECKN), Owner, May Stoker Co., 2858 N. Major Ave., Chicago 34, Ill.
May, Elliott D. (J'20) (BMW), Ch. Engr., Baxter D. Whitney & Son, Inc.; *for mail,* 32 Academy St., Winchendon, Mass.
May, Harold Edward (J'42) (LSBNZK), Indus. Engr., E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington 99, Del.; *for mail,* Jefferson Apts., Niagara Falls, N.Y.
May, James Edward (J'45) (ACD), Engrg. Dept., Curtiss-Wright Corp., Columbus, Ohio.
May, John P. (J'40) (AHMBZ), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 923 Francis Ave., Schenectady 3, N.Y.
May, John Willard ('45) (BHJN), Ch. Engr., Wright's Automatic Mchy. Co.; *for mail,* 624 Club Blvd., Durham, N.C.
May, Lynnford Edward ('45) (MCDLJA), Tech. Asst. Prod. Planning Mgr., Continental Can Co., Inc., 100 E. 42nd St.; *for mail,* 545—5th Ave., New York 17, N.Y.
Maybank, Herbert Arthur Gordon (J'45) (HKCSMZ), Jr. Engr., Gen. Engrg., Centrifugal Pump Dept., Canadian Ingersoll-Rand Co. Ltd., Des Forges St.; *for mail,* 195 London St., Sherbrooke, Que., Can.
Maybee, Bob (J'42) (CMDJB), Gen. Mgr., Peterborough Lock Mfg. Co. Ltd., 198 Simcoe St., Peterborough, Ont., Can.
Maybury, Richard Doe (J'40) (NSRMGB), Mech. Engr., Equip. Devel., Sylvania Elec. Products Inc., 75 Sylvan St.; *for mail,* 16 Chester St., Danvers, Mass.
Mayer, Charles Warren (J'43) (MSBHY), Lt. (j.g.), U.S.N.R., Cub 10, Div. E-8, F.P.O., San Francisco, Calif.; *for mail,* 945—80th St., Brooklyn 28, N.Y.
Mayer, Fred ('29; '35; '35) (EFKLS), Mech. Engr., Charge of Power Plant Engrg., Celotex Corp., 205 W. Monroe St., Chicago, Ill.
Mayer, John King (J'30) (HESBZJ), Assoc. Prof. & Head, Dept. of Exper. Engrg., Tulane Univ., Sta. 20, New Orleans 15, La.
Mayer, Leo ('16; '35), Pres., Henry Cole-F. C. Hersee Co., 54 Old Colony Blvd., Boston; *for mail,* 8 Grant Ave., Newton Center, Mass.
Mayer, Malvin J. ('39; '42) (OCML), Asst. Secy., Schwarz Labs., Inc., 202 E. 44th St., New York 17, N.Y.
Mayer, Mrs. Olive G. (J'39), Project Engr., Bell & Howell Co., 7000 McCormick Blvd., Chicago; *for mail,* 196 Atteridge Rd., Lake Forest, Ill.
Mayer, Theodore Paul (J'43) (AEMBS), 2nd Lt., Army Air Forces, West. Flying Trng. Comd., Santa Ana, Calif.; *for mail,* 400 E. 180th St., New York 57, N.Y.
Mayer, Wilbert Jacob (J'45), Ensign, U.S.N., LST-610, F.P.O., San Francisco, Calif.; *for mail,* 23 Pennsylvania Ave., Duquesne, Pa.
Mayers, Martin A. ('27; '37) (FKS), Engr., Spec. Projects Dept., M. W. Kellogg Co., Jersey City 3, N.J.
Mayes, Frank F. (J'41) (ABM), Engr., Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 6704 Passaic St., Huntington Park, Calif.
Mayes, Milton S. ('39) (FKS), Ch. Engr., Chesapeake Corp.; *for mail,* Box 455, West Point, Va.
Mayes, Thorn Leavell ('45) (CNBHK), Engr., Supt., Motor Div., Gen. Elec. Co., 5441 E. 14th St., Oakland 1, Calif.
Mayfield, Robert King (J'41) (TCL), Trainee, Merion Worsted Mills, Ford Rd., West Conshohocken, Pa.
Mayhew, B. Alan ('20; '33) (CLS), Supt., Power Plant, Natl. Sugar Refining Co., 2-03—55th Ave., Long Island City, N.Y.; *for mail,* 11 North Brae Court, Tenafly, N.J.
Mayhew, Kenneth H. (J'46), Mech. Engr., West. Elec. Co., Inc., 195 Broadway, New York, N.Y.
Mayhew, Ray Winfield (J'41), Lt., U.S.N.R., U.S.S. *Chenango,* c/o Postmaster, New York, N.Y.
Maynard, Charles Edgar ('31) (CYM), V.P., Opera. Mgr., Prophylactic Brush Co., Florence, Mass.
Maynard, Harold Bright ('43; '44) (CMO), Pres., Methods Engrg. Council, 822 Wood St., Pittsburgh 21, Pa.
Maynard, William G. (J'43) (ABCLYF), Design Engr., Glenn L. Martin Co., Middle River; *for mail,* 1508 Pentridge Rd., Baltimore 12, Md.
Mayo, Albert Richard ('27) (SFE), Engr., Design, Fed. Shipbldg. & Dry Dock Co., Kearny; *for mail,* Box 186, Basking Ridge, N.J.
Mayo, Clyde Robinson (J'42) (NZMDCA), Devel. Engr., Chefford Master Mfg. Co.; *for mail,* 212 S.W. 11th St., Fairfield, Ill.
Mayo, Edmund Cooper ('16), Pres., Gen. Mgr., Gorham Mfg. Co., Providence, R.I.
Mayo, Nestor Silvio (J'45) (SKEJFN), Trainee, Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1; *for mail,* 2028 S. 71st St., West Allis 14, Wis.
Mayo, Paul Henry (J'44) (HKS), c/o Melle Germaine Mestre, 6 Villa Jocelyn, Square Lamartine, Paris 16, France.
Mayr, Karl Alexander ('27; '35) (SETCBK), Pat. Atty., 21 E. 40th St., New York 16, N.Y.
Mayrose, Herman Everett ('29) (BJMGY), Prof. Engrg. Mechanics, Univ. of Detroit, Livernois & McNichols Rd.; *for mail,* 18286 Strathmoor Ave., Detroit 21, Mich.
Maytham, Walter J. ('05) (CFL), Ch. Engr., Charge Design, Northwest. States Portland Cement Co.; *for mail,* 936 N. Jefferson Ave., Mason City, Iowa.
Mazurie, Jas. V. ('29) (CDJ), Retired; Route 5, Union City, Ind.
Mazza, Charles Angelo ('45) (NYMAJS), Mch. & Turbine Designer, Gen. Elec. Co., 920 Western Ave.; *for mail,* 267 Chatham St., Lynn, Mass.
Mazzaglia, Joseph Dominic (J'45) (OMC), Engrg. Consultant, 389 Kingston Ave., Brooklyn; *for mail,* 75-31—184th St., Flushing, N.Y.
Mazzola, Ambrose (J'45) (CDO), Ch. Engr., Angelus Engrg. Co., 2466 E. 56th St., Los Angeles 11; *for mail,* 8915 Dorothy Ave., Southgate, Calif.
Mazzola, Joseph Anthony (J'44) (NBMJCY), Engrg. Designer, Internatl. Business Machines Corp., Endicott, N.Y.
McAdam, Lon James, II (J'44) (CMDLJH), Mem., McAdam & Sons, Box 1, Barker 1, N.Y.
McAdams, Jos. E. ('19; '22), Pres., Steel Products Engrg. Co., Dakota Ave. & Columbia, Springfield; *for mail;* Stonyridge Farms, New Carlisle, Ohio.
McAfee, Howard Cary (J'45) (KJHBC), Ensign, U.S.N., U.S.S. *Smith,* DD-378, F.P.O., San Francisco, Calif.; *for mail,* Box 26M, Route 1, New Orleans 14, La.
McAfee, W. Keith ('23; '31) (LC), Pres., Univ. Sanitary Mfg. Co., Box 300, Redlands, Calif.
McAleer, William Kearns (J'42) (CAMZ), Pat. Engr., Mine Safety Appliances Co., 201 N. Braddock Ave., Pittsburgh; *for mail,* 332 S. Pacific Ave., Pittsburgh 24, Pa.
McAlister, Anderson O'Jay (J'42) (JSEH), Engrg. Asst., E. I. du Pont de Nemours & Co., Wilmington; *for mail,* 1319 W. 6th St., Wilmington 34, Del.
McAllister, Alfred John ('26; '32; '35) (CMJDN), Pres., Fairfield Mfg. Co.; *for mail,* 1401 S. 9th St., Lafayette, Ind.
McAllister, John A. (J'43), Lt., Army Air Forces, Rapid City, S.D.; *for mail,* R.D. 1, Pleasant Valley, N.Y.
McAllister, William John (J'44) (BSMEDZ), 1st Lt., Ord. Dept., Commanding Officer, 250th Ord. Ballistics Det., Army Serv. Forces Trng. Center, Aberdeen Proving Ground, Md.; *for mail,* 3412 St. Vincent St., Philadelphia 24, Pa.
McAlpin, William J. ('37; '44) (CJMST), Pres., J.J. Finnigan Co., Inc., 455 Means St., N.W., Atlanta, Ga.
McAmis, John Carl (J'43) (CDMLBG), Head, Methods Analysis Sec., Carbide & Carbon Chemicals Corp., Oak Ridge; *for mail,* Westmoreland Heights, Knoxville, Tenn.
McAndrew, Robert Gavin ('41) (CMR), Tech. Asst. to V.P., Am. Loco. Co., Schenectady 5, N.Y.
McAndrew, Robert Gavin, Jr. (J'46) (JKR), Lt., U.S.N.R., Navy 3245, Box 150, F.P.O., San Francisco, Calif.; *for mail,* 5—1st Ave., Johnstown, N.Y.
McAninch, Herbert Austin ('34; '41) (JBCAHN), Asst. Factory Mgr., Warner Automotive Parts Div., Borg-Warner Corp.; *for mail,* 814 N. Main St., Auburn, Ind.
McAnulty, James Carstairs (J'38) (NBAG), Mech. Research Engr., Project 22, Elec. Engrg. Dept., Tech. Inst., Northwestern Univ., Evanston, Ill.
McArdell, Wesley E. ('15; '20; '26) (JM), Teacher, Brooklyn Tech. High Sch., 29 Ft. Greene Pl., Brooklyn; *for mail,* Emerson Hill, Stapleton, S.I., N.Y.
McArthur, Ralph Ford (J'40) (HBJY), Ch. Engr., Kobe, Inc., 3040 E. Slauson Ave., Huntington Park, Calif.
McArthur, Warren ('45), Pres., Warren McArthur Corp., 1 Park Ave., New York, N.Y.; *for mail,* Bantam, Conn.
McAulay, Frederick Lawrence (J'42) (SKACGO), Steam Engr., Westinghouse Elec. Corp., 411 N. 7th St., St. Louis 1, Mo.
McAulay, Hubert Joseph (J'42) (EBHSZN), Instr. in Mech. Engrg., Univ. of Detroit, McNichols Rd. at Livernois, Detroit 21, Mich.
McAuley, Benj. F. ('20) (ACD), Matl. Handling Engr., Hawthorne Sta., West. Elec. Co., Inc.; *for mail,* 3909 Van Buren St., Chicago, Ill.
McAuley, Kenneth James (J'42) (ABE), 855 N. LeClaire Ave., Chicago 51, Ill.
McAuley, McGill Matthew ('43) (CHLS), Sales Engr., Crawford Sprinkler Supply Co., Box 28, East Point; *for mail,* 973 Austin Ave., N.E., Atlanta, Ga.
McAuliffe, Pierce J. ('19) (H), Agt., Morris Mch. Works, 254 W. 31st St., New York 1, N.Y.
McBath, Bartley Russell ('39) (CBHRSK), Ch. Engr., Research Dept., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 11 Ferncliff Terrace, Glen Ridge, N.J.
McBean, Douglas M. (J'42), 315 Alexander St., Rochester 4, N.Y.
McBee, Edgar Love ('44) (BCFKS), Supt. Power Prod., Toledo Edison Co., 420 Madison Ave., Toledo 4, Ohio.
McBerty, Don R. ('34; '35) (NBCD), Mech. Engr., Champion Paper & Fibre Co.; *for mail,* 76 Newfound St., Canton, N.C.
McBerty, Ford Hapgood ('40) (CLMJND), Asst. to Asst. Ch. Engr., E. I. du Pont de Nemours & Co., Wilmington 98; *for mail,* 14 W. Delaware Ave., Newark, Del.
McBride, Edward John (J'45) (ABS), Devel. Engr., Worthington Pump & Mchy. Corp.; *for mail,* 29½ Central Pl., Wellsville, N.Y.
McBride, John J. ('17; '35) (ERY), Engr., Car Constr., Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.
McBride, Thomas C. ('08) (RSK), Cons. Engr., R.R. Dept., Worthington Pump & Mchy. Corp., 2905 N. Broad St., Philadelphia 32, Pa.
McBride, W. J. ('20) (ABJ), Asst. Engr., Charge of Turbo-Blower Dept., Ingersoll-Rand Co., Phillipsburg, N.J.
McBrien, Robert Elwood ('25; '32; '35) (BHKS), Designing Draftsman, Canadian Gen. Elec. Co. Ltd., Park St.; *for mail,* 485 McDonnel St., Peterborough, Ont., Can.
McBryde, Warren H. ('21; F'42) (OCMNRS), *Vice-President,* '37-'39, *President,* '40; Cons. Engr., Financial Center Bldg., San Francisco, Calif.
McBurney, John White ('26) (YB), c/o Natl. Bur. of Stands., Washington 25, D.C.
McBurney, Willard B. ('41) (S), Owner, McBurney Stoker & Equip. Co., 565 W. Peachtree St., N.E., Atlanta, Ga.
McCabe, Clarence Howard (J'40) (RCSA), Gen. Elec. Co., 920 Western Ave., Lynn 3; *for mail,* 24 Hamilton Rd., Wakefield, Mass.
McCabe, Frank Eugene ('20) (BCD), Retired; N. Miles Rd., Chagrin Falls, Ohio.
McCabe, Ira E. ('39) (BDF), Ch. Engr., Chmn. of Bd., Mercoid Corp., 4201 Belmont Ave., Chicago, Ill.
McCabe, Warren Lee ('46) (KLP), Head, Chem. Engrg. Dept., Carnegie Inst. of Tech., Pittsburgh 13; *for mail,* 325 Inglewood Dr., Pittsburgh 16, Pa.
McCall, Desmond (J'36) (CM), Asst. Gen. Supt., Columbia Steel Co.; *for mail,* 1335 Pine St., Pittsburg, Calif.
McCallum, Robert Anthony (J'38) (NBCM), Asst. Ch. Engr., Automatic Instrument Co., 679 N. Wells St., Chicago 10, Ill.
McCandless, William ('44) (ZYGCEM), Instrument Engr., Elec. Autolite Co., Toledo; *for mail,* 1732 Fairfax St., Toledo 6, Ohio.
McCann, Charles Sumner ('44) (HB), Dir. of Research, Walker Mfg. Co., Racine, Wis.
McCann, Warren E. ('42) (MOCJ), Consultant Engr., French Supply Mission of Indus. Prod., 1822 Massachusetts Ave., Washington 6, D.C.; *for mail,* 301 W. School Lane, Philadelphia 44, Pa.
McCanna, F. James (J'43) (SLMCWO), Assoc. Engr., Matl. Inspc., Bonneville Power Admin., Portland 8; *for mail,* 3515 N.E. Hassalo, Portland 13, Ore.
McCanna, Louis A. (J'44) (EJR), 1709—22nd Ave., Longview, Wash.
McCants, Robert Preston (J'40) (MNBCD), Welding Engr., Gen. Elec. Co., 6901 Elmwood St., Philadelphia 42; *for mail,* 250 Bridge St., Drexel Hill, Pa.
McCarthy, Edmund ('28; '34; '35) (FKRS), Fuel Engr., Fairmont Coal Bur., 122 E. 42nd St., New York 17, N.Y.
McCarthy, Edward Schenke (J'43) (ASR), Engr., Transcontinental & West. Air, Inc., 14 W. 3rd St., Kansas City, Mo.
McCarthy, Edward Wister (J'32) (CGHM), V.P., Naval Arch., Gulfport Boiler & Welding Works, Inc., Box 1179; *for mail,* 2605 Lake Shore Dr., Port Arthur, Tex.
McCarthy, Eugene Robert ('23; A'34) (CEMNS), Dist. Mgr., Crocker-Wheeler Elec. Mfg. Co., 629 Euclid Ave., Cleveland 14; *for mail,* 3077 Meadowbrook Blvd., Cleveland Heights 18, Ohio.
McCarthy, Harry ('09) (CESJMN), Retired; 700 S. Chestnut St., Kewanee, Ill.
McCarthy, Justin Howard ('22; '35) (WHSCDL), Ch. Engr., Soundview Pulp Co., Everett, Wash.
McCarthy, Justin J. ('29; '38) (SCR), Mgr., Philadelphia Spec. Sales, Cochrane Corp., 17th & Allegheny Ave., Philadelphia; *for mail,* 514 Monroe Rd., Merion, Pa.
McCarthy, Robert Hughes (J'41) (MTCNJY), Prod. Engr., Textiles, Barber-Colman Co., Loomis St.; *for mail,* 703 Locust St., Rockford, Ill.
McCarthy, Rollin H. ('28; '35) (CMDWB), Supt. Mfg. Engrg., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* Park Lane, Madison, N.J.
McCarty, Roy A. ('36) (C), Retired; 1124 Edmonds Ave., Drexel Hill, Pa.
McCathron, Claude B. ('42) (CNMZJA), Design Engr., Graflex Inc., 154 Clarissa St., Rochester 8, N.Y.

McCausland, James Russell, Jr. (J'46) (CTZ), Test Course Engr., Works Lab., Gen. Elec. Co., West Lynn, Mass.; *for mail,* 1206-B Dermond Rd., Drexel Hill, Pa.
McCausland, John Woods ('19; '25; '35), Design Engr., Universal Oil Products Co., 310 S. Michigan Ave., Chicago 4; *for mail,* 5739 Kimbark Ave., Chicago 37, Ill.
McCay, William Thompson ('43), 806 Parklane Apts., 5701 Jackson, Houston 4, Tex.
McChesney, Irving G. (J'24) (AHS), 609 S. Washington St., East Rochester, N.Y.
McClain, Richard Eugene (J'36) (EBCSJN), Supvg. Engr., Caterpillar Tractor Co., Peoria 8; *for mail,* 1101 Norwood Ave., Peoria, Ill.
McClay, Clarence Harvey (J'43) (AEFKLZ), Maj. Engr., Power Plant Controls, Boeing Aircraft Co., Seattle 14; *for mail,* 5810 Cowan Pl., Seattle 5, Wash.
McCleary, Royal E. (J'44) (FKSCM), Ch. Engr., Pa. Elec. Co.; *for mail,* Violet St. Ext., R.D. 3, Johnstown, Pa.
McClees, Alexander Newell (J'45), 50 W. Front St., Red Bank, N.J.
McClellan, Donald Robert (J'46) (KLM), 2002 Cortland Ave., Detroit, Mich.
McClellan, William ('05), Chmn. of Bd., Union Elec. Co. of Mo., 315 N. 12th Blvd., St. Louis 1, Mo.
McClelland, Cory Chalmer (J'13) (SFHEZ), Mgr., Cleveland Dist., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail,* 15848 Glynn Rd., East Cleveland 12, Ohio.
McClelland, Dean Smucker (J'46) (ABD) Test Engr., Seward Power Plant; *for mail,* Box 166, Seward, Pa.
McClelland, James Edmond ('46) (BES), 2632 Sennett Ave., Wichita 9, Kan.
McClintock, Frank Ambrose (J'44) (BAHJK), Analytical Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 32 Denison St., Hartford 5, Conn.
McClintock, Frank Stockton ('20) (SHE), Ch. Engr., Mchy. Div., Dravo Corp., 300 Penn Ave., Pittsburgh; *for mail,* 805 Amberson Ave., Pittsburgh 6, Pa.
McClung, James Matthew (J'36) (STZKFL), Engr., Arlington Mills, Broadway, Lawrence; *for mail,* 127 High St., North Andover, Mass.
McClure, Arthur Wilfred (J'42) (ABS), *Student Award,* '42; Sr. Detailer, Blower Engrg. Dept., Elliott Co., Jeannette; *for mail,* 1005 N. Sheridan Ave., Pittsburgh 6, Pa.
McClure, Charles R. ('22; '25; '35) (CFKS), Cleveland Sales Office, Babcock & Wilcox Co., 1515 Guardian Bldg., Cleveland 14, Ohio.
McClure, John Burns ('39) (SEFRB), Application Engr., Cent. Sta. Engrg. Div., Gen. Elec. Co., 1 River Rd.; *for mail,* R.D. 1, Hedgewood Lane, Schenectady, N.Y.
McClure, Samuel Sherman ('45) (CLMZ), V.P., Kimble Glass Co.; *for mail,* Landis & Valley Ave., Vineland, N.J.
McCoard, Arthur Palen ('45) (CSFKHZ), Plant Engr., Container Corp. of Am., Box 70, Circleville, Ohio.
McCollam, Charles H. ('39) (CKL), Gen. Sales Mgr., Steel & Tube Div., Timken Roller Bearing Co., Canton, Ohio.
McCollum, Paul Earnest (J'44) (AEF), Cons. Engr., 304—1st Natl. Bank Bldg., Santa Ana, Calif.
McConaghy, James William ('30; '42) (HNBS EL), Design Engr., Ingersoll-Rand Co.; *for mail,* 609 Morris St., Phillipsburg, N.J.
McConaghy, William Brooks (J'43), Avonbrooke Rd., Wallingford Hills, Pa.
McConathy, Donald Reed ('42; '45) (SOC), Project Engr., Burns & Roe, Inc., 233 Broadway, New York 7; *for mail,* 29-33—165th St., Flushing, N.Y.
McConnell, Glenver ('44) (CEOWY), Shell Oil Co., Inc., Box 1191, Tulsa 2; *for mail,* 1440 E. 17th Pl., Tulsa 5, Okla.
McConnell, Malcolm Findley ('44), Mgr., Research & Devel. Div., Carnegie-Ill. Steel Corp., 714 Frick Bldg., Pittsburgh 30, Pa.
McConnell, Malcolm Rosebrook ('37) (CJMS FW), Indus. Engr., Power Sales & Applications, West Penn Power Co., 14 Wood St., Pittsburgh 30, Pa.
McConnell, Richard Urquhart (J'43) (CKHM OB), Lt. (j.g.), U.S.N.R., 3706 Lime Ave., Long Beach 7, Calif.
McConnell, W. Mynard ('44), Ch. Engr., Mackintosh-Hemphill Co., 901 Bingham St., Pittsburgh; *for mail,* 3000 Myer Blvd., McKeesport, Pa.
McCool, Harry Schreck (J'41) (ACJM), Sales Engr., Steel & Tube Div., Timken Roller Bearing Co., 1601 Reading Rd., Cincinnati 2; *for mail,* 6968 Thorndyke Dr., Mariemont, Cincinnati 27, Ohio.
McCorkle, Lee Raible (J'43) (SZLHKD), Jr. Mech. Engr., Am. Maize Products Co., Roby, Ind.; *for mail,* 8344 Lafayette, Chicago 20, Ill.
McCormack, Daniel James ('18) (HSBC), Sales Mgr., S. Morgan Smith Co., Lincoln St., York, Pa.
McCormack, Francis Patrick (J'44) (CMDO), Ch. Indus. Engr., LaBelle Works, Crucible Steel Co. of Am., Ridge Ave. & Reedsdale St., N.S., Pittsburgh 12; *for mail,* 140 Victory Lane, Leetsdale, Pa.
McCormick, Charles Donald ('45) (MJOCDN), Mech. Engr., Indus. Engrg. Dept., E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington, Del.; *for mail,* 9 Seaside Pl., East Norwalk, Conn.
McCormick, Clarence Everett ('37) (YF), Ch. Engr., Republic Rubber Co., Albert St., Youngstown; *for mail,* Box 177, Poland, Ohio.
McCormick, F. Howard ('31) (NMJKBC), Asst. Ch. Engr., Frigidaire Div., Gen. Motors Corp., Dayton 1; *for mail,* 174 Lookout Dr., Dayton 9, Ohio.
McCormick, H. Barker, Jr. (J'43) (AZBM), Project Engr., Naval Air Exper. Sta., Naval Base, Philadelphia 12; *for mail,* 236 W. Garfield, Norwood, Pa.
McCormick, Thomas Joseph (J'43) (LYHCZS), Asst. to Plant Engr., Shawinigan Resins; *for mail,* 1104 Worthington St., Springfield, Mass.
McCort, James Edward (J'44) (RCM), Dist. Mgr., Ry. Div., Timken Roller Bearing Co., 1832 Union Commerce Bldg., Cleveland 14, Ohio.
McCoy, Marion Charles (J'43) (KSE), 937 Douglas St., Corinth, Miss.
McCoy, Richard Kern (J'43) (NKEOH), 1317 Cresse Ave., R.D. 7, Schenectady, N.Y.
McCoy, V. E. ('42) (RCK), Ch. Engr., Natl. Aluminate Corp., 6216 W. 66th Pl., Chicago 38, Ill.
McCoy, William I. ('03; '25) (NBOKS), 634 Adair Ave., Zanesville, Ohio.
McCoy, William Wayne (J'44) (ASNHFK), Design Engr., Airplane Div., Curtiss-Wright Corp.; *for mail,* 50 E. Oakland Ave., Columbus, Ohio.
McCracken, William Clyde ('15) (SFEK), Ch. Engr., Supt. of Bldgs., Ohio State Univ., Columbus 10; *for mail,* 2005 Berkshire Rd., Columbus 8, Ohio.
McCrady, Louis de B. ('21), 74 South Battery, Charleston, S.C.
McCrary, Otis Frederick, Jr. (J'43) (AEL), Lt. (j.g.), U.S.N.R.; 127 Brooks Ave., Raleigh, N.C.
McCray, Clarence Richard (J'41) (RAHYN), Design Engr., Garlock Packing Co.; *for mail,* 125 Washington St., Palmyra, N.Y.
McCready, John Thomas (J'46) (CH), 250 Normandy Lane, Concord, Calif.
McCuaig, Donald H. ('41) (FKS), Head, Sales Application, Worthington Pump & Mchy. Corp., Holyoke, Mass.
McCubbin, John Daniel Service ('46) (EOS), Mar. Inspr., U.S.C.G.R., 42 Broadway, New York; *for mail,* 784 Carroll St., Brooklyn, N.Y.
McCue, Jas. O. ('26) Pres., Stamford Rolling Mills Co., Springdale, Conn.
McCulloch, A. Donald (J'40) (CJM) V.P., McCulloch Mfg. Co., 200 Old Colony Ave., South Boston, Mass.
McCulloch, William McPherson (J'42) (CEK LY), Mech. Engr., Maint., Styrene Div., Dow Chem. Co., Box 500, Gardena; *for mail,* 1500 W. 53rd St., Los Angeles 37, Calif.
McCullough, William Thomas, Jr. ('20; '26; '35) (CEFKS), Dist. Mgr., Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3, Ill.
McCully, Harry McKee ('45) (GLM), Prof. & Head, Dept. of Drawing & Descriptive Geometry, Carnegie Inst. of Tech., Schenley Park, Pittsburgh 13; *for mail,* Oak Hill Farms, R.D. 2, Allison Park, Pa.
McCune, Jos. C. ('16; '27) (BCR), Dir. of Research, Westinghouse Air Brake Co., Wilmerding; *for mail,* 420 Locust St., Edgewood, Pittsburgh, Pa.
McCurdy, James David (J'43), Lt., U.S.A., 1383rd Engrg. Petroleum Distributing Co., A.P.O. 503, c/o Postmaster, San Francisco, Calif.; *for mail,* 319—17th St., N.W., Puyallup, Wash.
McCurdy, Robert B. (J'41) (DLYM), Sales Engr., B. W. Rogers Co., 850 S. High St., Akron; *for mail,* 839 Orlando Ave., Akron 2, Ohio.
McCutchan, Arthur (J'31) (SBJLF), Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 14466 Mansfield St., Detroit 27, Mich.
McDaniel, Weymouth Neil (J'46), Research Asst., Johns Hopkins Univ., Baltimore 18; *for mail,* 86 Kinship Rd., Dundalk 22, Md.
McDermott, Charles Clifford ('31) (CKJHDM), Pres., Gen. Mgr., McDermott Water Heaters, Inc., 514 Genesee St., Trenton 10, N.J.
McDermott, Gerald Francis (J'42) (SKAMCD), Foreman, Babcock & Wilcox Co., Stirling Ave.; *for mail,* Box 540, R.D. 2, Barberton, Ohio.
McDivitt, Elvin Theodore ('38; '42) (BHAJNL), Ch. Estimator, Lancaster Iron Works, Inc., 560 S. Prince St.; *for mail,* R.D. 3, Lancaster, Pa.
McDonald, Allan Arch (J'43), 819 W. Brooks, Norman, Okla.
McDonald, Daniel Everett (J'43) (BHLMNR), Quality Mgr., McKiernan Terry Corp., Harrison, N.J.; *for mail,* 4541—39th Pl., Sunnyside, L.I., N.Y.
McDonald, Edward Lincoln ('28) (EFKSZ), Results Engr., Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City 10; *for mail,* 5001 Virginia Ave., Kansas City 4, Mo.
McDonald, James F. ('45) (EKP), Dist. Engr., Gulf Oil Corp., Box 1111, Seminole, Okla.
McDonald, James M. (J'39) (CMNOJA), Mch. Devel. Engr., E. I. du Pont de Nemours & Co., Wilmington 98; *for mail,* 33 N. Pennewell Dr., Wilmington 261, Del.
McDonald, Raymond Neil ('37; '42) (FESANB), Asst. Prof., Mech. Engrg., Vanderbilt Univ., Nashville 4, Tenn.
McDonald, William Albert ('20) (BCFKSZ), Supt. of Power, Houston Ltg. & Power Co., Houston, Tex.
McDonnell, William Patterson ('45), Ch. Observer, Test Dept., Pa. R.R. Co., Altoona; *for mail,* 918 Walnut St., Hollidaysburg, Pa.
McDonough, P. W. ('21; '31) (CJM), Owner, Mgr., McDonough Steel Co., 800—75th Ave., Oakland, Calif.
McDonough, Thomas Lawrence (J'45) (NYMR), Jr. Mech. Engr., Design, Sylvania Elec. Products, Inc., 126 Washington St., Salem; *for mail,* 604 Western Ave., Lynn 3, Mass.
McDougall, George F. ('21) (OBHY), Cons. Engr., Board of Trade Bldg., Portland 4, Ore.
McDowell, Charles H. (J'41) (EJSBHF), Lt. (j.g.), U.S.N.R., U.S.S. *Altamaha,* F.P.O., San Francisco, Calif.; *for mail,* 128 Mayland St., Philadelphia 44, Pa.
McDowell, Robert Wesley (J'30) (CDFOZ), Plant Engr., Ford Instrument Co., 31-10 Thompson Ave., Long Island City 1, N.Y.
McDowell, Willis Earl ('22; '29; '35) (FSCK MZ), Supt., Tiger Steam Sta., Duke Power Co.; *for mail,* Box 37, Duncan, S.C.
McEachern, Joe Albert ('40; '44) (OCHNBM), Lt. Col., Corps of Engrs., U.S.A., 2571st Sta. Complement Unit, Sta. Unit, Ft. Belvoir; *for mail,* 373 N. Granada St., Arlington, Va.
McEachern, Thomas H., Jr. (J'39) (ACM), 4550 W. 63rd St., Los Angeles 43, Calif.
McEdwards, James Angus (J'45) (BCEGJN), Asst. to Ch. Engr., Ingalls Iron Works; *for mail,* Box 92, Route 2, Birmingham, Ala.
McEllhiney, William Alva ('26; '35) (HCELR), Pres., J. P. Miller Artesian Well Co., Box 359, Brookfield, Ill.
McElroy, John Henry ('40) (GMYAC), Ch. Draftsman, R. C. Coxhead Corp., 333—6th Ave., New York; *for mail,* Box 315, Pearl River, N.Y.
McElroy, John Jerome ('30) (TYM), Gen. Supt., Maverick Mills, East Boston; *for mail,* 267 Court Rd., Winthrop, Mass.
McElroy, Robert Kenneth (J'42) (NOMJDR), Mech. Engr., A. J. Boynton Co., 58 E. Washington Blvd., Chicago; *for mail,* 4952 N. Seeley Ave., Chicago 25, Ill.
McElwain, Sharp McCrea (J'40) (BFHJZ), Asst. to Welding Engr., Am. Bridge Co., Park Rd., Ambridge, Pa.
McElwee, John J. (J'42), U.S.N.R., 3019 W. Lincoln Ave., Milwaukee, Wis.
McEwan, Thomas Spring ('15; '17; '25) (CMO), *Manager,* '41-'44, *Vice-President,* '44-'46; V.P., McClure, Hadden & Ortman, 111 Washington St., Chicago 2; *for mail,* 1046 Dinsmore Rd., Winnetka, Ill.
McEwen, Clifford William (J'40) (KDC), Engr., S. A. Armstrong Ltd., 115 Dupont, Toronto; *for mail,* 35 Jenning Ave., Toronto 9, Ont., Can.
McEwen, Ewen Gordon (J'45) (BN), Asst. Dir., F.V.D.D., Ministry of Supply, London, W.C. 2; *for mail,* 29 Montpelier Row, Twickenham, Middlesex, England.
McEwin, John Ben (J'42) (ELHSN), Test Engr., Humble Oil & Refining Co.; *for mail,* 250½ Virginia St., Baytown, Tex.
McFadden, Benjamin Curtis ('40) (BDFHJS), Ch. Mech. Engr., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19, Pa.
McFarland, John D., Jr. (J'38) (SFMCE), 6405 Flander Dr., Hampshire Knolls, Hyattsville, Md.
McFarland, Robert E. (J'42) (CH), Lt., U.S. N.R., Route 3, Oklahoma City, Okla.
McFarland, Robert West (J'37) (ACB), Field Engr., Sales, Am. Blower Corp., 1003 Statler Bldg., Boston 16, Mass.
McGann, Robert Greaves ('23), Retired; 332 S. Michigan Ave., Chicago 4, Ill.
McGee, Hugh Paul (J'41) (MANSD), Mech. Engr., River Works, Gen. Elec. Co., Lynn; *for mail,* 14 Grove St., Reading, Mass.
McGee, J. John ('24; '35), Draftsman, Engrg. Dept., Canadian Pacific Ry., Rm. 401, Windsor Sta., Montreal; *for mail,* 5939 Dorocher Ave., Outremont, Que., Can.
McGee, Patrick A. ('44) (AEFKRS), Asst. Mgr., East. Region, Electro-Motive Div., Gen. Motors Corp., 230 Park Ave., New York 17, N.Y.
McGee, Wm. A. ('20) (BFKNRS), Mech. Engr., N.Y. Cent. R.R. Co., 1324 W. 3rd St., Cleveland 13, Ohio.
McGibbon, Donald G. (J'36) (CMR), 172 Prospect Pl., Rutherford, N.J.
McGinn, Leo Francis ('31; '35), Sales Engr., Grinnell Co., 260 W. Exchange St., Providence; *for mail,* 38 E. Main St., West Warwick, R.I.
McGinnis, C. Edwin ('38) (BCM), Gen. Mgr., Bd. of Mech. Engrs., City of Los Angeles, Rm. M-85, City Hall, Los Angeles, Calif.
McGinnis, David McKnight (J'44) (SKEO), Cons. Engr., 3138 W. Slauson Ave., Los Angeles 43, Calif.
McGinnis, Frank Darrel (J'42), Lt. (j.g.), U.S.N.R., E 8-33, Ship Repair Base, Navy 3864, F.P.O., San Francisco, Calif.
McGinnis, James Gates (J'45) (NOBHDJ), Ch. Draftsman, McPhee & Johnston, 90 West St., New York 6, N.Y.
McGirr, Robert (J'38) (NLYJ), Mech. Engr., RXL Lab., Atlas Powder Co., Reynolds; *for mail,* 204 W. Market St., Pottsville, Pa.
McGivern, James G. (J'37) (BFJ), Dean of Engrg., Gonzaga Univ., Spokane, Wash.

McGladrey, Lyle L. ('46) (W), c/o Wood Conversion Co., Cloquet, Minn.
McGonigle, Chas. ('19) (CJ), Partner, Poole & McGonigle, 6330 Halsey St., Portland, Ore.
McGorman, Donald G. (J'40) (CJM), Gen. Mgr., Natl. Hardware Specialties Ltd.; *for mail*, Box F, Wallaceburg, Ont., Can.
McGorman, William Gordon (J'45), 511 Devonshire Rd., Windsor, Ont., Can.
McGowan, Frank Kluber (J'42) (CMJ), Asst. Supt. Prod. Engrg., Carl L. Norden, Inc., 80 Lafayette St., New York, N.Y.; *for mail*, 47 Palisade Rd., Elizabeth, N.J.
McGowan, Joe Junior (J'44) (ACK), Research Engr., Douglas Aircraft Co., Inc., Santa Monica; *for mail*, 1865 Midvale Ave., Westwood Village, Calif.
McGrath, Philip Cushing ('28; '41) (LKHFEY), Sr. Engr., C. F. Braun & Co., 1000 S. Fremont Ave., Alhambra, Calif.
McGraw, Jas. H. ('26), Honorary Pres., McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York, N.Y.
McGraw, Jas. H., Jr. (A'38), Pres., McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York, N.Y.
McGraw, John T. (J'38), 5204 Village Green, Los Angeles, Calif.
McGraw, William James (J'43) (KCH), Ensign, U.S.N.R., U.S.S. *Hazard*, AM-240, F.P.O., New York, N.Y.; *for mail*, 1722 Walnut St., Berkeley 4, Calif.
McGreen, Thomas Campbell (J'43) (AJSBE), Instr., Mech. Engrg., Brown Univ., Prospect St., Providence 12; *for mail*, 49 Methyl St., Providence 6, R.I.
McGregor, A. Grant ('05; '12), Cons. Engr., Ministry of Aircraft Prod., Selection Trust Bldg., Mason's Ave., London, E.C. 2, England.
McGregor, Duncan G. ('37) (CJM), Col., Hdq., Xth Army Corps, A.P.O. 5731, c/o Postmaster, San Francisco, Calif.
McGregor, Harold Garland (J'44) (CJKN), Estimator & Ch. Engr., Cent. Boiler & Mfg. Co., 5818 Rivard St., Detroit 11; *for mail*, 1432 Vernier Rd., Grosse Pointe 30, Mich.
McGregor, Howard Leadley ('21; '35), Pres., Natl. Twist Drill & Tool Co., Rochester, Mich.
McGrew, J. A. ('16), Retired; Cons. Engr., 706 Madison Ave., Albany 3, N.Y.
McGuckian, Joseph Patrick (J'31) (CTM), Asst. Mgr., Cost Dept., Am. Woolen Co., 225—4th Ave., New York 3, N.Y.
McGuinness, John Peter (J'38) (BCDJMN), Ch. Engr., Advance Stamping Co., 7075 Lyndon St., Detroit 21; *for mail*, 21400 Bennett, Detroit 19, Mich.
McGuire, Donald E. ('34), Asst. to Gen. Works Mgr., Great Lakes Steel Corp., Ecorse; *for mail*, 22247 Long Blvd., Dearborn, Mich.
McGuire, Erwin John (J'29) (MCYJ), Buyer, Camera Works, Eastman Kodak Co., 333 State St., Rochester 4; *for mail*, 54 Roosevelt Rd., Rochester 10, N.Y.
McHale, Walter L. ('29; '35) (FOS), c/o Southland Paper Mills, Inc., Lufkin, Tex.
McHugh, Edw. (J'39) (DHJ), Instr. in Mech. Engrg., Clarkson College of Tech., Potsdam, N.Y.
McIlhenney, William ('13) (YBLSNK), Project Engr., E. I. du Pont de Nemours & Co., 626 Schuyler Ave.; *for mail*, 149 Linden Ave., Arlington, N.J.
McIlhiney, James Gregory (J'35), Sales Engr., Air Reduction Sales Co., Box 590, Tulsa 1, Okla.
McIlvaine, John H. (A'27) (CF), Pres., Landwehr Htg. Corp., 6th & Cayuga Sts., Philadelphia; *for mail*, 601 Pembroke Rd., Bryn Mawr, Pa.
McIlvaine, Robert L. ('39) (CDJ), Ch. Engr., Sales Mgr., Natl. Engrg. Co., 549 W. Washington St., Chicago; *for mail*, 368 Jackson Ave., Glencoe, Ill.
McIlveen, Edward E. (J'43) (JSY), Research Engr., Okonite Co., Passaic St., Passaic, N.J.
McInerney, Frank Thomas, Jr. (J'37) (ABM), Design Engr., Gen. Elec. Co., 1635 Broadway; *for mail*, 1933 Bayer Ave., Ft. Wayne 3, Ind.
McInerney, Fulton Clement (J'44) (NHKZAE), Mech. Engr., Rm. 244, Bldg. 285, Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
McIntosh, David C. ('39) (CMDGLT), Mgr., Indus. Engrg., Marshall Field & Co., 121 N. State St., Chicago 90; *for mail*, 803 Hinman Ave., Evanston, Ill.
McIntosh, Donald Grant (J'42) (ZLK), Engr., Ont. Hughes-Owens Co. Ltd., 3 Hamilton Ave.; *for mail*, 303 Cambridge St., Ottawa, Ont., Can.
McIntosh, Robert ('13), Mech. Supt., Calumet & Hecla Consltd. Copper Co., Calumet; *for mail*, Lake Linden, Mich.
McIntosh, William Gourlay ('30) (NJDMLB), Assoc. Prof. Mech. Engrg., Mech. Bldg., Univ. of Toronto, Toronto 5, Ont., Can.
McIntosh, William James ('36) (EBHRA), Mgr., Diesel Eng. Dept., Fairbanks, Morse & Co., 1226—1st Ave. S., Seattle 4; *for mail*, 2811—32nd Ave. S., Seattle 44, Wash.
McIntyre, Donald Frederick (J'46) (COS), Jr. Engr., Mech. Engrg. Dept., Philadelphia Elec. Co., 900 Sansom St.; *for mail*, 5139 Hazel Ave., Philadelphia, Pa.
McIntyre, Dougal Davidson ('45) (SKCJBO), Dist. Engr., Erection Dept., Alton & Co. Ltd., Stores Rd., Derby; *for mail*, 46 Springfield Rd., Thornton Heath, Surrey, England.
McIntyre, Harry John ('27; '30) (SEC), Prof. Mech. Engrg., Univ. of Wash., Seattle 5, Wash.
McIntyre, James K. ('40) (HBJLNS), Engr., Dean Bros. Pumps Inc., 323 W. 10th St., Indianapolis 7; *for mail*, 3435 N. Penn St., Indianapolis 5, Ind.
McIntyre, Malcolm ('18) (DYWARS), V.P., Mech. Engr., Bergen Point Iron Works, W. 5th St., Bayonne, N.J.; *for mail*, Cedar Cliff Rd., Riverside, Conn.
McIntyre, Roger (J'40) (RJE), Design Engr., Loco. Dept., Gen. Elec. Co., 2901 Lake Rd., Erie, Pa.
McIntyre, Wm. S. (J'36), 13 Artwill St., Milton, Mass.
McKaig, Alvin Williston ('44) (ABDNRY), Div. Mgr., Fabreeka Products Co., 16 S. Broad St., Philadelphia 2; *for mail*, 2331 N. 58th St., Philadelphia 31, Pa.
McKaig, Wm. Wallace (A'14) (BJN), Ch. Engr., Cumberland Steel Co., Cumberland, Md.
McKay, Donald B. (J'38) (AJM), Chief Machinist's Mate, C.P.O. Mess, Naval Trng. Center, San Diego, Calif.
McKay, Francis Joseph (J'44) (AJBHY), 362 Eastern Ave., East Lynn, Mass.
McKay, Juan Bautista (J'36) (CORMKH), Gen. Mgr., Tecnica Nacional, S.A., Apartado 1764; *for mail*, Apartado 85, Panama, Republic of Panama.
McKay, William Edgar (J'45) (CMD), Mem., Spec. Serv. Staff, Indus. Engr., Ernst & Ernst, 231 S. La Salle St., Chicago, Ill.
McKean, C. V. (J'44) (FJ), Box 996, Baytown, Tex.
McKean, Robert Knight (J'41) (BJC), Metallurgist, Homestead Works, Carnegie-Ill. Steel Corp., Munhall; *for mail*, 237 N. Dithridge, Pittsburgh 13, Pa.
McKeand, Marshall L. (A'44) (CDM), Ch. Indus. Engr., Dayton Malleable Iron Co., 400 Detrick St., Dayton; *for mail*, 4417 Queens Ave., Dayton 9, Ohio.
McKee, Arthur Glenn ('07; F'43) (OEK), Pres., Dir., Arthur G. McKee & Co., Box 5608, 2300 Chester Ave., Cleveland 1, Ohio.
McKee, Neal Trimble ('07; '12; F'44) (FKRS), V.P., Superheater Co., 60 E. 42nd St., New York 17, N.Y.
McKee, Thomas Cahoon ('18) (BK), Pres., Midwest Engrg. & Equip. Co., 617 Fulton St., Chicago 6; *for mail*, 4709 Beacon St., Chicago 40, Ill.
McKee, Waldo McCutcheon ('27; '35) (SJKCL), Asst. Sales Mgr., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.
McKee, Walter Reid (J'44) (DCKS), Asst. Pur. Agt., Air Preheater Corp., 121 S. Main St.; *for mail*, 117 Jefferson St., Wellsville, N.Y.
McKee, Wayne Sherwood ('39) (NBMJ), Asst. Prof. Mech. Engrg., Carnegie Inst. of Tech., Pittsburgh 13; *for mail*, 612 Arden Rd., Pittsburgh 16, Pa.
McKelvy, Francis Graham ('22) (C), Pres., Alpha Portland Cement Co., 15 S. 3rd St., Easton, Pa.
McKendrick, Leslie ('40) (FHKLSY), Philadelphia Dist. Mgr., Foster Wheeler Corp., 2214 Packard Bldg., Philadelphia 2, Pa.
McKenna, Joseph Francis ('35; '45) (BCEFKS), Ch. Draftsman, Wickes Boiler Co., 515 Washington St., Saginaw; *for mail*, 1022 Owen St., Saginaw 23, Mich.
McKenna, Joseph Victor (J'42) (CHM), c/o Can. Pumps Ltd., Kitchener, Ont., Can.
McKenna, Philip M. ('41) (M), Pres., Kennametal Inc., 1 Lloyd Ave., Latrobe, Pa.
McKenna, Roy Carnegie ('18) (JM), Chmn. of Bd., Vanadium-Alloys Steel Co., 2408 Grant Bldg., Pittsburgh 19, Pa.
McKenney, James Francis ('36; '45) (NBJGMH), Mech. Designer, Gen. Elec. Co., 100 Woodlawn Ave.; *for mail*, Box 399, 54 Wendell Ave., Pittsfield, Mass.
McKenzie, Allan Melvin (J'41) (SJB), Mar. Engr., Piping Design, Bur. of Ships, Navy Dept., Washington 25, D.C.; *for mail*, 4307 Eastern Ave., Mt. Rainier, Md.
McKenzie, Donald Hughes (J'39) (CMLA), Pres., Jackson Bangor Slate Co., Pen Argyl; *for mail*, 61 N. 4th St., Bangor, Pa.
McKenzie, James ('42) (KSBCFM), Ch. Engr., John Inglis Co., Ltd., 14 Strachan Ave., Toronto 2; *for mail*, 294 Wright Ave., Toronto 3, Ont., Can.
McKeon, Thomas Francis (J'41) (SHKF), Asst. Engr., Sta. Effic., Commonwealth Edison Co., 1111 W. Cermak Rd., Chicago 8; *for mail*, 7015 East End Ave., Chicago 49, Ill.
McKeown, Gregory M. (J'35) (ABH), Vickers, Inc., Hibbs Bldg., Washington, D.C.; *for mail*, Apt. 158, 1817 Queens Lane, Colonial Village, Arlington, Va.
McKeown, John Andrew ('31) (FGJ), 491 Staten Ave., Oakland 10, Calif.
McKernan, Hugh J. (J'25) (FKS), Effic. Supvr., Consumers Power Co., 212 W. Michigan Ave.; *for mail*, 108—2nd St., Jackson, Mich.
McKinney, Russell Edward ('44) (NJACDM), Dist. Mgr., Indus. Div., Timken Roller Bearing Co., 8500 Woodward Ave., Detroit 2, Mich.
McKinsey, Ellis Carl (J'42) (OCKE), Mech. Engr., Sacramento Dist., U.S. Engrs., War Dept., 8th & L Sts., Sacramento; *for mail*, 316 B St., Marysville, Calif.
McKinty, James ('42), Gen. Supt., Bottling Div., P. Ballantine & Sons, 57 Freeman St., Newark, N.J.
McKinzie, Daniel Joseph ('46) (AKS), Mech. Engr., Kellex Corp., 233 Broadway, New York; *for mail*, 91 Loring Ave., Yonkers, N.Y.
McKittrick, William Kennett (A'45) (BJN), V.P., Mchy. Builders, Inc., 38-50—38th St., Long Island City; *for mail*, Bayville, L.I., N.Y.
McKnight, Chas. H. ('28), Supt. Power Plants, Scranton Elec. Co.; *for mail*, 1625 Vine St., Scranton, Pa.
McKnight, Edward Wiley (J'36) (ESF), Oper., Greenwood County Elec. Power Comm., Greenwood; *for mail*, R.F.D. 1, Chappells, S.C.
McLain, Arnold Roy ('45) (BDJLMS), Welding Engr., Combustion Engrg. Co., Inc., 1032 W. Main St., Chattanooga, Tenn.
McLain, Robert Henry ('21) (SDWO), Sales Engr., Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
McLain, William Richard (J'43) (YCL), Pres., Plastic Div., Kusan, Inc., Henderson, Ky.
McLane, Roye Miller (J'27) (AREZ), Instrument Mechanic Engr., Glenn L. Martin Co., Baltimore; *for mail*, 1706 E. Lanvale St., Baltimore 13, Md.
McLane, Warren Jay (J'41) (BKS), Serv. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 582 Ontaria St., Schenectady 6, N.Y.
McLaren, Thomas Arthur (J'41) (MWS), Asst. Shipyard Mgr., W. Coast Shipbldrs. Ltd., 205 W. 1st Ave., Vancouver; *for mail*, 2532 Haywood Ave., West Vancouver, B.C., Can.
McLarren, Robert (J'44) (ABF), Tech. Dir., Pub. Relations Staff, Aero. Chamber of Commerce of Am., Inc., 610 Shoreham Bldg., Washington 5, D.C.
McLaughlin, Edmund Fennessey ('38; '41), Woodland Dr., Pleasantville, N.Y.
McLaughlin, George Edward ('20; '30) (HLM), Mech. Engr., Turner Tanning Mchy. Co., 65 Walnut St., Peabody, Mass.
McLaughlin, John Francis ('39) (SFCK), Supt. of Power, Iowa Power & Light Co., 312—6th Ave., Des Moines 8; *for mail*, 1049—38th St., Des Moines 11, Iowa.
McLaughlin, Joseph William (J'44) (CDL), 201 E. 62nd St., New York, N.Y.
McLaughlin, Richard Allen ('40) (CSDM), Works Mgr., Poor & Co., 2025 Dueber Ave., S.W., Canton 6; *for mail*, 827 Colonial Blvd., N.E., Canton 4, Ohio.
McLaughlin, Robert James (J'39) (BJM), 36 Sleeper St., Laconia, N.H.
McLaughlin, Robert Theodore (J'44), Sgt., Hdq. & Serv. Co., 6th Engrs., A.P.O. 6, c/o Postmaster, San Francisco, Calif.
McLaughlin, W. G. (J'36) (BHW), Engr., Aiken & MacLachlan, Ltd., St. Catharines; *for mail*, Cumberland, Ont., Can.
McLauthlin, Martin Bernard ('09) (CDMN), Treas., Engr., Geo. T. McLauthlin Co., 120 Fulton St., Boston; *for mail*, 65 Rockland Ave., Malden, Mass.
McLean, George Davidson (J'44), 411 St. Johns Pl., Brooklyn 17, N.Y.
McLean, Harvey Duard (J'36) (CDJM), Mgr., Fdy. Div., J. B. Beaird Co., St. Vincents Ave., Shreveport 72; *for mail*, 3220 Samford, Shreveport 19, La.
McLean, Mark Linwood (J'44) (BN), Jr. Engr., Sylvania Elec. Products Inc.; *for mail*, 36 E. 4th St., Emporium, Pa.
McLean, Robert W. ('07; '13) (DJLMNS), Res. Mgr., United Shank & Findings Co., Branch of United Shoe Mchy. Corp., Myrtle Ave., Whitman; *for mail*, 91 Bedford St., Bridgewater, Mass.
McLean, William George (J'38) (CBYZ), Asst. to Supt., Spec. Products Dept., Eastman Kodak Co., 333 State St., Rochester 4, N.Y.; *for mail*, 333—5th Ave., Scranton 5, Pa.
McLeish, Duncan Robert (J'43) (LCYSD), Maint. Engr., Firestone Indus. Products Co., Inc., 16th & Division Sts.; *for mail*, 1665 Clinton St., Noblesville, Ind.
McLellan, Edward Alden (J'36) (DMBWSH), V.P., Mgr., F. N. Johnston, Inc., 2207 Am. Bank Bldg., New Orleans 12, La.
McLellan, John Michael (J'43), c/o Procter & Gamble Defense Corp., Milan, Tenn.
McLenegan, David Wallace ('43) (CKFBHS), Asst. to Mgr. of Engrg., Air Conditioning Dept., Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.
McLennan, J. A. ('33; '35) (BKS), 1415 S. 33rd St., Birmingham, Ala.
McLoughlin, Francis Eugene (J'37) (SCFKJ), Mech. Supt. of Constr., Tidd Constr. Dept., Ohio Power Co., Brilliant, Ohio.
McMackin, Carl A. (J'38), Residence Park, Palmerton, Pa.
McMackin, George Edgar (J'42) (FKS), Lt. (j.g.), U.S.N.R., YP-241, Navy 121, F.P.O., New York, N.Y.; *for mail*, 254 Lincoln St., Easton, Pa.
McMahan, Raymond Glen (J'33) (EFJS), Automotive Engr., Socony-Vacuum Oil Co., Inc., 17 W. Market St., Indianapolis 9; *for mail*, 6032 Carrollton Ave., Indianapolis 5, Ind.
McMahon, Chas. M. (J'33) (CDM), Ch. Draftsman, Bay State Abrasive Products Co., Union St., *for mail*, 76 Milk St., Westboro, Mass.
McMahon, Jerome B. ('31; '33) (ZLSHB), Application Engr., Republic Flow Meters Co., 2240 Diversey Pkwy., Chicago 47, Ill.
McMahon, John B. (J'44), 3900 Coleman St., Pittsburgh 7, Pa.

McMahon, Joseph Patrick, Jr. (J'42) (OKSC), P. J. McMahon, Inc.; *for mail,* 700 N. Jackson St., Media, Pa.
McMahon, Thomas Edward (J'42) (SFK), Sales Engr., Combustion Engrg. Co., Inc., 1616 Walnut St., Philadelphia; *for mail,* 108 Pilgrim Lane, Drexel Hill, Pa.
McManmon, Joseph Charles (J'31) (FLS), Engr., Power & Process, Mich. Sugar Co., 2nd Natl. Bank Bldg., Saginaw; *for mail,* 260 N. Van Buren St., Bay City, Mich.
McManus, Joseph DeLee ('28; '35) (MS), Asst. Ch. Engr., Walworth Co., Huff Ave., Greensburg, Pa.
McMaster, Alexander C. (J'45) (ADP), 305 S. Washington, Ypsilanti, Mich.
McMeekin, Bowman M. ('20; '23), Welsh Rd. above Ashton Rd., Holmesburg, Philadelphia, Pa.
McMillan, James E. (J'40) (FKM), 2nd Lt., Army Air Forces, Adjutant, 83rd Sch. Sqd., Maxwell Field, Ala.
McMillen, Albert Knox ('13) (BFJKO), Retired; 1025 Washington Rd., Pittsburgh 16, Pa.
McMinn, Bryan T. ('21; '28) (JNBSE), Prof. Mech. Engrg., Univ. of Wash. Seattle 5, Wash.
McMullen, George Clifton ('30) (CRBM), V.P., Asst. to Pres., Tyson Bearing Corp., Massillon, Ohio.
McMullen, Herbert Wilkes (J'37) (CJMDL), Lt., U.S.C.G.R.; *for mail,* 41 Oakwood Ave., Upper Montclair, N.J.
McMurray, John H. ('17; '26) (CLM), Asst. Mgr., Calco Chem. Div., Am Cyanamid Co., Box 550; *for mail,* R.D. 1, N. Mountain Ave., Bound Brook, N.J.
McMurrich, Robert Peebles (J'44) (BCLAFK), Draftsman, Grade II, Cent. Drawing Office, Ministry of Munitions, Commonwealth of Australia, Maribyrnong; *for mail,* 96 Duke St., Sunshine, Melbourne, Victoria, Australia.
McNair, F. Chaloner ('41) (CMJHSF), V.P., Charge Mfg., Taylor Forge & Pipe Works, Box 485, Chicago 90, Ill.
McNally, Keenan James ('20; '27; '35), Night Mech. Supt., New York Times Co., 229 W. 43rd St., New York; *for mail,* 125 Maple Dr., Great Neck, L.I., N.Y.
McNally, Thos. ('39) (DF), Pres., McNally Pittsburg Mfg. Co., Pittsburg, Kan.
McNamara, Roger Alton (J'44) (LDSFNK), Asst. to Mech. Engr., Consltd. Rendering Co., Atlantic Ave., Boston 10; *for mail,* Depot St., South Easton, Mass.
McNamer, Jack Dempsey (J'41) (EYFKH), Jr. Project Engr., Chrysler Corp., 12800 Oakland Ave.; *for mail,* 56 Colorado Ave., Detroit 3, Mich.
McNaughton, The Hon. Andrew G. L. (H'43) (BAHS), Gen., 28 Goulburn Ave., Ottawa, Ont., Can.
McNeal, D. Raymond ('16; '25; '35) (KLM), Pres., Gen. Mgr., Andale Co., 1740 Cherry St., Philadelphia 3; *for mail,* 308 Jericho Rd., Abington, Pa.
McNear, William Fairlie ('42; '46) (SKBNF), Instr., Dept. of Mch. Design, Stevens Inst. of Tech., 5th St., Hoboken; *for mail,* 20 Gilbert Pl., West Orange, N.J.
McNeely, George Gardiner (J'39) (EMC), Shop Engr., Mack Mfg. Corp.; *for mail,* 315 N. 7th St., Allentown, Pa.
McNeil, Merritt C. ('12) (DLM), Pres., Osgood Co.; *for mail,* 610 Girard Ave., Marion, Ohio.
McNeile, George Russell (J'41) (MJC), Exper. Design Engr., Am. Fdy. Equip. Co., Mishawaka; *for mail,* 806 E. Miner St., South Bend 17, Ind.
McNeill, Francis R. (J'43) (MJNCDL), R.D. 1, New Carlisle, Ohio.
McNeilly, Vance Hill (J'44) (BZNO), Ord. Engr., Ballistic Research Lab., Aberdeen Proving Ground; *for mail,* R.F.D. 2, Aberdeen, Md.
McNellis, Ralph Stewart (J'45) (BEKHNM), Project Engr., Contract Estimator, John Inglis Co.; *for mail,* 179 Eglinton Ave. E., Toronto, Ont., Can.
McNew, John T. L. ('45) (COHB), V.P. for Engrg., Agric. & Mech. College of Tex., College Station, Tex.
McNickle, Robert C. (J'41) (ZSL), Engr., Gen. Mgr., Brooke Engrg. Co., Inc., 4517 Wayne Ave., Philadelphia 44, Pa.
McNulty, Dwight L. ('46) (JLY), Mfrs.' Agent, Box 1066, Chattanooga, Tenn.
McPhee, Alexander Hector ('34; '43) (ONJBHD), Partner, McPhee & Johnston, 90 West St., New York 6, N.Y.
McPherson, Harry Jay ('44), 801 Dwight Bldg., 1004 Baltimore, Kansas City 6, Mo.
McPherson, John Alexander ('21) (DLMSTY), Pres., John A. McPherson Co., S. Main St.; *for mail,* Poinsett Hotel, Greenville, S.C.
McPheters, Leonard Lamont (J'44) (BEGJM), Ord. Engr., Springfield Armory, Springfield; *for mail,* 30 Seymour Ave., Springfield 9, Mass.
McPheters, Linwood Snider (J'44) (MCDKN), Asst. to Shop Supt., Air Preheater Corp., 121 S. Main St.; *for mail,* 143 W. State St., Wellsville, N.Y.
McQuaid, Dan J. ('26) (BES), 1026—17th St., Denver, Colo.
McQuaide, Louis Bernard, Jr. (J'39) (FLNZSR), Engr., Tech. Plant Research Sec., E. I. du Pont de Nemours & Co., Box 993, Charleston 24; *for mail,* 4002 Venable Ave., Charleston 4, W.Va.
McQuilkin, George, Jr. ('44), 22 Franklin Ave., Merchantville, N.J.
McQuillan, John ('17; '35) (CMBS), Div. Engr., Reconstr. Finance Corp., Rm. 720, 70 Pine St., New York; *for mail,* 5 W. 63rd St., New York 23, N.Y.
McQuiston, W. Bryce ('26; '37) (SEFKO), Sales Engr., Bigelow-Liptak Corp., 310 Renshaw Bldg., Pittsburgh 22, Pa.
McRae, Robert C. (J'41) (CLNW), Mgr., Coast Iron & Mch. Works, 1131 Dock St., Tacoma 2, Wash.
McRee, Kenneth O. (J'41), Ensign, 119 Coast Blvd., La Jolla, Calif.
McSorley, Donald Chrysler (J'38) (EJMBAC), *Student Award,* '38; Lt. (j.g.), U.S.N.; *for mail,* Apt. 7A, 43 E. 27th St., New York 16, N.Y.
McSweeney, Jeremiah (J'35), 1319 Loring St., San Diego 9, Calif.
McSweeney, William Timothy ('31; '35; '36) (SFKHCB), Power Plant Shift Oper., Caribbean Petroleum Co. & Venezuelan Oil Concessions, Ltd.; *for mail,* c/o Venezuelan Oil Concessions, Ltd., Pueblo Viejo, Lagunillas, Estado Zulia, Venezuela, S.A.
McVeigh, John Ross ('46) (ACO), Project Engr., Continental Aviation & Engrg. Corp., 12801 E. Jefferson, Detroit 14; *for mail,* 15515 Glenwood, Detroit 5, Mich.
McVetty, Percy George ('20; '21) (JBSEYZ), Mech. Engr., Research Labs., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 1023 La Clair Ave., Pittsburgh 18, Pa.
McVicar, Angus Earl (J'40) (CMJSE), Process Engr., Torrance Brass Fdy., 1825 W. 213th St.; *for mail,* 2562 Torrance Blvd., Torrance, Calif.
McVicker, Thomas E. (J'39) (ESKFHN), Worthington Pump & Mchy. Corp., Wellsville, N.Y.
McWane, Gerould Robert (J'36) (BJ), Plant Engr., Sandusky Fdy. & Mch. Co., Market & Fulton Sts., Sandusky, Ohio.
McWhorter, Marshall Jeter ('23; '25; '35), Mgr., Atlanta Office, Bailey Meter Co.; *for mail,* 75½ Hunter St., S.W., Atlanta, Ga.
McWorkman, Delamar ('45) (CDO), Box 591, Eustis, Fla.
Mead, Charles Adriance ('13) (RWYDHJ), 165 Wildwood Ave., Upper Montclair, N.J.
Mead, Daniel W. ('08), Cons. Engr., 550 State St., Madison 3, Wis.
Meade, Harold Edward ('44) (SKZLFE), Exec. V.P., New Orleans Pub. Serv. Inc., 317 Baronne St., New Orleans 9, La.
Meade, Joseph Frederick ('43) (ABCDJM), Pres., Gen. Mgr., Mercury Aircraft Inc., Wheeler Ave.; *for mail,* 24 Main St., Hammondsport, N.Y.
Meade, Leonard Price ('32; '46) (EK), Engrg. Dept., Phillips Petroleum Co.; *for mail,* 2050 Dewey Ave., Bartlesville, Okla.
Meaden, John Alexander, Jr. (J'43) (BCM), Meaden Screw Products Co., 2310 S. Kedzie Ave., Chicago, Ill.
Meador, Barclay F. ('39) (FGHORS), Chief, Relocation Branch, Office of Chief of Engrs., U.S.A., 21st & Virginia Sts., Washington; *for mail,* 1742 Kilbourne Pl., N.W., Washington 10, D.C.
Meadors, Jack Williamson, Jr. (J'45) (CEO), c/o A. H. Guion & Co., Box 1188, Charlotte, N.C.
Meagher, Robert Douglas (J'45) (EFJKLS), Asst. Engr., British Am. Oil Co., Ltd., 703 Royal Bank Bldg., Toronto, Ont., Can.
Mealand, Alfred ('29) (CGM), c/o Dept. of Munitions, 83 William St., Melbourne, C. 1, Victoria, Australia.
Meals, Russell William (J'41) (ACJ), Serv. Mgr., Grand River Chevrolet Co., 5100 W. Grand River, Detroit 8, Mich.
Means, Herbert Enfield (J'44) (EACRLH), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp.; *for mail,* Box 224, East Hartford, Conn.
Meany, James M. ('17; '21) (DEJC), Mgr., Logging Dept., Loggers & Contractors Mchy. Co., 240 S.E. Clay St., Portland; *for mail,* 2630 N.E. Thompson St., Portland 12, Ore.
Mears, Edward W. ('31) (CFK), Pres., Mears-Kane-Ofeldt, Inc., 1907 E. Hagert St., Philadelphia 25, Pa.
Mecklenburger, Jerome Walter (J'41) (BACNYW), Liaison Engr., Hamilton Stand. Propellers Div., United Aircraft Corp., East Hartford; *for mail,* 28 Bretton Rd., West Hartford 7, Conn.
Medbery, Edward W. (J'37) (CLMT), Maj., Signal Corps, 1 N. La Salle St., Chicago; *for mail,* 1810 Sherman Ave., Evanston, Ill.
Medici, Joseph John (J'43) (A), Lt., Design & Devel. Officer, Engrg. Div., Struc. Branch, Aircraft Lab., Hdq., Air Tech. Serv. Comd., Area B, TSEAC-12E, Wright Field, Dayton, Ohio.
Medlar, Lewis Abbott (J'41) (NBJO), Mech. Engr., Fox Products Co., 4720 N. 18th St., Philadelphia; *for mail,* 14 Rose Lane, Philadelphia 18, Pa.
Medley, Harold Cockrell ('43) (BDFHNR), Coal Preparation Engr., Jeffrey Mfg. Co., Columbus; *for mail,* 345 Crestview Rd., Columbus 2, Ohio.
Medlin, John Bodey (J'43) (ABEN), Ensign, U.S.N.R., Boeing Aircraft Co., Seattle; *for mail,* 8311 Wolcott Ave., Seattle 8, Wash.
Medlin, John Wilson (J'41) (NBJK), Asst. Prof. Mech. Engrg., Univ. of Minn., Minneapolis 14, Minn.
Meeg, Alfred B. ('26; '33; '35) (FCSKE), Branch Mgr., Iron Fireman of Milwaukee, Inc., 4507 W. Wisconsin Ave., Milwaukee 13; *for mail,* 130 N. 87th St., Wauwatosa 13, Wis.
Meehan, James ('33; '35) (ACJMN), Sales Engr., Brown & Sharpe Mfg. Co., Providence 1, R.I.
Meek, George William (J'32) (CJL), 408 State Tower Bldg., Syracuse 2, N.Y.
Meeks, Charles Davenport (J'46) (EKS), Forge Instr., Clemson College, Clemson; *for mail,* 145 Brown St., Anderson, S.C.
Meenen, R. J. A. ('40), 6525 Regent St., Philadelphia, Pa.
Mees, Robert T. ('32; '38) (CE), Asst. to Dir. of Research, Caterpillar Tractor Co., Peoria 8; *for mail,* 303 Barker Ave., Peoria 5, Ill.
Meguire, Kenneth Urquhart (J'45) (CEB), Charge Export Div., Enterprise Eng. & Fdy. Co., 18th & Florida Sts., San Francisco 10; *for mail,* 1200 Emerson St., Palo Alto, Calif.
Mehagan, John Greenleaf (J'43) (MWACDR), Ensign, U.S.N.; 330 S. 8th Ave., La Grange, Ill.
Meharg, Laurence ('42) (BDNS), Ch. Engr., Hazel Atlas Glass Co., 15th & Jacobs Sts.; *for mail,* 36 Rockledge Rd., Wheeling, W.Va.
Mehmel, Louis Ernest (J'28), Asst. to Head of Mar. Engrg. Dept., Foster Wheeler Corp., 165 Broadway, New York; *for mail,* 4 S. Sycamore St., West Hempstead, N.Y.
Mehringer, Frank John (J'40) (BJYZAN), Instr. in Mech. Engrg., Mass. Inst. of Tech., Cambridge 39, Mass.
Mehta, Manvantrai Dayalal (A'45) (CEJMRZ), Managing Dir., Motor House (Gujarat) Ltd., Mirzapur Rd., Ahmadabad, India.
Meier, Charles Kiss (J'41), Ensign, U.S.N.R.; *for mail,* 83-44 Lefferts Blvd., Kew Gardens, N.Y.
Meier, George Henry (J'43) (AEFYSJ), Tool Designer, Bell Aircraft Corp., Marietta; *for mail,* R.F.D. 1, Smyrna, Ga.
Meier, Joseph B. ('40) (CJMO), Ch., Castings Sec., Metals Branch, Office of Price Admin., 2nd & D Sts., S.W., Washington 25; *for mail,* 1369 Monroe St., N.W., Washington 10, D.C.
Meier, William George (J'43), Mech. Engr., 416 S. Barnes St., Mason, Mich.
Meile, Carl H. (J'41) (BAEC), Project Engr., Stress Analysis, Continental Aviation & Engrg. Corp., 12801 E. Jefferson St., Detroit 14; *for mail,* 3763 Richton St., Detroit 6, Mich.
Meiners, Eugene Bernard (J'45) (HMS), Research Engr., Research Dept., Wrought Iron Range Co., 5661 Natural Bridge Rd., St. Louis; *for mail,* 1419 Hamilton Ave., St. Louis 12, Mo.
Meiners, Richard Henry ('44) (CJM), Ch. Engr., A. J. Lindemann & Hoverson Co., 601 W. Cleveland Ave., Milwaukee 7; *for mail,* 2510 N. 86th St., Wauwatosa 13, Wis.
Meinhart, Fred John (J'42) (SB), Mech. Engr., Leslie Co., Lyndhurst, N.J.; *for mail,* 133-10 Francis Lewis Blvd., Laurelton, L.I., N.Y.
Meiselman, Sumner (J'40) (AEH), Tank Dept., TK-5, Armd. Force Sch., Ft. Knox, Ky.
Meisenzahl, Thomas W. (J'41), 695 Portland Ave., Rochester 5, N.Y.
Meissner, John F. ('30) (DJ), 423 S. Madison Ave., La Grange, Ill.
Meister, Jacque Louis (J'44) (CMN), Indus. Engr., Methods & Stands., Victor Adding Mch. Co., 3900 N. Rockwell St., Chicago 18; *for mail,* 4838 W. Warner Ave., Chicago 41, Ill.
Mekler, L. A. ('44) (LFKZ), Engrg. Consultant, Armour Research Foundation, Inst. of Gas Tech., Tech. Center, Chicago 16, Ill.
Melas, William ('25; '37) (ZHBSNL), Ch. Engr., V.P., Penn Indus. Instrument Corp., 3116 N. 17th St., Philadelphia 32; *for mail,* 6301 Sherwood Rd., Philadelphia 31, Pa.
Melcher, Conway Hillman (J'42) (AJS), 228 Park St., Montclair, N.J.
Melczer, Henry Lynn (J'43) (FAS), Jr. Research Engr., Shell Devel. Co., 4560 Horton St., Emeryville; *for mail,* 1720 Short St., Berkeley 2, Calif.
Mele, Thomas William (J'36) (ACM), Pvt., Army Air Forces, 4000th Base Unit, Sec. BP, Wright Field, Dayton, Ohio.
Melendez, Victor Emmanuel (J'43) (CDEHOS), Mgr., Equipos Mecanicos de Guatemala, S.A., Apartado 472, Guatemala, Guatemala, C.A.
Melendy, Jesse G. ('42) (LMKJDF), Chem. Engrg., Gen. Chem. Co., 40 Rector St., New York 6; *for mail,* 56 Grove St., Tarrytown, N.Y.
Meleney, Robert C., Jr. (J'39) (AES), Capt.; 720 Mills Bldg., Washington, D.C.
Melichar, Joseph Thomas (J'33) (JMNASW), Prod. Engr., Thomson Industries, Inc., 29-05 Review Ave., Long Island City 1; *for mail,* 41-37—67th St., Woodside, L.I., N.Y.
Melick, Neal A. ('20), 2506 Watrous Ave., Tampa 6, Fla.
Melin, Hans E. ('38) (BHJ), Asst. Sales Mgr., Wean Engrg. Co., 2nd Natl. Bank Bldg., Warren; *for mail,* 4037 Windsor Rd., Youngstown, Ohio.
Mellanby, Alex. L. ('26), Westwood, Bridge of Weir, Renfrewshire, Scotland.
Mellett, Jesse Brooks (J'40) (AMS), Lt., 5317th Air Depot (Prov.), A.P.O. 492, c/o Postmaster, New York, N.Y.
Mellon, George W. ('36; '41) (MRS), 52—8th Ave., Westwood, N.J.
Mellor, Charles ('40) (ZFKSWH), Managing Dir., Mellor-Goodwin, Sociedad de Responsibilidad, Ltda., 4 piso, Paseo Colon 221, Buenos Aires, Argentina, S.A.
Mellor, Clarence Leonard ('42) (ACMNOR), Gen. Mgr., Barco Mfg. Co., 1801 Winnemac Ave., Chicago 40, Ill.

Melrose, George Bert, Jr. (J'45) (ABHENG), Exper. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., Main St., East Hartford, Conn.
Melrose, Robbie George Rubidge ('29) (RCBE), Acting Ch. Mech. Engr., Cent. Sta., Cent. Uruguay Ry. Co.; *for mail,* English Club, Calle 25 de Mayo 506, Montevideo, Uruguay, S.A.
Meltzer, Joseph (J'46) (BWY), 2759 Barker Ave., New York 67, N.Y.
Melville, Thomas (J'43) (BZ), Testing Engr., Am. Steel & Wire Co., 4300 E. 49th St., Cleveland; *for mail,* 1260 E. 115th St., Cleveland 8, Ohio.
Melzig, Alfred H. J. ('23), Method & Equip. Overseer, Singer Mfg. Co., Elizabethport; *for mail,* 277 E. 2d Ave., Roselle, N.J.
Memory, Nichol Harding ('30), Asst. to Pres., Dir. of Admissions, Stevens Inst. of Tech., 5th & Hudson, Hoboken; *for mail,* 85 Cedar St., Maplewood, N.J.
Mendelson, Alexander (J'45), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport; *for mail,* 11123 Ada Ave., Cleveland, Ohio.
Mendelson, Irving (J'42) (JAHSBE), Sr. Test Engr., Wright Aero. Corp., Beckwith Ave.; *for mail,* 653—11th Ave., Paterson, N.J.
Mendelson, Ralph R. (J'39) (C), 135 N. Lafayette, Grand Rapids 3, Mich.
Mendenhall, Earl ('28; '31) (BCDGMN), V.P., Gen. Mgr., Sterling Elec. Motors, Inc., 5401 Anaheim-Telegraph Rd., Los Angeles 22; *for mail,* 2779 N. Gainsborough Dr., San Marino 9, Calif.
Mendow, Frank Robert ('41) (SFHKE), Mech. Engr., A. M. Lockett & Co. Ltd., New Orleans 7; *for mail,* 2320 Cambronne St., New Orleans 18, La.
Mengel, John Franklin (J'44), 1560 Linden St., Bethlehem, Pa.
Mengel, Walter Alvin (J'43) (MCJDNO), Mfg. Engr., Sperry Gyroscope Co., Inc.; *for mail,* 18 Clover Dr., Great Neck, L.I., N.Y.
Mengers, Charles F. (J'42) (AEFS), 137 Kendall Blvd., Oaklyn, N.J.
Menke, John Roger (J'45) (BCJ), Cons. Engr., 210 W. 70th St., New York 23, N.Y.
Menkes, Sherwood Bradford, III (J'42) (MJCBDA), Prod. Engr., Dept. 9764, West. Elec. Co., Inc., 300 Central Ave., Kearny, N.J.; *for mail,* 290 Riverside Dr., New York 25, N.Y.
Menner, Frederic ('19) (DO), Engrg. Sales Rep., 915 Olive St., St. Louis 1, Mo.
Menon, V. K. A. ('23; '33; '35), Cochin Govt., Trichur, Cochin State, India.
Menshon, William R. ('28; '35) (CLS), Barrett Div., Allied Chem. & Dye Corp., 40 Rector St., New York, N.Y.
Menson, John Lewis (J'37) (FKRS), Engr., Superheater Co., East Chicago, Ind.
Mensonides, Sjoerd ('22) (SE), V.P., Charge Oil Fields Div., Farrar & Trefts, Inc., 20 Milburn St., Buffalo 12, N.Y.
Mentzer, Ralph Bender (J'37) (CJMNZ), Mch. Exper. Supvr., Hamilton Watch Co., Lancaster, Pa.
Menz, Charles Newman (J'38) (LMCN), Plant Engr., Marshall Eclipse Div., Bendix Aviation Corp.; *for mail,* 190 Hoosick St., Troy, N.Y.
Menzl, Leon ('21; '24) (DKOS), Cons. Engr., 129 Front St., New York 5, N.Y.
Mercer, Aaron Loveridge, Jr. (J'42) (COMRDE), 2472 Overlook Rd., Cleveland Heights, Ohio.
Mercer, Charles Franklin ('37) (KS), Head., Dept. of Physics, Univ. of S.C., Columbia 19; *for mail,* 219 S. Waccamaw St., Columbia 48, S.C.
Mercer, George Anderson, Jr. ('23; '31) (C), Pres., Steel Products Co., Inc., Box 1007, Savannah, Ga.
Merchant, Mylon Eugene ('45) (JBM), Research Physicist, Research Dept., Cincinnati Milling Mch. Co., Cincinnati 9, Ohio.
Mercier, Harvey O. ('30) (LFK), Natl. Biscuit Co., 449 W. 14th St., New York 11, N.Y.
Mercier, Joffre Pershing (J'41) (JOCK), Asst. Supt., Dixie Mch. Welding & Metal Works, 1031 Annunciation St., New Orleans 13, La.
Mercier, Stanley M. ('36) (DCBJHG), Ch. Engr., Jeffrey Mfg. Co., Columbus 16; *for mail,* 2565 Bexley Park Rd., Columbus 9, Ohio.
Meredith, Diven (J'36) (KMCS), Gen. Delivery, Lockeford, Calif.
Meredith, Henry H., Jr. (J'39) (JLM), Dist. Petroleum Engr., Humble Oil & Refining Co., Encino, Tex.
Meredith, Wynn ('13) (BCE), Res. Partner, Sanderson & Porter, 1 Montgomery St., San Francisco, Calif.
Meriam, James Lathrop (J'42) (BNMZGJ), Asst. Prof. Mech. Engrg., Dept. of Engrg., Univ. of Calif., Berkeley 4, Calif.
Merkin, Eugene (J'45), Stress Analyst, Republic Aviation Corp., 42 Broadway, New York; *for mail,* 3360 Bronx Blvd., New York 67, N.Y.
Merkle, Richard William ('24; '40) (CDJM), Asst. Prod. Mgr., West. Cartridge Co., East Alton; *for mail,* 112 Springer Ave., Edwardsville, Ill.
Merl, Milton Francis ('28; '34; '36) (FLSZ), Sales & Serv. Engr., Republic Flow Meters Co., 213 Red Rock Bldg., Atlanta 3, Ga.
Merrell, Frank Laird (J'36) (CMWYFR), Sgt. Maj., Adj. Gen. Dept., Hdq. & Hdq. Det., 560th QM. Gp., A.P.O. 887, New York, N.Y.; *for mail,* 3401 Fremont Ave. S., Minneapolis 8, Minn.
Merrell, Spencer Atkins ('44), V.P., Mgr., Waterways Transport, Inc., 16-D Railway Exchange Bldg., St. Louis, Mo.
Merriam, Carroll Fuller ('16; '21; '24) (H), Spec. Studies, Pa. Water & Power Co., Baltimore 3, Md.
Merriam, Henry Parker ('87; '96), Retired; Hubbardston, Mass.
Merriam, Kenneth Gerald ('23; '30; '35) (ZBAHN), Prof., Aeromechanics, Worcester Poly. Inst.; *for mail,* 56 Havelock Rd., Worcester, Mass.
Merrick, Charles M., III ('26; '36) (CMADJL), Indus. Engr., Mgmt. Engrg. Dept., Glenn L. Martin Co., Baltimore 3; *for mail,* 609 Allegheny Ave., Towson 4, Md.
Merrill, Bennett Wallace (J'44) (NMJCAD), Spec. Engr., Maint., Carnegie-Ill. Steel Corp., 3426 E. 89th St., Chicago 17; *for mail,* 8216 Harper Ave., Chicago 19, Ill.
Merrill, Carle J. ('16), Treas., Mgr., C. J. Merrill, Inc., 54 St. John St., Portland, Me.
Merrill, David R. ('35) (LEY), Devel. Dept., Rohm & Haas Co., Inc., Box 219, Bristol, Pa.
Merrill, Donald G. (J'19) (FKL), Devel. Engr., Hartford-Empire Co., P.O. Drawer 1620, Hartford 1; *for mail,* 534 Fern St., West Hartford 7, Conn.
Merrill, Geo. H. ('98; '17), Pres., Merrill Bros., 56-02 Arnold Ave., Maspeth, L.I., N.Y.
Merrill, Robert Dodge (J'45), Engr., Naval Research Lab., Anacostia Sta., Washington; *for mail,* Apt. 3, 10 Danbury St., S.W., Washington 20, D.C.
Merrill, S. Clifford ('18; '23; '30) (CBAM), Asst. Gen. Mgr., Automotive Div., Timken Roller Bearing Co., Inc., 8500 Woodward Ave., Detroit 2, Mich.
Merritt, George Irvin (J'43), 359—3rd St., Newburgh, N.Y.
Merritt, Harold W. (J'21) (BHKZ), Assoc. Prof. Physics, Cooper Union Sch. of Engrg., Cooper Sq., New York 3, N.Y.
Merritt, Jos. ('07) Pres., Treas., Mech. Engr., Hartford Spec. Mchy. Co., 287 Homestead Ave., Hartford, Conn.
Merritt, Richard Todd, Jr. (J'46) (JLZ), Tech. Asst., Carbide & Carbon Chemicals Corp., South Charleston 3; *for mail,* 28 Ruffner Ave., Charleston 1, W.Va.
Mershon, Clarence Earl (J'44) (ABJ), Ch. Draftsman, Aluminum Co. of Am.; *for mail,* 100 Ash Ave., Troutdale, Ore.
Mershon, Ralph Davenport ('00) (BAYE), 2000 Tiger Tail Ave., Miami 33, Fla.
Mervine, William G. ('43) (SECHO), Box 127, Phoenixville, Pa.
Merwin, Harry H. ('19; '35) (NBDMH), Mch. Designer, West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 254 Springfield Ave., Rutherford, N.J.
Meseroll, Vincent Francis (J'38) (JCLD), 33 N. 6th Ave., Highland Park, N.J.
Mesiboff, Milton Joseph (J'38) (SKNMBJ), Mar. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 628 Montgomery St., Brooklyn 25, N.Y.
Mesick, Benjamin Schultz ('39) (JLCBDM), Sub-Office Chief, Research & Devel. Serv., Calif. Inst. of Tech., Pasadena 4; *for mail,* 454 W. 7th St., Claremont, Calif.
Mesinger, Frederick W. ('24; '29), V.P., Norma-Hoffmann Bearings Corp., Stamford; *for mail,* Glenbrook, Conn.
Messaros, Frank Charles ('40) (SCFN), Sales Mgr., Mar. Div., Am. Engrg. Co., Aramingo & Cumberland Sts., Philadelphia 25; *for mail,* 1128 E. Price St., Philadelphia 38, Pa.
Messenger, Joseph Alden ('38) (KLS), V.P., Buell Engrg. Co., Inc., 70 Pine St., New York 5, N.Y.
Messenger, Paul B. (J'41) (ABM), Cardington, Ohio.
Messenger, Robert P. ('13; '31) (NMOC), V.P., Internatl. Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Messer, Eugene W. (J'41) (SEBAJ), c/o T. J. Laird, R.D. 1, Muncy, Pa.
Messer, John Dickinson (J'43) (KBE), Mech. Engr., Engrg. Dept., Weyerhaeuser Timber Co., Tacoma Bldg.; *for mail,* 5 California, Lincoln Heights, Tacoma, Wash.
Messer, Rowland E. (J'36) (WSCMHK), Assoc. Mar. Engr., Puget Sound Naval Shipyard; *for mail,* Box 2512, Route 4, Bremerton, Wash.
Messersmith, Charles William ('33; '37) (SEZKF), Prof., Exper. Mech. Engrg., Purdue Univ., West Lafayette, Ind.
Messersmith, Eldon Maurice ('37; '45) (DCMOE), Asst. to Sales Mgr., Robison Mchy. Co., 245 W. South Temple St., Salt Lake City; *for mail,* 868 N. 2nd W., Lehi, Utah.
Messick, William Wallace (J'43) (HKS), Lt., U.S.N.R., 816 S. Lake Ave., Pasadena, Calif.
Messinger, Joe Prout (J'21) (ACEF), Sales Engr., Stand. Oil Co. of N.J., 26 Broadway, New York 4, N.Y.
Messner, Manfred ('13; '17; '31) (OKFHCA), Ch. Engr., Treas., Bing & Bing, Inc., 119 W. 40th St., New York 18; *for mail,* 2 N. Clover Dr., Great Neck, L.I., N.Y.
Metcalf, Geo. R., Jr. ('16; '26), V.P., Erie Malleable Iron Co.; *for mail,* 214 W. 9th St., Erie, Pa.
Metcalf, Robert Fisher, Jr. ('43; '44) (CJMLHB), Maj., Ord. Dept.; 390 Charles River Rd., Watertown 72, Mass.
Metcalf, Stanley C. ('35) (LKZ), Designing Engr., Universal Oil Products Co., 310 S. Michigan Ave., Chicago 4; *for mail,* 410 Holly Ave., Elmhurst, Ill.
Metcalfe, Gordon Stuart (J'45) (ABJ), Design Engr., Cowles & Co., Inc., 612 N. Michigan Ave., Chicago; *for mail,* 3527 S. Damen Ave., Chicago 9, Ill.
Mettler, Albert Jacob, Jr. (J'44) (LRSJOC), Maint. Engr., Foreman, Hawaiian Tuna Packers Ltd., Kewalo Basin, Honolulu; *for mail,* 258-C Ohua Ave., Honolulu 30, T.H.
Metz, Carl Louis (J'44), Lt. (j.g.), Engr. Officer, U.S.N.; *for mail,* 415 Shelby Pkwy., Louisville 3, Ky.
Metz, Doan Elvan (J'37) (DHS), 1st Lt., A.U.S.; 1445 Fernside Blvd., Alameda, Calif.
Metz, Donald Theodore ('44) (HMP), Sales Engr., Byron Jackson Co., Box 2017, Terminal Annex, Los Angeles; *for mail,* 323 N. Cloverly Ave., Temple City, Calif.
Metz, George Wesley ('44), Sec. Mgr., Hdq., Mfg. Engrg., Dept. 5G34, Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 543 Coursin St., McKeesport, Pa.
Metz, George William ('44) (CKS), Ch. Engr., Garment Center Capitol, Inc., 512—7th Ave., New York 18; *for mail,* 667 E. 232nd St., New York 66, N.Y.
Metz, Walter Richard ('02; '11) (SFOE), Appraisal Engr., Veterans Admin., Washington 25; *for mail,* 14 Crescent Pl., Takoma Park, Washington 12, D.C.
Metzger, Herbert A. ('37) (SMBZJH), Owner, J. Metzger Co., 2165 Spring Grove Ave., Cincinnati 14, Ohio.
Metzger, John Emil (J'41) (MNZ), Project Engr., Eagle Pencil Co., 110 E. 14th St., New York; *for mail,* 82-15 Britton Ave., Elmhurst, L.I., N.Y.
Metzner, Bruno C. ('27; '29; '35) (BKA), Research Librarian, Stand. Oil Devel. Co., Box 243, Elizabeth; *for mail,* 807 West End Pl., Cranford, N.J.
Metzner, Maxwell Wallace ('44) (REZ), Head, Test Sec., Gen. Elec. Co., E. Lake Rd.; *for mail,* 1853 E. Lake Rd., Erie, Pa.
Meyer, A. William ('25; '32; '35) (BCGHMN), Asst. Dir. of Design, Brown & Sharpe Mfg. Co., Providence 1; *for mail,* 16 Washington St., Warren, R.I.
Meyer, Adolf ('44) (NSCBEA), *Calvin W. Rice Lecturer;* Managing Dir., Engr. in Chief, Brown Boveri & Co., Ltd., Baden, Switzerland.
Meyer, Adolph Frederick ('21) (HOW), Cons. Hyd. Engr., Meyer Governor Co., 284 Baker Bldg., Minneapolis 2, Minn.
Meyer, Alfred H. ('44) Exec. Dir., Secy., Engrg. Societies Personnel Serv., Inc., 8 W. 40th St., New York 18, N.Y.
Meyer, Arnold I. (J'36) (JDM), Mech. Design Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 3783 Wellington Rd., Los Angeles 16, Calif.
Meyer, Carl William (J'44) (ACY), Sr. Draftsman, Glenn L. Martin Co., Baltimore; *for mail,* 623 N. Bentalou St., Baltimore 16, Md.
Meyer, Charles A. ('40) (SLFHZ), Jr. Engr., E. I. du Pont de Nemours & Co., Belle; *for mail,* 1023 Stadium Pl., Charleston 1, W.Va.
Meyer, Charles Aloysius (J'33) (AB), Thermodynamic Sec. Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester, South Philadelphia; *for mail,* 303 Eldon Ave., Drexel Hill, Pa.
Meyer, Ernst B. (J'37), Time Study, Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 519 Halsey Ave., Pittsburgh 21, Pa.
Meyer, Erwin C. ('11; '15; '35) (CDLSZ), Plant Mgr., Pure Food Factory "Hansa," 746 Mamaroneck Ave., Mamaroneck; *for mail,* 44 Woodbine Ave., Larchmont, N.Y.
Meyer, Frank Louis ('27; '40) (FKCMJA), Pres., Meyer Furnace Co., Peoria 2, Ill.
Meyer, Frederick John ('44) (BCT), Reinforced Plastics Specialist, Owens-Corning Fiberglas Corp., Nicholas Bldg.; *for mail,* 4330 Beverly Dr., Toledo, Ohio.
Meyer, Gene Lee (J'44) (KABZHN), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 2345 Euclid Heights, Cleveland Heights 6, Ohio.
Meyer, Harold F. (J'23), Engr., Vacuum Oil Co., Cairo, Egypt.
Meyer, Henry C., Jr. ('94; '03), Pres., Meyer, Strong & Jones, Inc., 101 Park Ave., New York, N.Y.
Meyer, Henry C. E. ('21), Gibbs & Cox, Inc., 1 Broadway, New York, N.Y.
Meyer, Ira Leonard ('35; '45) (BKS), Bur. of Ships, Code 445, Navy Dept.; *for mail,* Apt. 13, 2950—2nd St., S.E., Washington, D.C.
Meyer, John K. (J'36), Maint., Corhart Refractories Co., 16th & Lee Sts., Louisville, Ky.
Meyer, Joseph Andrew ('23) (N), Propr., J. A. Meyer Engrg. Co., 427 Peshine Ave., Newark 8, N.J.
Meyer, Lawrence William (J'37) (ACJO), Tube Div. Engr., Aluminum Co. of Am., New Kensington, Pa.

Meyer, Peter ('20; '32), Owner, Meyer Mch. & Tool Co., 59 McWhorter St., Newark, N.J.
Meyer, Richard E. (J'40) (MCE), Ord. Engr., Ord. Dept., 4C-440 The Pentagon, Washington; *for mail,* Apt. 203, 2515 K St., N.W., Washington 7, D.C.
Meyer, Robert B. (J'40) (JAM), Process Engr., Pacific Div., Bendix Aviation Corp., 11600 Sherman Way; *for mail,* 5617 Hinds Ave., North Hollywood, Calif.
Meyer, Robert Max (J'45) (SKBHEA), Engr., Thermodynamics Branch, Research Sec., U.S. Maritime Comm., Washington 25, D.C.; *for mail,* 1823 N. Stafford St., Arlington, Va.
Meyer, Royal L. ('25; '35), Ch. Engr., Asst. Mgr., Swartwout Co., 18511 Euclid Ave., Cleveland; *for mail,* Route 1, Willoughby, Ohio.
Meyer, Vaughan Benjamin (J'41) (CMN), Lt., U.S.N.R., Asst. to Officer-in-Charge, Design Dept., Naval Gun Factory, Washington, D.C.
Meyer, Walter E. (J'41) (BCM), 3221 N. 24th Pl., Milwaukee, Wis.
Meyerdick, Clarence Edward ('45) (HMO), U.S. Engrs., War Dept., Wright Bldg., Sacramento; *for mail,* Box 104, Route 1, Sunset Ave., Fair Oaks, Calif.
Meyerhaus, Robert Hall (J'42) (KMDA), Design Engr., Rm. Cooler Sec., Air Conditioning Dept., Gen. Elec. Co., 5 Lawrence St.; *for mail,* 238 Baldwin St., Bloomfield, N.J.
Meyers, Alvin W. (J'46), Methods Engr., Gen. Elec., X-Ray Corp., 2012 W. Jackson Blvd., Chicago, Ill.; *for mail,* 602 W. Pearl St., Batesville, Ind.
Meyers, Dale John (J'44), Ont. Apt. 306, 264 W. Market, Akron 3, Ohio.
Meyers, Donald N. (J'40) (ABJNHM), Ch. Mech. Engr., P-V Engrg. Forum, Inc., Sharon Hill; *for mail,* 6524 N. 13th St., Philadelphia 26, Pa.
Meyers, Edward Clarence (J'33), Rotary Helper, Gen. Petroleum Corp. of Calif., 108 W. 2nd St., Los Angeles 12; *for mail,* 1619 Stanley Ave., Long Beach, Calif.
Meyers, Frederick H. ('25; '31; '35) (SKFEZB), Dist. Mgr., Republic Flow Meters Co., 3197 Linwood Ave., Cincinnati 8, Ohio.
Meyerson, Morris H. (J'29) (BACM), War Prod. Bd., Washington, D.C.; *for mail,* 80 Vassar Ave., Newark 8, N.J.
Meyerstein, Anthony ('42), Anthony M. Meyerstein, Inc., 66 Court St., Brooklyn, N. Y.
Meykar, Orest Albert ('46) (JLO), Design Engr., Shoe Hardware Div., U.S. Rubber Co., 191 Hudson St., New York 13; *for mail,* 28-17 Astoria Blvd., Long Island City 2, N.Y.
Mezger, Robert H. (J'42) (HNJCMY), Design Engr., Vickers Inc., 1400 Oakman Blvd., Detroit; *for mail,* Apt. 110, 20 Tyler St., Detroit 3, Mich.
Micallef, Joseph M. ('24; '35) (FHS), Power Supt., E. I. du Pont de Nemours & Co., Box 152, Penns Grove, N.J.
Michael, Loren P. ('21), Ch. Mech. Engr., Chicago & Northwest. Ry., Chicago; *for mail,* 198 Myrtle Ave., Elmhurst, Ill.
Michal, Frank Walter (J'42) (MLCN), Mech. Project Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 39 Oxford St., Newark 5, N.J.
Michalec, George William (J'45) (BNMAYZ), Mech. Engr., Bell Tel. Labs., 180 Varick St., New York, N.Y.; *for mail,* 385 Pleasant Ave., Cliffside Park, N.J.
Miche, John Anthony (J'43) (K), Ensign, U.S.N.R., PCE-846, A.P.O., New York; *for mail,* 651—36th Ave., San Francisco 21, Calif.
Michel, Frederick J. (J'45) (BLMTYZ), 322 E. Mason Ave., Alexandria, Va.
Michel, John N. (J'43) (ANMZ), Product Designer, Kollsman Instrument Div., Square D Co., 80-08—45th Ave.; *for mail,* 93-11—49th Ave., Elmhurst, N.Y.
Michel, John Raymond ('30; '35) (SCF), Commonwealth Edison Co., 72 W. Adams St., Chicago 90, Ill.
Michel, Leopold Robert (J'36) (ABHO), Polaroid Corp., Cambridge 39, Mass.
Michel, Rudolph ('17; '25; '35) (SNBKCE), Prin. Engr., Bur. of Ships, Navy Dept., Constitution Ave. & 17th St., N.W., Washington 25, D.C.; *for mail,* 6518—40th Ave., University Park, Md.
Michelena, Justo L. ('27; '35; '35) (BFS), Asst. Elec. Engr., Hershey Corp., Central Hershey, Havana, Cuba.
Michelsen, Henry ('37) (HNZ), Ch. Draftsman, New Departure Div., Gen. Motors Corp., N. Main St.; *for mail,* 64 Putman St., Bristol, Conn.
Michetti, Albert Louis (J'42) (BAKLMO), Sr. Test Engr., Charge Lab. Designing, Glenn L. Martin Co., Baltimore 3; *for mail,* 5344 Wright Ave., Baltimore 5, Md.
Micklethwaite, William E. (J'35) (HBCDJL), Engr., Baldwin Loco. Works of Can. Ltd., 304 Bay St.; *for mail,* 12 Kings Lynn Rd., Toronto 9, Ont., Can.
Middlehurst, Donald James (J'40) (CMDOBY), Ch. Engr., Victor Equip. Co., 844 Folsom St., San Francisco 7; *for mail,* 882—31st Ave., San Francisco 21, Calif.
Middleman, David (J'41) (JNBMYC), Matls. Engr., Black & Decker Mfg. Co., Towson 4; *for mail,* 3725 Belle Ave., Baltimore 15, Md.
Middlemiss, George Henry ('18; '35) (SEFH CK), Mgr. of Prod., Commonwealth & So. Corp. of N.Y., 600 N. 18th St., Birmingham, Ala.
Middleton, C. W. ('32), V.P., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Middleton, Leslie H. ('36) (AJMZ), V.P., Engrg., Elec. Autolite Co., Champlain St., Toledo, Ohio.
Middleton, Percy H. ('94; '01), 22 Elm St., Wellesley Hills, Mass.
Middleton, Ralph Edgar ('43), Ch. Engr., Aircraft Accessories Corp., 166 W. Olive Ave., Burbank, Calif.
Midgette, Ernst Leland (J'35) (BHNSDO), Dir., Research & Design, Am. Engrg. Co., Aramingo & Cumberland Sts., Philadelphia; *for mail,* 3008 Coulter St., Philadelphia 29, Pa.
Midgley, Frederick William ('04; '14) (BDM), 67 Edgecliff Terrace, Yonkers 5, N.Y.
Midgley, John McMillan (J'44) (ACE), Functional Test Foreman, Boeing Airplane Co., Wichita; *for mail,* 307 S. Roosevelt, Wichita 8, Kan.
Midtlying, C. R. (J'37), Comdr., U.S.N.R., 670 Ferne Ave., Drexel Hill, Pa.
Miedendorp, Henry, Jr. (28) (TCMDSO), Cons. Engr., Box 11, Hohokus, N.J.
Mielke, Carl Emil ('44) (LJH), Dept. Head, Engrg., Koppers Co., Inc., Koppers Bldg., Pittsburgh 19; *for mail,* 235 Princeton Ave., Pittsburgh 2, Pa.
Mierke, Frederick William ('29; '32; '35) (SEB), Res. Engr., Travelers Ins. Co., 332 Main St., Worcester 8, Mass.
Miesel, Christian F. (J'36) (SNDBFC), Mech. Engr., Kennedy-Van Saun Mfg. & Engrg. Corp., 2 Park Ave., New York; *for mail,* 58-43—79th St., Elmhurst, L.I., N.Y.
Mihalopoulos, Dan J. (J'41) (BEH), Mech. Engr., Moline Tractor Works, 3rd St. & 3rd Ave., Moline; *for mail,* 623—18th Ave., East Moline, Ill.
Mikels, John William ('30) (CEFKS), Ch. Prod. Engr., Pa. Power Co., 19 E. Washington St.; *for mail,* 318 Sumner Ave., New Castle, Pa.
Mikeska, P. Lawrence (J'28) (BNCMZH), Project Engr., Dole Valve Co., 1901-41 Carroll Ave., Chicago 12; *for mail,* 501 N. Central Ave., Chicago 44, Ill.
Mikhalevsky, Nicholas (J'45), 604 W. 115th St., New York 25, N.Y.
Mikina, Stanley Joseph ('30; '36) (ABHSYZ), *Junior Award,* '35; Research Engr., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Mikita, Joseph J. ('44), Assoc. Dir. of Research, Tex. Co., 135 E. 42nd St., New York 17, N.Y.
Mikovec, John S. (J'42) (SNDCKL), Asst. M.M., J. E. Decker & Sons, Mason City, Iowa.
Mikulich, Alexander (J'42) (LCAYON), 4th Mar. Div., Fleet Mar. Force, PAC, U.S.M.C.; *for mail,* Box 274, R.F.D. 1, Seymour, Conn.
Milan, Daniel Adolph (J'45) (BKO), Supt., Naval Repair Works, Todd-Johnson Dry Docks, Algiers; *for mail,* 6401 Milne St., New Orleans 19, La.
Milas, Raymond John (J'44) (OSEJMC), Lt. (j.g.), U.S.N.R., Engrg. Officer, LST-508, F.P.O., New York, N.Y.; *for mail,* 24 N. George St., Meriden, Conn.
Milazzo, John (J'44) (JMY), 546 Walnut St., Elizabeth 4, N.J.
Milbrook, Alfred Theodore ('44) (SFEKHZ), Asst. Ch. Engr., Charge Utilities Design, Stand. Oil Co. of Ind., Whiting, Ind.; *for mail,* 8129 Avalon Ave., Chicago 19, Ill.
Miles, Aaron Jefferson ('43) (BANSKJ), Prof. Mech. Engrg., Mo. Sch. of Mines & Metal, Box 250; *for mail,* 103 W. 14th St., Rolla, Mo.
Miles, Forrest Arthur (J'42) (ACW), Lt. (j.g.), U.S.N.R., Design Unit, Bur. of Ord., Navy Dept., Calif. Inst. of Tech., Pasadena 4, Calif.
Miles, George N. (J'43), 150 Mayflower Pl., Milford, Conn.
Miles, John C. (J'39) (SEFCLM), Asst. Prof. Mech. Engrg., Univ. of Ill., Urbana, Ill.
Miles, Robert Samuel (J'38) (CMJ), Supvr., Indus. Engrg. Dept., Gary Works, Natl. Tube Co.; *for mail,* 4069 Delaware St., Gary, Ind.
Miley, Frederick Benedict (J'45) (BEFH), Boiler Estimator, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 1721 Montgomery Pl., New York 61, N.Y.
Miley, Harrie William (J'39) (CDLMO), Safety Engr., Rochester Products Div., Gen. Motors Corp., 1000 Lexington Ave., Rochester 3, N.Y.
Milford, Albert M. (J'30), V.P., Capital Fdy. Corp., 4th St. & 26th Ave., Long Island City; *for mail,* 145-18—21st Ave., Whitestone, N.Y.
Milici, Frank, Jr. (J'43) (ABCM), Jr. Engr., Calibration Unit, Wood-Ridge Plant, Wright Aero. Corp., Wood-Ridge; *for mail,* 21 N. 13th St., Haledon, Paterson 2, N.J.
Mill, Victor John, Jr. (J'43) (HCDJ), Gen. Mgr., Lawrence Mch. & Pump Corp., 371 Market St., Lawrence, Mass.
Millar, Robert L. (J'40) (ACJ), 5 Banks St., Waltham 54, Mass.
Miller, Adam James, II (J'41) (MCDEFA), Lt. (j.g.), U.S.N.R., 221 Bailey St., New Cumberland, Pa.
Miller, Albert Raymond ('29; '35) (FSEK), Power Engr., Gen. Foods Corp., 250 Park Ave., New York, N.Y.; *for mail,* 190 Churchill Rd., Teaneck, N.J.
Miller, Alten Sidney ('14; F'45) (EJM), Cons. Engr., 80 Westcott Rd., Princeton, N.J.
Miller, Archibald Tearse ('25; '37) (ACRKS), Sales Engr., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 125 Godwin Ave., Ridgewood, N.J.
Miller, Arnold Henry ('21; '24) (MOW), Cons. Engr., 140 Nassau St., New York 7, N.Y.
Miller, Arthur A. (J'37) (LMSZ), Devel. Engr., Titanium Div., Natl. Lead Co., Sayreville, N.J.; *for mail,* 2180 Richmond Rd., Staten Island 6, N.Y.
Miller, Arthur Morris ('38; '45) (CMDTG), Sales Engr., Willcox & Gibbs Sewing Mch. Co., 214 W. 39th St., New York 18, N.Y.; *for mail,* 924 Castle Point Terrace, Hoboken, N.J.
Miller, Aubrey Bernard (J'41) (BAS), Asst. Project Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford; *for mail,* 359 Fern St., West Hartford 7, Conn.
Miller, Burton Rich (J'39) (CEFK), Staff Engr., Plant Engrg. Dept., Caterpillar Tractor Co., Peoria; *for mail,* 1014 W. McClure, Peoria 5, Ill.
Miller, Carl E. ('36; '41) (FSECDK), Chief, Htg. Sec., Office of Chief of Engrs., 21st & Virginia Ave., Washington, D.C.; *for mail,* 33 Ft. Hunt Rd., Alexandria, Va.
Miller, Charles Brookshire (A'43) (COYZ), V.P., Automatic Sprinkler Corp.; *for mail,* 447 Catalina Ave., Youngstown, Ohio.
Miller, Charles Marshall (J'43) (CMJA), Ensign, Civ. Engr. Corps, U.S.N.R., 29th Naval Constr. Bn., Navy 3705, F.P.O., San Francisco, Calif.; *for mail,* 133 Peterborough St., Boston 15, Mass.
Miller, Charles Thomas (J'43) (OMCDAY), Naval Arch., Naval Shipyard, Boston; *for mail,* 393 W. Broadway, South Boston 27, Mass.
Miller, Clarence A. ('16; '26; '35) (RABGS), Pres., Lake Erie, Franklin & Clarion R.R. Co., Clarion; *for mail,* Rosemont Farm, Franklin, Pa.
Miller, David (J'45) (MYZCAD), Supvr., Einson-Freeman Co., Inc., 53-06 Grand Ave., Maspeth, L.I.; *for mail,* 105 E. 177th St., Bronx 53, N.Y.
Miller, David, Jr. (J'43) (SEFLZO), Asst. Mech. Engr., J. G. White Engrg. Corp., 80 Broad St., New York 4; *for mail,* 3636 Greystone Ave., New York 63, N.Y.
Miller, David Jacob (J'42) (EFA), Sr. Test Engr., Wright Aero. Corp., Paterson; *for mail,* 132 S. Broad St., Ridgewood, N.J.
Miller, Donald E. (J'45) (LSCEFK), Constr. Engr., Procter & Gamble Co., Ivorydale 17; *for mail,* 2423 Jefferson Ave., Cincinnati 12, Ohio.
Miller, Donald James (J'45), Kimberly-Clark Corp., Neenah; *for mail,* 546—3rd St., Menasha, Wis.
Miller, Durben Erie (J'44) (MYEFRB), U.S.N.R., Engrg. Officer, LST-702, F.P.O., San Francisco, Calif.; *for mail,* Box 101, Route 1, Paragould, Ark.
Miller, Edgar B. ('38) (MJHBS), Asst. Ch. Engr., Charge of Devel., Commercial Iron Works, Box 2230, Portland 14; *for mail,* 654 S.E. 52nd Ave., Portland 15, Ore.
Miller, Edgar Harriott ('44), Ch. Engr., Packing Engrg. Corp.; *for mail,* 4 Pittsfield St., Cranford, N.J.
Miller, Edward Earl (J'40) (BMRCSN), Mar. Elec. Engr., Md. Drydock Co., Box 6306, South Sta., Baltimore 30; *for mail,* 2411 Garrison Blvd., Baltimore 16, Md.
Miller, Edward Godfrey ('13; '18; '35) (CKLZ), Pres., Compañia Azucarera Atlántica del Golfo; V.P., Cuban Atlantic Sugar Co., Apartado 69, Havana, Cuba.
Miller, Edward Harry (J'46), Ensign, U.S.N.R.; 1147 Keyes Ave., Schenectady, N.Y.
Miller, Edward L. (J'37) (BJ), Woodbury Rd., Woodbury, L.I., N.Y.
Miller, Elton Willard ('19) (CABKJ), Chief, Admin. Dept., Natl. Advis. Com. for Aero., Langley Field, Va.
Miller, Emanuel Jacques (J'46), U.S. Navy; 682 Montrose Ave., S.E., Roanoke 13, Va.
Miller, Ernest F. ('40) (ABC), Mgr., Engrg. Operas., Westinghouse Elec. Corp., Lester, Branch P.O., Philadelphia 13; *for mail,* 79 Bryn Mawr Ave., Lansdowne, Pa.
Miller, Erwin E. (J'39) (HJM), Capt., Ord. Dept., U.S.A., Engrg. Div., Raritan Arsenal, Metuchen, N.J.
Miller, Frank Dana (J'39) (LHBNE), Mech. Engr., with Douglas M. McBean, 315 Alexander St., Rochester 4; *for mail,* 109 Edgewood Ave., Rochester 10, N.Y.
Miller, Frank William ('28; '34; '35) (NCSL JZ), Works Mgr., Yarnall-Waring Co., 102 E. Mermaid Lane; *for mail,* 7805 Cobden Rd., Philadelphia 18, Pa.
Miller, George Henry (J'40) (MNCLHA), Process Engr., Danly Mch. Specialties Inc., 2100 S. 52nd Ave., Cicero 50; *for mail,* 9532 Jackson Ave., Brookfield, Ill.
Miller, George P. (J'38), Draftsman, Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 9036 Hargis St., Los Angeles 34, Calif.
Miller, H. C. L., Jr. (J'23), Life Ins. Co. of Va. Bldg., Richmond, Va.
Miller, Harry (J'37) (NZSBH), Engr., Shirgun Corp., 136 W. 52nd St., New York; *for mail,* 1473 Dahill Rd., Brooklyn 4, N.Y.
Miller, Harry (J'41) (AES), Midshipman, U.S.N.R., Sec. 25, Midshipmen's Sch., Notre Dame, Ind.
Miller, Harry Conold ('44), Field Engr., Gibbs & Hill, S.A., San Juan de Letran 8-402, Mexico, D.F., Mex.
Miller, Harry Garfield ('44) (RJBCES), Mech. Engr., Chicago, Milwaukee, St. Paul & Pacific R.R., 321 Everett St., Milwaukee 3; *for mail,* 1933 S. 82nd St., West Allis 14, Wis.

Miller, Henry Oscar (J'43) (ABMJCK), Electronics Technician's Mate, 3rd Class, U.S.N.; *for mail,* 2235 S. 60th Court, Cicero 50, Ill.
Miller, Howard E. (J'35) (ACD), 901 N. Frederic St., Burbank, Calif.
Miller, J. Melvin (J'42) (A), Maj., Army Air Forces, U.S.A., Exec. Officer, Working Com., The Aero. Bd., 4311 Munitions Bldg., Washington 25, D.C.; *for mail,* Box 284, R.F.D. 3, Vienna, Va.
Miller, Jack (J'44) (NZMOBJ), Mech. Engr., Design, Austin Co., 19 Rector St., New York; *for mail,* 1640 Park Pl., Brooklyn 33, N.Y.
Miller, James Gilbert ('39) (CDF), Supt. of Engrg., Westvaco Chlorine Products Corp., South Charleston; *for mail,* 1003 Valley Rd., Charleston, W.Va.
Miller, Jo Zach, IV (J'41) (FJS), Lt., Staff, Comdr. Serv. Force, U.S. Pacific Fleet, Fleet Maint. Office, F.P.O., San Francisco, Calif.
Miller, John Andrew ('13) (DFR), 7 S. New St., Nazareth, Pa.
Miller, John Andrew (J'38) (ABFGN), Research Engr., Calif. Research Corp., Richmond; *for mail,* 87 Ross Circle, Oakland, Calif.
Miller, John Calvin (J'40) (NBMK), Mech. Design Engr., D. W. Onan & Sons, 43 Royalston Ave., Minneapolis 5; *for mail,* 4018 Thomas Ave. N., Minneapolis 12, Minn.
Miller, John Edwin (J'43) (AHKMN), Test Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* 38 Heights Rd., Wyckoff, N.J.
Miller, John G. (J'34) (HKS), Lt. (j.g.), U.S.N.R., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* 304 Duncan Ave., Alexandria, Va.
Miller, John William ('44) (NMCBYJ), Managing Dir., Thornhill Industries Ltd., 2 College St., Toronto; *for mail,* 223 Fairlawn Ave., Toronto 12, Ont., Can.
Miller, Joseph Earl (J'44) (NSMHBA), Mech. Engr., Test Apparatus Designer, Natl. Advis. Com. for Aero., Langley Field, Va.; *for mail,* Box 180, R.R. 1, Clinton, Ind.
Miller, Joseph Francis (J'45) (ABM), Radio Corp. of Am., Camden, N.J.; *for mail,* 98 Hartley Rd., East Lansdowne, Pa.
Miller, Joseph Paul ('44) (RBNEA), Mech. Engr., Electro-Motive Div., Gen. Motors Corp., La Grange; *for mail,* 4820 Highland Ave., Downers Grove, Ill.
Miller, Karl Frederick (J'37) (ENRS), Combustion Expert, N.Y. Cent. System, 466 Lexington Ave., New York 17, N.Y.
Miller, Karl Otto ('41) (EFHKLS), Mgr. Steam Apparatus, Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4, Pa.
Miller, Kay ('45) (MHEBJY), Devel. Engr., Cent. Engrg. Dept., Food Mchy. Corp., 333 W. Julian St., San Jose, Calif.
Miller, Keith William (J'42), R.F.D. 1, Coloma, Mich.
Miller, Laurence George ('45) (SCKF), Asst. Supvg. Engr., Inventory Control Div., Commonwealth Edison Co., 72 W. Adams St., Chicago 90; *for mail,* 810 Columbian Ave., Oak Park, Ill.
Miller, Lawrence Lantz (J'40) (CMDEJL), 5418 Lemon Grove Ave., Los Angeles 38, Calif.
Miller, Melville Stone (J'42) (CZA), Student, Pomona College, Claremont; *for mail,* 248 S. Normandie Ave., Los Angeles 4, Calif.
Miller, Morton Alex (J'42) (KMC), Mech. Engr., Refrigeration Engrg. Inc., 6107 S. Central Ave.; *for mail,* 811 E. 74th St., Los Angeles 1, Calif.
Miller, Newby Lewis ('28; '46) (SFKH), Test Engr., Kansas City Power & Light Co., P.O. Sta. F, Kansas City 1, Mo.
Miller, Nils ('36) (AB), Dist. Mgr., SKF Industries, Inc., 591 Peachtree St., Atlanta 3, Ga.
Miller, Norman E. ('40) (AHM), Pres., Dir. of Engrg., Norman E. Miller & Associates, 648 Buhl Bldg., Detroit 26, Mich.
Miller, Norval Richard (J'43) (NBJDCM), Draftsman, United Engrg. & Fdy. Co., 1st Natl. Bank Bldg., Pittsburgh 22; *for mail,* 525—4th St., Pitcairn, Pa.
Miller, Paul Harry ('45) (MNBLJR), Process Engr., Carboloy Co. Inc., Box 237, Roosevelt Park P.O., Detroit 32, Mich.
Miller, Paul V. ('30) (DBMNO), Mgr., Small Tool Div., Taft-Peirce Mfg. Co., Woonsocket, R.I.
Miller, Paul William (J'46), 154 Monroe St., N.W., Warren, Ohio.
Miller, Ralph Hansen ('19; '35) (EBNJR), Ch. Engr., Cycle Div., Nordberg Mfg. Co., Chase St., Milwaukee 7, Wis.
Miller, Richard Clayton (J'42) (ACMYJL), Tech. Engr., Gen. Elec. Co., Western Ave.; *for mail,* 138 Williams Ave., Lynn, Mass.
Miller, Richard O. ('43) (OMC), Engr., Trundle Engrg. Co., Bulkley Bldg., 1501 Euclid Ave., Cleveland 15, Ohio; *for mail,* 733 W. Lasalle St., South Bend, Ind.
Miller, Robert Anderson ('31) (LMOAYC), Tech. Sales Engr., Pittsburgh Plate Glass Co., 2000 Grant Bldg., Pittsburgh 19, Pa.
Miller, Robert C., Jr. (J'37) (AMC), Prod. Engr., Bell Aircraft Corp., Marietta; *for mail,* Terrace Apts., 140 Alden Ave., N.W., Atlanta, Ga.
Miller, Robert H. (J'35) (MFR), Lt. (j.g.), U.S.N.R., 1927 S. 95th St., West Allis 14, Wis.
Miller, Robert Irving ('46) (CDO), Mgr., Curtis Publ. Co., Independence Sq., Philadelphia 5, Pa.
Miller, Robert Oscar (J'42) (ANCMBJ), Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 12 Banks Circle, Swampscott, Mass.
Miller, Robert P. (J'40) (ABE), 1884 Bedford Rd., Columbus, Ohio.
Miller, Robert Stoecker (J'40) (CMDY), Capt., Coast Arty. Corps (AA), Automotive Dept., Antiaircraft Arty. Sch., Bldg. 1915, Camp Davis, N.C.; *for mail,* 8025—11th Ave., N.E., Seattle 5, Wash.
Miller, Robert Wright (J'36) (COM), Staff Consultant, with Frederick E. Anderson, 375 Riverside Dr., New York 25, N.Y.; *for mail,* 234 Speedwell Ave., Morristown, N.J.
Miller, Rolla Leonard ('37; '41) (CFS), Inspr., Hartford Steam Boiler Inspc. & Ins. Co., 807 Van Nuys Bldg., Los Angeles; *for mail,* 2039 Camino Real, Arcadia, Calif.
Miller, Roswell ('22; '26; '35), Migdale House, Millbrook, N.Y.
Miller, Sheldon Martin (J'39) (MOCWLK), Partner, Tech. Serv. Co., 910 S. Boston, Tulsa, Okla.
Miller, Sherman Nussey ('44) (BRS), Ch. Mech. Engr., Am. Loco. Co., Jay St., Schenectady 5; *for mail,* 2009 Lexington Pkwy., Schenectady 8, N.Y.
Miller, Spencer, Jr. ('42), 217 Turrell Ave., South Orange, N.J.
Miller, Spencer, Sr. ('88; F'36) (A), *Manager,* '14-'17, *Vice-President,* '17-'19; 661 Glenneyre St., Laguna Beach, Calif.
Miller, Stanley O. (J'39) (CMYAO), Secy., Treas., New Products Corp.; *for mail,* Box 273, Benton Harbor, Mich.
Miller, Stephen Chas. ('24) (DMK), Drafting Supvr., Matson Navigation Co., 215 Market St., San Francisco 5; *for mail,* 416—17th Ave., San Francisco 21, Calif.
Miller, Theodore H. ('06), 12 Forbus St., Poughkeepsie, N.Y.
Miller, Walter ('22) V.P., Continental Oil Co., Ponca City, Okla.
Miller, Walter A. (J'42) (BDYZLR), Research Engr., Bakelite Corp., 230 Grove St.; *for mail,* 153 Franklin St., Bloomfield, N.J.
Miller, Walter Hubert (A'43), Instr. in Mech. Engrg., Univ. of Mo.; *for mail,* 1313 Wilson Ave., Columbia, Mo.
Miller, Wesley Albert ('42) (LBK), Supvg. Engr., Charge Pressure, Chem. Plants Div., Blaw-Knox Co., 5001 Baum Blvd., Pittsburgh 6, Pa.
Miller, Wesley Barrington ('42), Capt. Army Air Forces, U.S.A., Wright Field; *for mail,* 205 Apple St., Dayton, Ohio.
Miller, William Dawes (J'42) (ACM), 1021 Greenville Ave., Glendale, Ohio.
Miller, William Glenn ('44) (CBMJ), Mgr., Hdq. Mfg. Dept., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh; *for mail,* 1148 Cedar Blvd., Pittsburgh 16, Pa.
Miller, William Henry (J'43) (YZSKBL), 1363 Tutwiler Ave., Memphis 7, Tenn.
Milligan, Robert Galt ('38) (FLS), Supvr., Ocean Accident & Guarantee Corp., 1 Park Ave., New York, N.Y.
Millikan, Robert Andrews (Non-Member), *A.S.M.E. Medallist,* '26; Prof. of Physics, Calif. Inst. of Tech., Pasadena 4, Calif.
Millinger, William A. F. ('18; '19; '30) (AOE NHW), 1864 North Ave. 51, Los Angeles 42, Calif.
Mills, Blake David, Jr. ('35; '44) (NJBYO), 938—22nd Ave. N., Seattle 2, Wash.
Mills, David L. ('44) 919 N. 5th Ave., Maywood, Ill.
Mills, Ernest A. ('25), Ch. Engr., Mgr., Electricity Dept., Metro. Borough of Hackney, 18-24, Lower Clapton Rd., Hackney, London, E. 5; *for mail,* 26 Bancroft Ave., E. Finchley, London, N. 2, England.
Mills, Eugene Booth (J'43) (COSJHM), Ensign, U.S.N., LST-1087, F.P.O., San Francisco, Calif.; *for mail,* 538 S. Fountain St., Wichita 9, Kan.
Mills, Frank C., Jr. (J'41), Lt., U.S.N.R., Merion Cricket Club, Haverford, Pa.
Mills, Frederic ('28; '35) (BMR), Ch. Mech. Engr., West. Australian Govt. Rys., Midland Junction, Western Australia.
Mills, Frederick (J'41) (ABJ), Graduate Student in Bacteriology, Univ. of N.C.; *for mail,* Box 1678, Durham, N.C.
Mills, Fulton Sherman (J'34) (G), Ch. Draftsman, Div. of Highway Planning, Miss. State Highway Dept., Box 1638, Jackson 113; *for mail,* 539 S. Jackson St., Brookhaven, Miss.
Mills, Halstead Henry (A'22), Ch. Safety Engr., City of Detroit, 555 Clinton St., Detroit 26, Mich.
Mills, Harold H. (J'13), Exec., Dryden & Palmer Inc., 43-44—21st St., Long Island City, L.I., N.Y.
Mills, Joseph Irvin (J'41) (NJYHMA), Devel. Engr., Plant B, Wheel & Brake Div., Goodyear Aircraft Corp.; *for mail,* Apt. D-766, Wren Walk, Akron 11, Ohio.
Mills, Victor Blackford (J'43) (JMC), Engr., Am. Sheet Metal Works Inc., 920 N.E. Glisan St., Portland 14; *for mail,* 7316 S.E. 31st Ave., Portland 2, Ore.
Mills, William Earl (J'42) (MKJHGF), Engr., Draftsman, W. Coast Shipbuilders Ltd., 205 W. 1st Ave.; *for mail,* 1820 W. 10th Ave., Vancouver, B.C., Can.
Millson, John Case (J'36) (JBANRS), Metallurgist, Aluminium Labs. Ltd., Box 84; *for mail,* 10 Regent St., Kingston, Ont., Can.
Millspaugh, Wm. H. ('03; '14), Pres., Centrifugal Steel, Inc., Box 547, Sandusky, Ohio; Chmn., Millspaugh, Ltd., Vulcan Rd., Sheffield 9, England.
Milne, Gordon Ross ('44) (SFKC), Supt., East River Sta., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.; *for mail,* 57 Wayland Dr., Verona, N.J.
Miltenberger, Geo. K. ('13; '25) (FHS), Gen. Supt., Elec. Opera., Union Elec. Co. of Mo., 315 N. 12th Blvd., St. Louis, Mo.
Mims, William Henry (J'44) (RMBNFC), Shop Engr., Cent. of Ga. Ry. Co., Pine St., Macon, Ga.
Minarik, Rudolf Glenn ('40) (NBWK), Div. Engr., Research & Devel., Kimberly-Clark Corp., Neenah, Wis.
Mindlin, Raymond D. ('38) (BNAJHO), Assoc. Prof. Civ. Engrg., Columbia Univ., New York 27; *for mail,* 45 Christopher St., New York 14, N.Y.
Mindnich, Charles Edward (J'42) (CMS), 79 Henderson Ave., Kenmore 17, N.Y.
Miner, Anson Wadsworth ('40) (CBL), Sales, Waterbury Brass Goods Branch, Am. Brass Co., Box 829, Waterbury; *for mail,* 21 Frederick St., Waterbury 38, Conn.
Miner, Harold L. ('17; '35) (ACDLMY), Mgr., Safety & Fire Protection Div., E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington 98, Del.; *for mail,* 4237 Osage Ave., Philadelphia 4, Pa.
Miner, Irving O. ('41) (ZBH), Ch. Engr., Charge Design, Builders-Providence, Inc., 9 Codding St., Providence 1; *for mail,* 36 Hill Court, Rumford 16, R.I.
Ming, Frederick W. ('18; '25; '35) (MNG), Assoc. Prof. Mech. Engrg., Poly. Inst. of Brooklyn, 99 Livingston St., Brooklyn 2, N.Y.
Minges, John Victor (J'42) (BLS), Asst. Project Engr., E. B. Badger & Sons Co., 75 Pitts St., Boston, Mass.
Mingledorff, Walter Lee, Jr. (J'37) (CMBF), V.P., Savannah Mch. & Fdy. Co., 632 Indian St.; *for mail,* 138 E. 50th St., Savannah, Ga.
Miniberger, George Vaclav ('29; '42) (NBMD), Designing Engr., Harnischfeger Corp., 4400 W. National Ave., Milwaukee 14; *for mail,* 902 S. 24th St., Milwaukee 4, Wis.
Minieka, Edward Thomas (J'40) (ACGKSZ), Asst. Engr., Commonwealth Edison Co., 72 W. Adams St., Chicago 90; *for mail,* 3303 S. Lowe Ave., Chicago 16, Ill.
Mink, Joseph Andrew J'44), Design Draftsman, De Laval Steam Turbine Co.; *for mail,* 219 William St., Trenton, N.J.
Minkema, Wm. H. ('18; '25; '35), Universal Oil Products Co., 310 S. Michigan Ave.; *for mail,* 10716 S. State St., Chicago, Ill.
Minkin, Herbert Laurence (J'45) (ABF), Mech. Engr., P-2, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 955 E. 130th St., Cleveland 8, Ohio.
Minns, Rupert Garfield ('41) (CNYBMS), Ch. Engr., Natl. Rubber Mchy. Co., 917 Sweitzer Ave., Akron 11, Ohio.
Minor, Burt Stanley ('30) (BCJMHD), V.P., Charge Engrg., Regan Forge & Engrg. Co., 1301 Regan St., San Pedro, Calif.
Minor, John C. ('38), Cons. Engr., 110 E. 42nd St., New York, N.Y.
Minshall, Jay Roland (J'44) (MCGNOB), Asst. Engr., Engrg. & Maint. Dept., Eastman Kodak Co., Kodak Park, Rochester 13; *for mail,* 14 Ave. D, Rochester 5, N.Y.
Mirsky, William (J'44), 18 York St., Hartford 6, Conn.
Misch, Charles E. ('22; '30; '35) (YDLC), Metro. Dist. Mgr., Engrg. & Sales, Read Mchy. Co. Inc., York, Pa.; *for mail,* 11½ E. 87th St., New York 28, N.Y.
Miser, John Pyle (J'44) (NMKDCA), Design Engr., Knox Stove Works, 2016 Ailior Ave., Knoxville 7; *for mail,* 3908 Bellemeade Ave., Knoxville 16, Tenn.
Mises, Richard von ('43) (AB), Prof., Graduate Sch. of Engrg., Harvard Univ., Cambridge; *for mail,* 31 Concord Ave., Cambridge 38, Mass.
Misner, David M. (J'46), 335 Newbury St., Boston, Mass.
Misselhorn, Howard John (J'41) (MCDBHJ), Engr., Gen. Elec. Co.; *for mail,* 71 Pomeroy Ave., Pittsfield, Mass.
Mistron, Carl Andrew (J'42) (MABCJ), Methods Engr., Control Instrument Co., 65-67—35th St., Brooklyn; *for mail,* 4801—43rd Ave., Long Island City 4, N.Y.
Mitcha, John Ludvig (J'42) (SE), Turbine Field Engr., Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Mitcham, Edward Hartwell ('23; '37) (ONM YAB), Lt. Col., Corps of Engrs., U.S.A.; Secy., Treas., Standard Associates, Inc.; *for mail,* Edgewood, Katonah, N.Y.
Mitchel, Albert H. ('45) (NMLOJ), Designing Engr., 311 Lincoln Bank Bldg., Syracuse 2, N.Y.
Mitchell, A. Hoadley (J'41) (ACM), Capt., Army Air Forces, Prod. Line Maint. Officer, Stockton Field, Calif.
Mitchell, Alexander Louis, Jr. ('38) (SFCK), West. Sales Mgr., Riley Stoker Corp., 122 S. Michigan Ave., Chicago 3; *for mail,* 422 S. Crescent Ave., Park Ridge, Ill.

Mitchell, Archie Raymond ('40) (FSJB), Supvg. Engr., Knott Corp., 439 Madison Ave., New York 22; *for mail,* Hotel Shelton, 49th St. & Lexington Ave., New York, N.Y.
Mitchell, Cecil G. (J'42) (EFNSRA), Ch. Mar. Engr., Shepard Shipping Co., 801—2nd Ave., New York; *for mail,* 17 Hilltop Acres, Yonkers 4, N.Y.
Mitchell, Charles Ambrose ('42) (CFSZB), Sales & Serv. Engr., Hagan Corp., 18 E. 4th St., Cincinnati 2; *for mail,* 1253 Springfield Pike, Wyoming 15, Ohio.
Mitchell, David Burr (J'43) (MCBS), Capt., Ord. Dept., U.S.A., Chief, Statistics & Record Sec., Spare Parts Branch, Hdq., Ord. Sec., Am. Forces, West Pac., A.P.O. 707, San Francisco, Calif.; *for mail,* 306 W. Dudley Ave., Westfield, N.J.
Mitchell, Frank Kenneth ('41) (EMR), Asst. Gen. Supt., N.Y. Cent. System, 466 Lexington Ave., New York; *for mail,* 141 Bellair Dr., Dobbs Ferry, N.Y.
Mitchell, Fred Tom, Jr. (J'43) (CMDL), 50 Peterboro, Detroit 1, Mich.
Mitchell, Harold Clifton (J'38) (BSKLYR), Mech. Dept., Tide Water Associated Oil Co., E. 22nd St., Bayonne, N.J.; *for mail,* 71 Orange St., Brooklyn 2, N.Y.
Mitchell, Harry ('43) (CMTBD), Gen. Supt., Whitin Mch. Works, Main St.; *for mail,* 379 Hill St., Whitinsville, Mass.
Mitchell, Henry M. (J'39), 2020 Campbell Ave., Schenectady, N.Y.
Mitchell, James Ronaldson, Jr. (J'45) (BOY), Welding Engr., Farrel-Birmingham Co., Inc.; *for mail,* 243 Wakelee Ave., Ansonia, Conn.
Mitchell, Jean F. (J'35), Sales & Devel. Engr., Chamberlain Corp., Waterloo, Iowa.
Mitchell, John Earl, Jr. (J'38) (MJCDES), Mgr., Planning & Estimating, Engrg. Dept., Lukenweld, Inc., Coatesville, Pa.
Mitchell, John Gordon ('44) (TYKLBN), Research Engr., Celanese Corp. of Am.; *for mail,* 801 Braddock Rd., Cumberland, Md.
Mitchell, Lester Felter ('45) (BCHJ), Mgr., Engrg., Addressograph-Multigraph Corp., 1200 Babbitt Rd., Cleveland 17; *for mail,* 2673 Wrenford Rd., Shaker Heights 22, Ohio.
Mitchell, Nathaniel McDonald ('37) (CMOTZ), Pres., Mgr., Barnes Textile Associates, Inc., 10 High St., Boston 10, Mass.
Mitchell, Norman Tuley ('27; '35) (SAECFK), Supvr., Power Engr., Cadillac Motor Car Div., Gen. Motors Corp., Clark St., Detroit 32; *for mail,* 9125 Birwood, Detroit 4, Mich.
Mitchell, Orville ('44) (BKM), V.P., Ch. Engr., John E. Mitchell Co., 3800 Commerce St.; *for mail,* 8441 San Fernando Way, Dallas, Tex.
Mitchell, Robert Barringer (J'40) (HJNCGZ), Westco Works, Fairbanks, Morse & Co., 4301 S. Spring Ave.; *for mail,* 2404 Bristow Ave., St. Louis 16, Mo.
Mitchell, Robert Franklin (J'41) (WMJNYB), Capt., Trans. Corps, Contract Termination Officer, Procurement Div., Office of Chief of Trans., New Orleans Port of Embarkation, New Orleans; *for mail,* 33 Papworth St., New Orleans 20, La.
Mitchell, Stephen George (J'45) (EKH), Asst. Engr., West. Elec. Co., Inc., New York, N.Y.; *for mail,* Box 453, University, Ala.
Mitchell, W. G. (J'41), Ch. Engr., Independent Pneumatic Tool Co., State & Claim Sts.; *for mail,* 3 S. View St., Aurora, Ill.
Mitchell, William F. (J'36) (CFLSRZ), Supt., Pa. Salt Mfg. Co., 4655 Biddle Ave., Wyandotte, Mich.
Mitchell, William Henry ('13; '35) (HJR), Mech. Engr., Niagara Falls Power Co., Elec. Bldg., Buffalo 3; *for mail,* 532—12th St., Niagara Falls, N.Y.
Mitchell, William Howard (J'41) (NKHJB), Mech. Design Engr., Westinghouse Elec. Corp., Lima, Ohio.
Mitchell, William John (J'41) (NOJHMG), Welding Engr., Stress Analyst, E. W. Bliss Co., 53rd St. & 1st Ave., Brooklyn; *for mail,* 1212 Ocean Ave., Brooklyn 30, N.Y.
Mitchell, William Montgomery ('42) (ABHMN), Stress Analyst, Consltd. Vultee Aircraft Corp., Vultee Field, Downey; *for mail,* 1147—6th St., Santa Monica, Calif.
Mitchell, William Sibley (J'44) (CJMDN), 1220 S. Center St., Terre Haute, Ind.
Mitelman, Leon V. ('42) (EFHKOS), Mech. Engr., Neiler, Rich & Bladen, 431 S. Dearborn St., Chicago 5; *for mail,* 1140 Pratt Blvd., Chicago 26, Ill.
Mitsch, Edward Henry ('21; '26; '35) (CFS), Results Engr., Elec. Generation, Cincinnati Gas & Elec. Co., Cincinnati 2; *for mail,* 1007 Omar Pl., Cincinnati 8, Ohio.
Mitsui, Koei (J'45) (HMY), 234—1st Ave., Toronto 8, Ont., Can.
Mittelberger, Frank (J'40) (BONCGA), Cons. Engr., 88-11 Elmhurst Ave., Elmhurst, L.I., N.Y.
Mittendorf, William ('16) (FS), Fuel Engr., Holmes-Darst Coal Corp., Trans. Bldg., Cincinnati 2; *for mail,* 6269 Grand Vista Ave., Cincinnati, Ohio.
Mittleman, Joel Jerry (J'46), West. Mchy. Co., 50 Church St., New York; *for mail,* 7814—15th Ave., Brooklyn 28, N.Y.
Mittman, Carl Friedrich ('45) (LO), Consultant, Engr., Buflovak Equip. Div., Blaw-Knox Co., 1543 Fillmore Ave., Buffalo 11; *for mail,* 323 Westgate Rd., Kenmore, Buffalo 17, N.Y.
Mitzelfeld, Marvin Walter (J'44) (BEHJNS), Mech. Engr., Mch. Designer, Natl. Twist Drill & Tool Co., Rochester, Mich.
Mixter, George Webber ('99; '08), V.P., Day & Zimmermann, Inc., 165 Broadway, New York 6, N.Y.
Mjolsnes, Elliot Leffler (J'35) (BMZ), Checker, Designer, Vinco Corp., Schaefer Highway & Joy Rd., Detroit; *for mail,* 19465 Glastonbury Rd., Detroit 19, Mich.
Mlynko, Walter Francis (J'44), Pat. Atty. Trainee, Bosworth & Sessions, 1337 Guardian Bldg., Cleveland; *for mail,* 14508 Strathmore Rd., East Cleveland 12, Ohio.
Moats, William L. (J'40), Ensign, U.S.N.R., U.S.S. *Downes,* DD-375, F.P.O., New York, N.Y.
Moberg, Eric Skoglund (J'39) (SEKBA), Test Mar. Engr., Naval Shipyard, Boston; *for mail,* 153 Waverly St., Arlington 74, Mass.
Mochel, Myron George ('30; '36) (CNJBSM), Supvr. Trng., Moore Steam Turbine Div., Worthington Pump & Mchy. Corp., Wellsville, N.Y.
Mock, Loyal Kay (J'37) (MOYH), Sr. Surveyor, Shell Oil Co., Inc., Box 744; *for mail,* 907 Stone, Great Bend, Kan.
Mockridge, Chester R. ('22; '39) (HLBJNS), Centrifugal Pump Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 24 Coeyman Ave., Nutley 10, N.J.
Modes, Edward E. (J'37) (ZBKH), Devel. Engr., Powers Regulator Co., 2720 Greenview Ave., Chicago 14; *for mail,* 2316 Granville Ave., Chicago 45, Ill.
Moe, Clarence M. ('42) (CHS), Sr. Civ. Engr., Charge of Recreational Devel. of Lake Areas, U.S. Engr. Office, 300 Broadway, Little Rock, Ark.
Moeller, Harry A. ('42) (RCMJB), Asst. Ch. Mech. Engr., Buckeye Steel Castings Co., 2211 Parsons Ave., Columbus 7, Ohio.
Moeller, Henry George ('44), Gen. Partner, Gen. Mgr., Sioux Alfalfa Meal Co., Vermillion, S.D.
Moeller, Robert (J'37) (NBJHYK), Mech. Engr., Engrg. & Design Sec., David Taylor Model Basin, Washington, D.C.; *for mail,* 9 Froude Circle, Bethesda 14, Md.
Moeller, William ('17) (CJM), Ch. Engr., Lone Star Cement Corp., 1217—1st Natl. Bank Bldg., Dallas 1, Tex.
Moen, Leclanché ('17), Engrg. and Investment Banking, Rm. 429, 522—5th Ave., New York 18, N.Y.
Moen, Levi W. ('40), 123 Lynwood Ave., Syracuse, N.Y.
Moen, Richard H. (J'44) (JCMNYO), Partner, Schmid Bros., 317 Filbert St., Lancaster; *for mail,* Millway Farm, Millersville, Pa.
Moen, Walter B. (J'40) (SKB), Instr. in Mech. Engrg., Pratt Inst., Brooklyn 5; *for mail,* 70 Lincoln Ave., Rockville Centre, N.Y.
Moerk, John Carroll ('43) (NBDM), Div. Engr., Fairbanks, Morse & Co.; *for mail,* 614 W. Stephenson St., Freeport, Ill.
Moesinger, Fred, Jr. (J'41) (CJK), c/o Cudworth, 753 Lyons Ave., Irvington, N.J.
Moffat, George Nichol ('24; '35) (JNYBW), Prof. Engrg. Matls., Ohio State Univ., Columbus 10, Ohio.
Moffatt, Charles Richardson (J'42) (COJKYZ), Asst. Divisional Engr., Calco Chem. Div., Am. Cyanamid Co., Bound Brook, N.J.
Moffett, Henry C. ('21) (CEJOSY), Ch. Engr., Moffett & Troop, Inc., 4 Smithfield St., Pittsburgh 22; *for mail,* Box 152, R.D. 1, Library, Pa.
Moffett, John Theodore ('45) (BEKORS), Turbine Application Engr., Gen. Elec. Co., 1405 Locust St., Philadelphia 2, Pa.
Moffitt, Robert Colborn ('25; '32) (CNE), Pres., Thunderbird Mines Ltd., N.P.L., Box 743, Seattle 11, Wash.
Moglia, Guido Albert (J'43) (JBHYAC), Pvt., Army Air Forces, Sqd. A-4, A.P.O. 19909-A, c/o Postmaster, New York; *for mail,* 84-38 Daniels St., Jamaica 2, N.Y.
Mohler, Lee Jason (J'38) (AESRC), Elec. Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Mohler, Robert Claude ('31; '35), M.M., O'Okiep Copper Co., Ltd., O'Okiep; *for mail,* Nababeep, Namaqualand, South Africa.
Mohler, Valentine Joseph ('41) (BCLMOZ), Ch. Loan Engr., Reconstruction Finance Corp., 208 S. La Salle St., Chicago; *for mail,* 563 Ashland Ave., River Forest, Ill.
Mohn, Henry LeRoy ('43) (CKBFNO), Ch. Engr., Fitzgibbons Boiler Co., Inc., Oswego, N.Y.
Mohn, Paul Eugene ('28; '34) (SEFKAH), Prof. Mech. Engrg., Head, Engrg. Dept., Univ. of Buffalo, Buffalo 14, N.Y.
Moir, Harry L. ('42) (FEAM), Tech. Adviser, Marketing Dept., Pure Oil Co., 35 E. Wacker Dr., Chicago 1, Ill.
Molé, Harvey E. ('01; F'41), Box 205, Summit, N.J.
Moler, Frank White, Jr. ('34; '45) (KMS), Mech. Engr., Supvr., Design, Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.
Molinari, Wilfred Henry ('35; '44) (BCJLMN), Engr., Chambers Works, E. I. du Pont de Nemours & Co.; *for mail,* Golfview Pk., Penns Grove, N.J.
Moline, Adolph Alexander ('31; '35) (CBHJN), Ch. Engr., Canadian Westinghouse Co., Ltd., 286 Sanford Ave. N., Hamilton, Ont., Can.
Molitor, Arvid Axel (J'44) (NMBC), Prod. Engr., Process Dept., Zenith Radio Corp., 6001 W. Dickens Ave., Chicago; *for mail,* 228 Hamilton Ave., Elgin, Ill.
Moll, Harvey Donald (J'45) (S), 108 E. 3rd, Lansdale, Pa.
Moller, Carl G. (J'46) (CEF), 455 W. 23rd St., New York, N.Y.
Moller, H. Frederick ('23; '25; '35) (CM), Mgr., Valve Div., Reed Roller Bit Co., Box 2119; *for mail,* 2812 Wheeler, Houston 1, Tex.
Moller, Jos. A. ('24; '30; '34) (AEF), Pure Oil Co., 35 E. Wacker Dr., Chicago 1, Ill.
Molleson, Gilbert Cox ('27) (SBHND), Chief, Bldg. & Mech. Sec., Engrg. Div., Corps of Engrs., U.S.A., Office of Div. Engr., S. Atlantic Div., 50 Whitehall St., Atlanta 3; *for mail,* 1426 Boulevard, N.E., Atlanta, Ga.
Molokie, Stephen W. ('17; '25; '35) (HLS), 1622 Edison Dr., San Antonio, Tex.
Molony, Noblett Joseph ('27) (CDLA), Mech. Engr., Natl. Biscuit Co., 449 W. 14th St., New York 11; *for mail,* 4 E. 95th St., New York 28, N.Y.
Molter, Frank H. (A'22), 27 Harding Ave., West Hempstead, L.I., N.Y.
Molvie, William A. (J'41) (OSKFZ), Dravo Corp., Dravo Bldg., 300 Pennsylvania Ave., Pittsburgh; *for mail,* 1135 Hillsdale Ave., Pittsburgh 16, Pa.
Momot, William Eovich (J'43) (CJMN), Owner, Gen. Mgr., Momot Metal Products Co., 42-45—27th St., Long Island City 1; *for mail,* 1738—53rd St., Brooklyn 1, N.Y.
Monaco, Anthony Philip (J'38) (ABCNMK), Ch. Engr., Bunyard Aircraft Co., 37-26—32nd St., Long Island City 1; *for mail,* 4127—76th St., Jackson Heights, N.Y.
Monaco, John Charles ('45) (CJKL), Engr., Patterson-Kelley Co. Inc., East Stroudsburg; *for mail,* R.D. 1, Stroudsburg, Pa.
Moncrief, Ernest (J'37) (LEFYZK), Asst. Dir. of Process, Fluor Corp. Ltd., Box 2889, Grand Ave. Sta., Kansas City, Mo.
Mongello, Thomas (J'43) (ANM), 101-21—102nd St., Ozone Park, N.Y.
Monich, Michael Theodore (J'41) (BJG), Project Engr., New Departure Div., Gen. Motors Corp., Bristol; *for mail,* 97 Circle St., Forestville, Conn.
Monks, George S., Jr. (J'39), 205—41st St., Manhattan Beach, Calif.
Monnier, Rene A. ('45) (BC), Mech. Engr., R. Hoe & Co., Inc., 910 E. 138th St., New York, N.Y.; *for mail,* 1 W. Harriet Ave., Palisades Park, N.J.
Monro, Donald A. ('45) (CPS), Acting Ch. Engr., Stand. Oil Co. of Ind., 910 S. Michigan Ave., Chicago, Ill.
Monroe, Paul Hill, Jr. (J'42) (HCMNSZ), Jr. Designer, Mech. Engr., Peerless Pump Div., Food Mchy. Corp., 301 W. Ave. 26, Los Angeles; *for mail,* 8924 Bleriot Ave., Los Angeles 45, Calif.
Monroe, William Stanton ('96; '01; F'41), Retired; 105 S. La Salle St., Chicago 3, Ill.
Monseth, Ingwald T. ('44) (CSKOGM), Southwest. Engrg. & Serv. Div. Mgr., Westinghouse Elec. Corp., 411 N. 7th St., St. Louis 1, Mo.
Monson, Harry O. (J'41) (AEM), 627 Bonnie Brae, River Forest, Ill.
Montagno, Antonio Marino (J'42) (BEJ), Maint. Supt., Welbilt Stove Co., 5718 Flushing Ave.; *for mail,* 59-27—59th Dr., Maspeth, N.Y.
Montague, Edward ('42) (CDJ), Supt., Schweitzer & Conrad Inc., 4435 Ravenswood Ave., Chicago 40; *for mail,* 4110 N. Ashland Ave., Chicago 13, Ill.
Montague, Edwin Newell ('29; '35) (CDL), Capt., Army Air Forces; *for mail,* 3406 Abingdon St., Arlington, Va.
Montague, Joseph F. ('21; '35) (BEFHKM), Gen. Mgr., Camden Serv. Stas. Inc., R.F.D. 1, Sewell, Gloucester Co.; *for mail,* 176 Carlisle Rd., Audubon, N.J.
Montague, Larry Dupre (J'37) (CDJM), Mgr., B. L. Montague Co. & Carolina Industries, E. Liberty St.; *for mail,* 509 W. Hamton, Sumter, S.C.
Monteith, Alexander Crawford ('43) (SRA), Westinghouse Elec. Corp., East Pittsburgh, Pa.
Montgomery, Bryant Smith (J'38) (KWABSO), Mech. Engr., Tenn. Valley Authority, Knoxville, Tenn.
Montgomery, Charles Dennison (J'40) (AES), 262 Seneca Ave., Havre de Grace, Md.
Montgomery, O. Duncan (A'45), Mgr. Foreign Student Trng., Foreign Mfg. Dept., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Montgomery, Wallace ('23; '24) (OMLKF), The Econometric Inst., Inc., 500—5th Ave., New York, N.Y.
Montgomery, Walter Lee (J'41) (JSW), Mech. Engr., Tenn. Valley Authority, 508 Arnstein Bldg., Knoxville, Tenn.
Montgomery, William J. ('17; '35), Head, Mch. Dept., Brockton High Sch., Brockton; *for mail,* 79 Western St., Brockton 46, Mass.
Montgomery, Willian Adams (J'43) (NBESJK), Mech. Engrg. Draftsman, Am. Bosch Corp., Main St., Springfield 7; *for mail,* 441 Sumner Ave., Springfield 8, Mass.
Montgomrey, James E. ('39), Institution of Mechanical Engineers, Storey's Gate, St. James Park, London, S.W. 1, England.
Montillon, George Duncan (J'41) (CDM), Indus. Engr., Procter & Gamble Co., Ivorydale 17, Ohio.

Montin, J. Ragnar Theodore (J'45) (JCDMYL), R. Montin Constr. Co., 1350 Main St., Walnut Creek, Calif.
Montresor, Joseph Jerome (J'44) (CES), Tech. Aide, N.Y. Ord. Dist., War Dept., 80 Broadway, New York, N.Y.; *for mail,* 17 Crest Rd., Hillsdale, N.J.
Moodie, Andrew ('30; '35) (EMS), Plant Engr., Falls Co., Yantic St.; *for mail,* 109 Sachem St., Norwich, Conn.
Moodie, David ('45) (SEFKJ), Mech. Engr., New Orleans Pub. Serv. Inc., Market St., S.E. Sta., New Orleans 13, La.
Moodie, Wm. C. ('21; '23; '35) (CNMZ), Pres., Calculagraph Co., 306 Sussex St., Harrison, N.J.
Moody, Arthur Maurice Greene ('35; '45) (SB HNE), Ch. Blower Engr., Elliott Co., Jeannette, Pa.
Moody, Charles F. ('24; '33) (NCBM), Plant Engr., Habirshaw Cable & Wire Div., Phelps Dodge Copper Products Corp., Foot of Point St., Yonkers 2, N.Y.
Moody, Howard N. ('20; '35), Cons. Engr., 823 Perdido St., New Orleans, La.
Moody, Lewis Ferry ('10; F'44) (HB), Prof. Hyd. Engrg., Princeton Univ.; *for mail,* 146 Hodge Rd., Princeton, N.J.
Moody, Lucian Barclay ('46), Col., Consultant, Office of Spec. Asst. to the President of the United States, 228 State Bldg., Washington; *for mail,* 2019 Allen Pl., N.W., Washington 9, D.C.
Moody, Sharon Clay (J'45), 100 N. Arden Blvd., Los Angeles 4, Calif.
Moody, William Montague ('19; '21; '25) (CJ HS), Matls. Supvr., Pelton Water Wheel Co., 2929—19th St., San Francisco 10; *for mail,* 725 Spruce St., Berkeley 7, Calif.
Moolhuyzen, Thomas ('23) (TLK), Propr., Thomas Moolhuyzen Co., Box 2065, Paterson, N.J.
Moon, Harry Albert (J'46) (EJS), Lt. (j.g.), U.S.N.R., U.S.S. *Charger,* CVE-30, F.P.O., New York, N.Y.; *for mail,* 423 Eleanor St., Pitcairn, Pa.
Moon, Leo C. (J'40) (AJC), Lt. Col., Army Air Forces, TSEPS-4G, Area B, Wright Field, Dayton, Ohio.
Moon, Ralph Marks ('44) (HELSKY), Sales Engr., Ingersoll-Rand Co., 1012 Baltimore Ave., Kansas City 6, Mo.
Moon, Robert Frank (J'42) Serv. & Constr. Engr., Air Conditioning, B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., Hyde Park, Boston 36; *for mail,* 252 S. Central Ave., Wollaston 70, Mass.
Mooney, David Alfred (J'45) (BHS), Asst. Prof., Mech. Engrg., Mass. Inst. of Tech., Cambridge 39; *for mail,* 35 Cherry Lane, Dedham, Mass.
Mooney, Dwight D. (J'40) (EKS), Sgt., 38th Div., U.S.A., Camp Shelby, Miss.
Mooney, James D. ('19), Pres., Willys Overland Inc., Toledo, Ohio.
Mooney, John Prendergast (J'43) (JNCSEB), Metal Inspr., Humble Oil & Refining Co.; *for mail,* Box 742, Baytown, Tex.
Mooney, Robert James (J'44), Lt., U.S.N.R., Navigator, Electronics Officer, U.S.S. *San Juan,* CL-54, F.P.O., Seattle, Wash.; *for mail,* 376 Edgewood Ave., New Haven, Conn.
Mooney, Weldon ('29; '35) (SZLCD), Mgr., Mooney Engrg. Co., 207 W. 8th St., Kansas City 6, Mo.
Moore, Charles Gale (J'45) (NWOCSE), Ensign, Tactical Radar Trng., U.S.N., K & P Hotel, St. Simon Island, Ga.; *for mail,* Box 175, Wauna, Ore.
Moore, Charles Newell (J'45), Apprentice Seaman, U.S.N.; 5 Oak St., Danvers, Mass.
Moore, Clarence Pascal, Jr. (J'45) (NOHJCK), Ensign, U.S.N.R., Asst. Damage Control Officer, U.S.S. *G. K. MacKenzie,* DD-836, F.P.O., New York, N.Y.; *for mail,* Box 538, Bristol, Va.
Moore, Chauncey Clark (A'46) (MACY), Mgr., Asphalt Sales Dept., Barber Asphalt Corp., Rm. 3329, 50 W. 50th St., New York 20, N.Y.
Moore, Coleman Bassett ('44) (ZCNLMH), Sr. Partner, Dir. of Engrg., Moore Products Co., H & Lycoming Sts., Philadelphia 24, Pa.
Moore, Edward B. (J'41) (FKSM), Devel. Engr., David Bradley Mfg. Works, Bradley; *for mail,* 545 S. Small, West Kankakee, Ill.
Moore, Edward Fredrick ('41) (MCH), Cons. Engr., 80 Broadway, New York 5, N.Y.
Moore, Ernest Allen ('44) (AELNBS), Mech. Designer, River Works, Gen. Elec. Co., 920 Western Ave.; *for mail,* 18 Spruce Rd., Lynn, Mass.
Moore, Frank E. (A'19) (CD), Pres., Mathews Conveyer Co., Ellwood City, Pa.
Moore, Frank H. ('26) (BHDN), Mech. Engr., Hardesty & Hanover, 101 Park Ave., New York 17; *for mail,* 417 Riverside Dr., New York 25, N.Y.
Moore, Frederick Clouston ('05) (C), Retired; 68 Muskoka Rd., Grosse Pointe Farms 30, Mich.
Moore, Greer O. (J'41) (AYJ), Test Engr., Boeing Aircraft Co., Seattle 14; *for mail,* 606 E. Thomas St., Seattle 2, Wash.
Moore, Harold Thompson (A'07) (MLOCDA), Cons. Engr., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2, Pa.
Moore, Harry Frederick ('43) (ZLH), Sr. Engr., Stand. Oil Devel. Co., Box 37, Elizabeth B, N.J.
Moore, Henry H. ('27), Opera. Engr., East River Generating Sta., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 150—95th St., Bay Ridge, Brooklyn, N.Y.
Moore, Herbert Fisher ('13) (BJN), Research Prof. Engrg. Matls., Emeritus, Univ. of Ill.; *for mail,* 803 Indiana Ave., Urbana, Ill.
Moore, James Donald (J'45) (SKH), Jr. Petroleum Engr., Humble Oil & Refining Co., Box 2180; *for mail,* Box 719, Y.M.C.A., Houston 1, Tex.
Moore, James P. (J'40) (CKM), Capt., Ord. Dept., 2nd Bn., 304th Ord. Regt., Camp Forrest, Tenn.; *for mail,* Box 88, Route 3, Tahlequah, Okla.
Moore, James Weston ('17; '22) (DJLS), Mem. President's Staff, Research & Sales, Am. Cast Iron Pipe Co., Box 2603, Birmingham 2, Ala.
Moore, Jesse Atlee (J'43), R.F.D. 1, Mentor Headlands, Painesville, Ohio.
Moore, John E. (J'36) (EJ), Engrg. Dept., Stand. Oil Co., Sugar Creek, Mo.
Moore, John Randle ('22) (EFS), Cons. Engr., 520 N. Michigan Ave., Chicago 11; *for mail,* 2006 Fargo Ave., Chicago 45, Ill.
Moore, Leonard S. (J'41) (BCD), Capt., Automotive Div., Ord. Research & Devel. Center, Aberdeen Proving Ground, Md.
Moore, Mark Brown ('44) (NJKRH), Acting Chmn., Mech. Engrg. Dept., Swarthmore College, Swarthmore, Pa.
Moore, Marvin L. (J'39) (DMP), 6506 Myrtle St., Houston, Tex.
Moore, Morgan MacVean ('42) (CMZ), Col., Consultant, Bur. of Manpower Utilization, War Manpower Comm., 1778 Pennsylvania Ave., N.W., Washington; *for mail,* 601—19th St., N.W., Washington 6, D.C.
Moore, Mortimer J. P. ('20; '35) (CEFKSZ), Asst. Ch., Htg. & Fuel Sec., 8th Serv. Comd., War Dept., Dallas 2; *for mail,* Box 2279, Dallas 7, Tex.
Moore, Raymond Pierson ('27) (SKFJHR), Mech. Engr., Buffalo Niagara Elec. Corp., 535 Washington St., Buffalo 3; *for mail,* 242 Knowlton Ave., Kenmore 17, N.Y.
Moore, Richard Calvin (J'42) (ACM), Ch. Engr., Moore-Eastwood & Co., Dayton; *for mail,* 304 Forrer Blvd., Dayton 9, Ohio.
Moore, Robert Layton (J'46), 625½ W. Santa Barbara Ave., Los Angeles 37, Calif.
Moore, Stanley Milward (J'42) (JMO), Lt. (j.g.), U.S.N.R., Salvage Diving Officer, U.S. Naval Ship Salvage, Pier 88, North River, New York 19, N.Y.; *for mail,* 7065 Flaccus Rd., Ben Avon, Pittsburgh 2, Pa.
Moore, Thomas Goode ('37; '44) (LKMNJ), Mech. Engr., Charge Superpressure Engrg., Am. Instrument Co., 8030 Georgia Ave., Silver Spring; *for mail,* 237 Willow Ave., Takoma Park 12, Md.
Moore, W. D. ('21), Pres., Am. Cast Iron Pipe Co., Box 2603, Birmingham, Ala.
Moore, W. Joe (J'33) (JCDM), Supt. Foundries, Am. Cast Iron Pipe Co., Box 2603, Birmingham; *for mail,* 207 Mecca Ave., Birmingham 9, Ala.
Moore, Walter Albert ('30; '35) (GM), Meadows Critoph, Cons. Engrs.; *for mail,* 17 Spencer Ave., Toronto, Ont., Can.
Moore, Walter Graham (J'41) (NMDHBC), Ch. Draftsman, Mech. Design, Pan Am. Engrg. Co., 820 Parker St., Berkeley 2, Calif.
Moore, Wesley Robert ('36) (CZKFSY), Regional Mgr., Minneapolis-Honeywell Regulator Co., 5005 Euclid Ave., Cleveland 3, Ohio.
Moore, William ('45), Supt. M.P., Erie R.R. Co., Cleveland; *for mail,* c/o C. F. McKinney, 1342 W. 11th St., Cleveland 2, Ohio.
Moore, William Enoch ('03) (BEHNSZ), Pres., W. E. Moore & Co. Engrs., Box 1257, Pittsburgh 30, Pa.
Moore, William James ('18), 802 Main St., Toms River, N.J.
Moore, William S., Jr. (J'40) (ASK), Engrg. Asst., Process Engrg. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 15 July Walk, Long Beach, L.I., N.Y.
Moorehouse, Glen William ('46) (BCD), Engr., Tenn. Eastman Corp.; *for mail,* 1138 Cherry St., Kingsport, Tenn.
Moorehouse, William Sidney ('38) (SKCF), Supt. of Services, Tenn. Eastman Corp., Kingsport, Tenn.; *for mail,* Route 2, Gate City, Va.
Moores, Robert de la Hey ('25; '32; '35) (ES MW), Lt., U.S.N.R., U.S.S. *Capricornus,* AKA-57, F.P.O., San Francisco; *for mail,* 3360 Octavia St., San Francisco 23, Calif.
Moorhead, Dwight G. ('24; '27) (HLSKGC), Hyd. Sales Engr., Newport News Shipbldg. & Dry Dock Co., Newport News, Va.
Moorhouse, Milton (J'34) (ZMC), Asst. Mech. Supvr., Imperial Oil Ltd., E. Calgary St., Calgary, Alta., Can.
Moorman, William Taylor (J'41) (LCMD), Asst. Plant Engr., Monsanto Chem. Co., Anniston, Ala.
Morain, Willard A. (J'37) (EBSZ), Project Engr., Gen. Mchy. Corp., N. 3rd St.; *for mail,* 810 Webster Ave., Hamilton, Ohio.
Moran, George Upham ('44) (RSJZFE), Research Investigator, Assn. of Am. Railroads, 59 E. Van Buren St., Chicago 5, Ill.
Moran, Joseph J. (J'39) (CL), Pur. Engr., E. B. Badger & Sons Co., 75 Pitts St., Boston; *for mail,* 103 Wildwood St., Winchester, Mass.
Moran, Willard Royse (J'36) (SFRACD), Statistician, Toledo Edison Co., Edison Bldg., Toledo 4; *for mail,* 3312 Beverly Dr., Toledo 9, Ohio.
Morehead, F. Hugh ('19; '35) (HS), V.P., Ch. Engr., Walworth Co., Inc., 60 E. 42nd St., New York 17, N.Y.
Morehiser, John Edward, Jr. (J'45) (ESRKFM), Asst. Prof. Mech. Engrg., Tulane Univ., 6823 St. Charles, New Orleans 15, La.
Morehouse, J. Stanley ('21; '25; '35) (BEFH JS), Dean of Engrg., Prof. Mech. Engrg., Villanova College, Villanova, Pa.
Moreland, Edward Leyburn ('21) (OSELRG), Sr. Partner, Jackson & Moreland, 31 St. James Ave., Boston 16, Mass.
Moreland, William Benjamin (J'44), Assoc. Mar. Engr., Engrg. Plan Approval Sec., U.S. Maritime Comm., 348 Baronne St., New Orleans 12, La.
Moreland, William J. ('38) (BHNSZ), Prof., Rensselaer Poly. Inst., Troy, N.Y.
Morey, Albert A. (A'31), Asst. V.P., Marsh & McLennan Inc., 164 W. Jackson Blvd., Chicago, Ill.
Morey, Arthur Herbert ('40) (RSBEF), Project Engr., Gen. Elec. Co., East Lake Rd.; *for mail,* 3929 Eliot Rd., Erie, Pa.
Morey, Ronald H. (J'37), Lt. (j.g.), U.S.N.R.; *for mail,* 1325 Bryant St., Palo Alto, Calif.
Morgan, Albert H. ('18; '26; '35) (CFO), Dir., Div. of Operas., Dept. of Pub. Works, City of New York, Mun. Bldg., New York 7; *for mail,* 2475 Palisade Ave., New York 63, N.Y.
Morgan, Alva Britton ('30; '34) (ELCMS), Consultant, Edison Elec. Inst., 420 Lexington Ave., New York 17, N.Y.; *for mail,* 12 Sunset Rd., Darien, Conn.
Morgan, Antony John Andrew (J'44) (ABCKH), 881 N. Michigan St., Pasadena 6, Calif.
Morgan, Burton Davis (J'39) (YCHT), Ch. Engr., Adhesive Plaster Mill, Johnson & Johnson, 500 George St., New Brunswick, N.J.
Morgan, David William Rowsen ('36) (SRME), *Vice-President,* '43-'45; Mgr., Steam Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* 913 Strath Haven Ave., Swarthmore, Pa.
Morgan, Donald King (J'32) (CJK), Engr., Charge of Devel. Project, Hartford-Empire Co., 333 Homestead Ave., Hartford; *for mail,* 10 Highland Ave., Buena Vista, West Hartford, Conn.
Morgan, Everett Kellogg ('18; '23) (CNR), Gen. Sales Mgr., Giddings & Lewis Mch. Tool Co., 142 Doty St., Fond du Lac, Wis.
Morgan, George Eben ('42) (MCKFS), Mech. Engr., Stand. Brands, Inc., Charles Point, Peekskill, N.Y.
Morgan, George Rose ('42) (SFKEOT), Partner, Steam Dept. Head, J. E. Sirrine & Co., S. Main St., Greenville, S. C.
Morgan, Glen Weaver (J'43) (YJMNOS), Ch. Engr., Mech. Branch, Ogden Arsenal, Ogden; *for mail,* 224 F St., Salt Lake City, Utah.
Morgan, H. H. ('22) (JORS), V.P., Ch. Engr., Robert W. Hunt Co., Engrs., 175 W. Jackson Blvd., Chicago 4, Ill.
Morgan, Harry Taylor ('43) (ESRALB), Supvg. Engr., Mar. Serv. Dept., South Philadelphia Works, Westinghouse Elec. Corp., Lester; *for mail,* 4527 Walnut St., Philadelphia 39, Pa.
Morgan, Ivor N. R. (J'33) (BNSFEH), Mech. Engr., Stand. Brands, Inc., 595 Madison Ave., New York, N.Y.
Morgan, James E. (J'39) (MONWYS), 9845 Lake Ave., Cleveland 2, Ohio.
Morgan, James L. ('41) (CDJLMO), Indus. Engr., K. H. Davis Wire & Cable Corp., 2417 E. 23rd St., Los Angeles 11, Calif.
Morgan, John Davis ('17; '35), Cons. Engr., 60 Wall St., New York, N.Y.; *for mail,* "Highhickory," Wyoming Ave., South Orange, N.J.
Morgan, John Irby, Jr. (J'38), 2nd Lt., Air Transport Comd., A.P.O. 489, Sta. 11, Area 3, c/o Postmaster, New York, N.Y.
Morgan, John Thoburn ('12; '19; '25; '41) (CM), Pres., Charleston Elec. Supply Co., 914 Kanawha Blvd., Charleston 29; *for mail,* 1538 Quarrier St., Charleston 1, W.Va.
Morgan, Paul B. ('91; '00) (JBAFSN), Chmn. of Bd., Morgan Constr. Co., 15 Belmont St., Worcester 5, Mass.
Morgan, Reginald William ('41) (CLDFKE), Ch. Engr., Great Lakes Carbon Corp., 333 N. Michigan Ave., Chicago 1, Ill.
Morgan, Robert Clinton (J'43) (BKAJMS), Stress Analyst, Consltd. Vultee Aircraft Corp., Pacific Highway, San Diego; *for mail,* 4623 Mississippi, San Diego 3, Calif.
Morgan, Thos. A. ('29), Pres., Sperry Corp., 30 Rockefeller Plaza, New York, N.Y.
Morgenroth, Robert John ('25; '34) (MCJB), Prod. Engr., Christiana Mch. Co., Christiana, Pa.
Morikawa, George K. (J'39) (ABHK), Research Engr., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.
Morin, Louis H. ('30) (TJMY), Ch. Engr., Crown Fastener Corp., 745—5th Ave., New York 22; *for mail,* 888 Grand Concourse, New York 51, N.Y.
Moritz, Adrianus J. L. ('30), Tech. V.P., Am. Enka Corp., Enka, N.C.
Moritz, Harold K. (J'21) (HBS), Assoc. Prof. Civ. Engrg., Univ. of Wash., Seattle 5, Wash.

Morken, Carl H. ('37) (JDC), Gen. Supt., Centrifugal Casting Co.; *for mail,* 228 E. Central St., Bluffton, Ind.
Morland, Jules Anatole ('44) (BMR), Gen. Mgr., Timken Roller Bearing Co. of S.A., Caixa Postal 208-B, São Paulo, Brazil, S.A.
Moro-Lin, Joseph John ('42; '44) (KLS), Process Design Engr., Colgate-Palmolive-Peet Co., 1 Newark Ave., Jersey City 2, N.J.; *for mail,* 1922 McGraw Ave., New York 62, N.Y.
Moroney, William (J'43) (CKWEA), Sales Engr., Marley Co., Inc., 2 Rector St., New York; *for mail,* 136 Senator St., Brooklyn 20, N.Y.
Moroz, Peter Joseph (J'40) (CMNERH), Engr., E. I. du Pont de Nemours & Co.; *for mail,* 506 Barth Ave., Richland, Wash.
Morrell, Lester William (A'43) (JKMAC), Mgr., Engr., Revere Copper & Brass, Inc., 1010 Pine St., St. Louis 1; *for mail,* 3745 Lindell Blvd., St. Louis 8, Mo.
Morrell, Milton Carson (J'44) (ABS), Research Engr., De Laval Steam Turbine Co.; *for mail,* 2929 Nottingham Way, Trenton, N.J.
Morrill, Frank Baxter ('45), Ch. Engr., James Hunter Mch. Co., 6 Main St.; *for mail,* 518 W. Main St., North Adams, Mass.
Morrin, Earl Hamilton (J'43) (KBHAFZ), Research Engr., College of Engrg., Univ. of Calif., Los Angeles 24, Calif.
Morris, Benj. F. ('30; '35) (CDL), V.P., Div. Mgr., Thomas A. Edison, Inc., Bellville Pike, Kearny, N.J.
Morris, Charles Thomas (J'42), 440 E. Lockwood Ave., Webster Groves, Mo.
Morris, David Lionel (J'42) (BYMA), Asst. Ch. Engr., Circle Wire & Cable Corp., 5500 Maspeth Ave., Maspeth; *for mail,* 42-42 Ithaca St., Elmhurst, L.I., N.Y.
Morris, Glenn L. ('37; '46) (LJK), Engr., Combustion Engrg. Co., Inc., 1032 W. Main St., Chattanooga 1, Tenn.
Morris, Jack Benson (J'44) (ESLFCN), Mech. Research Engr., Acetylene Research Div., Prest-O-Lite Co., Inc., Main & 16th Sts., Indianapolis 8, Ind.
Morris, John Kidd ('35; '44) (ACOMNB), Staff Engr., Engrg. Consultant, Design Devel., West. Gear Works, 2600 E. Imperial Blvd., Lynwood; *for mail,* 10016 Orange Ave., South Gate, Calif.
Morris, Matthew Krause (J'40) (FZKJRS), Field Engr., Combustion Dept., Bethlehem Steel Co., 3rd St., Bethlehem, Pa.
Morris, Richard H. ('36) (FHK), *Student Award,* '21; Engrg. Editor, *Power Plant Engineering,* 53 W. Jackson Blvd., Chicago, Ill.
Morris, Thomas Bray ('10; '26) (CJ), V.P., Sales, United Steel Fabricators, Inc., Wooster, Ohio.
Morris, Thomas Carlyle ('18) (BHL), Mgr. of Engrg., Nitrogen Div., Solvay Process Co., Hopewell, Va.
Morris, Wallace Ronald (J'42) (CLMNO), Ch. Engr., Phin Sales Co., 18 Dickens Ave., Toronto 8; *for mail,* 32 Rumsey Rd., Leaside, Toronto, Ont., Can.
Morris, Walter King (J'45) (ANFBHC), Engr., Weight & Balance Gp., United Air Lines Trans. Corp., U.S. Highway 101, Mun. Airport, South San Francisco; *for mail,* 212 El Camino S., San Mateo, Calif.
Morris, William C. (J'39) (ERS), 708—14th Ave., Prospect Park, Pa.
Morris, Wm. Cullen ('14), 105 Station Rd., Great Neck, L.I., N.Y.
Morris, William Stanley ('24; '32; '35) (BSJMDH), Designer, Mech. Engr., Lunkenheimer Co., Beekman & Waverly Sts., Cincinnati 14; *for mail,* 1742 Portman Ave, Bond Hill, Cincinnati 29, Ohio.
Morrisett, O. L. ('45) (NMJSZY), Ch. Design Engr., Halliburton Oil Well Cementing Co., Duncan, Okla.
Morrish, Murray (J'34) (BFHKSL), Mech. Engr., Lever Bros. Ltd., 299 Eastern Ave., Toronto; *for mail,* Highland Creek, Ont., Can.
Morrison, Burton Wentworth, Jr. (J'45) (MCBJRD), Asst. Ch. Draftsman, Manning, Maxwell & Moore, Inc., 5 Watson St., Boston 18; *for mail,* R.F.D. 2, Concord, Mass.
Morrison, Clyde William (J'43) (AF), Devel. Engr., Wright Aero. Corp., Paterson; *for mail,* 228 Midland Ave., Montclair, N.J.
Morrison, Edwin V., Jr. (J'38), 1st Lt., Co. H, 4th Inf., U.S.A., A.P.O. 947, Seattle, Wash.
Morrison, Jared P. ('16) (S), Cons. Engr., Boiler Div., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford 2, Conn.
Morrison, Joe W. (J'39) (CDGJMN), Charge Engrg., Fabricating Div., West. Cartridge Co., East Alton; *for mail,* 705 Madison Ave., Wood River. Ill.
Morrison, Kenneth L. (J'38) (LAM), Petroleum Engr., Sinclair Refining Co.; *for mail,* Box 51, Sinclair, Wyo.
Morrison, Laurence Albert (J'42) (AJMYCO), Engrg., Aero. Parts Mfg. Co., N. Water St., Wichita; *for mail,* 1407 S. Terrace, Wichita 9, Kan.
Morrison, Richard Tydings, Jr. (J'43) (ESBNAH), Fire Controlman, 2nd Class, "O" Div., U.S.S. *Charles P. Cecil,* DD-835, F.P.O., San Francisco, Calif.; *for mail,* Hampton Ave., Nashville 5, Tenn.
Morrison, Theodore James ('46), Day & Zimmermann, Inc., Packard Bldg., Philadelphia, Pa.
Morrissey, John P. ('24), Pres., John P. Morrissey Elec. Co., Inc., 405 Lexington Ave., New York, N.Y.
Morro, John J. (J'45) (EFKMS), Gen. Mgr., Ch. Engr., Paragon Oil Burner Corp., 50 Van Dam St., Brooklyn 22, N.Y.
Morrow, Cole Harding (J'40) (SFMACZ), Asst. to V.P., J. I. Case Co., 700 State St.; *for mail,* 500 Augusta St., Racine, Wis.
Morrow, John Gordon ('38; '45) (CDKLMS), Plant Engr., Jacob Ruppert Brewery, 1639—3rd Ave., New York 28; *for mail,* 1 Overlook Dr., Port Washington, N.Y.
Morrow, Joseph H. ('26; '30) (BCDKNO), V.P., Fuller Co., Fuller Bldg., Catasauqua, Pa.
Morrow, Lester C. ('16; '24) (CDGM), Editor, *Factory Management and Maintenance,* McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.
Morrow, Robert Lee ('22; '29) (C), Cons. Engr., Lecturer, Dept. of Admin. Engrg., N.Y. Univ., University Heights, New York; *for mail,* 2728 Henry Hudson Pkwy., New York 63, N.Y.
Morse, Benjamin Wyman ('45) (CJW), War Shipping Admin., 39 Broadway, New York 6; *for mail,* 75 State St., Ossining, N.Y.
Morse, Chas. H. ('97; '04), Retired; 600 S. Michigan Ave., Chicago, Ill.
Morse, Daniel P. (J'39) (ACM), Assembly Engr., Allison Div., Gen. Motors Corp., Speedway City; *for mail,* 4001 Guilford Ave., Indianapolis 5, Ind.
Morse, Earl Peaslee (J'44) (CMNHYW), Devel., Research, Shipbldg. Div., Bethlehem Steel Co., Howard St., Quincy; *for mail,* 90 Sherman St., Quincy 70, Mass.
Morse, Edward Phinly, Jr. ('18), Cons. Mar. Engr., 630 Victory Blvd., Staten Island 1, N.Y.
Morse, Francis (J'39) (ABR), Airship Designer, Goodyear Aircraft Corp., Massillon Rd.; *for mail,* Turkeyfoot Island Club, Lahn Dr., Akron 1, Ohio.
Morse, Frederick Baker (J'46), Mech. Engrg. Dept., Purdue Univ., Lafayette, Ind.
Morse, Harold Lane (J'46) (LKDHCM), Tech. Sec., E. I. du Pont de Nemours & Co., Charleston, W.Va.
Morse, Henry S. ('18) (KBCWJY), Retired; 4166 Lindell Blvd., St. Louis 8, Mo.
Morse, James Stanley (J'42) (CMJD), Asst. Engr., Canadian Ingersoll-Rand Co., Ltd.; *for mail,* 56 Moore St., Sherbrooke, Que., Can.
Morse, James W. ('30; '35) (BHC), Ch. Engr., John Bean Mfg. Co., 735 E. Hazel St., Lansing 4; *for mail,* 180 S. Harrison St., East Lansing, Mich.
Morse, Louis Shepard, Jr. ('33; '41) (JLMNTY), Regional Engr., Mann, Roller & Golibart, Inc., 1427 E. Jefferson Ave., Detroit 7, Mich.
Morse, Louis Shepard, Sr. ('32) (BCJLNS), Exec. Engr., York Corp., Roosevelt Ave., York, Pa.
Morse, Ralph Lincoln ('43) (FLSZ), New England Mgr., Republic Flow Meters Co., 201 Devonshire St., Boston 10; *for mail,* 47 Alderwood Rd., Newton Centre 59, Mass.
Morse, Robert Virgil ('22) (AOMC), Pat. Atty., Professional Engr., 521 Wyckoff Rd., Ithaca, N.Y.
Morse, William Howard (J'41) (BJKHS), Mech. Engr., A.C. Gen. Dept., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 9207 Van Tine St., Pittsburgh 21, Pa.
Morss, Charles Anthony ('26; '35) (AS), Designer Aircraft Engs., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford; *for mail,* 91 Ledyard Rd., West Hartford 5, Conn.
Mortimer, Jas. D. ('20) (BES), R.D. 4, Belfast, Me.
Morton, Allen Waller ('28) (AER), V.P., Gen. Mgr., Am. Hammered Piston Ring Div., Koppers Co. Inc., Box 626, Baltimore 3, Md.
Morton, Arthur Benjamin ('17) (HBKSY), Ch. Designing Engr., Thompsons Engrg. & Pipe Co., Castlemaine, Victoria, Australia.
Morton, Byron Benson ('36) (ER), Metal. Engr., Internatl. Nickel Co., 67 Wall St., New York 5, N.Y.
Morton, Earl Eugene (J'42) (ASCYBO), Sr. Aerodynamicist, Chance Vought Aircraft Div., United Aircraft Corp., Stratford; *for mail,* 2290 Madison Ave., Bridgeport 6, Conn.
Morton, Harry E. ('41), 11250 Union Ave., Chicago 28, Ill.
Morton, Henry S. (J'40) (BCH), Lt. Comdr., U.S.N.R., Planning Div., Norfolk Naval Shipyard, Portsmouth, Va.
Morton, Roscoe W. ('24; '30; 32) (EFKLSZ), *Manager,* '42-'45; Prof. & Head, Mech. Engrg., Univ. of Tenn., 100 Estabrook Hall, Knoxville 16, Tenn.
Morton, Stanley ('44) (BCWE), Mgr., Ch. Engr., Renold Coventry Ltd., 1066 Seymour St.; *for mail,* 3370 W. 43rd Ave., Vancouver, B.C., Can.
Morton, Walter Benson ('44) (CMOS), Sta. Elec. Engr., Pa. Power & Light Co., 901 Hamilton, Allentown; *for mail,* 1165 W. Rosemont Dr., Bethlehem, Pa.
Mosbacher, Bruce Henderson (J'42) (B), 434 N. Belmont, Wichita, Kan.
Mosbacher, Karl J., Jr. (J'41) (ACFMS), Sales Engr., Gen. Elec. Co., 1 River Rd.; *for mail,* 201 Seward Pl., Schenectady 5, N.Y.
Mosbrook, Joseph Sylvester ('45) (CDJMRS), Dir., Gen. Services, E. I. du Pont de Nemours & Co., 10th & Market St.; *for mail,* 10 Dickinson Lane, Westhaven, Wilmington, Del.
Moseley, Alex W. ('00), Project Engr., Hanna Engrg. Works, 1765 Elston Ave., Chicago; *for mail,* 2039 Orrington Ave., Evanston, Ill.
Moser, Chris John (J'43) (MSO), Lt., Corps of Engrs., U.S.A., Field Engr., Staff & Faculty Engr. Sch., Ft. Belvoir; *for mail,* 3916 Elbert Ave., Alexandria, Va.
Moser, Kenneth J. ('37; '41) (SKEB), Asst. Prof. Mech. Engrg., Stevens Inst. of Tech., Hudson St., Hoboken; *for mail,* 720 E. 22nd St., Paterson 4, N.J.
Moser, Norman W. (J'37) (SFHEKZ), Mech. Engr., Dairyland Power Cooperative, 417 Rivoli Bldg., La Crosse, Wis.
Moses, Amaziah Jones ('45) (CSMJLF), V.P., Gen. Mgr., Hedges-Walsh-Weidner Div., Combustion Engrg. Co., Inc., 1032 W. Main St., Chattanooga 2, Tenn.
Moses, Eliot B. ('23; '30) (HRANSO), Engr., Hyd. Equip., U.S. Bur. of Reclamation, 355 Custom House, Denver 2, Colo.
Moses, Fred Callender ('18; '26) (SFOZKL), Mech. Engr., West. Elec. Co., Inc., 195 Broadway, New York 7, N.Y.; *for mail,* 130 Stelle Ave., Plainfield, N.J.
Moses, Fred'k T. ('16; '23), V.P., Engr., Fireman's Mutual Ins. Co., 10 Weybosset St. & 560 Lloyd Ave., Providence, R.I.
Moses, Richard (J'41) (ZSYN), Instrument Engr., E. I. du Pont de Nemours & Co., Belle; *for mail,* 219½ Truslow St., Charleston 1, W.Va.
Moses, Warren G. (J'41) (ACK), Aviation Cadet, Class 42-5, Bks. T-187, Aviation Cadet Det., Chanute Field, Ill.
Moses, William G. ('45) (CDLM), Sales Mgr., Rochester Ropes, Inc., 2409 S. 1st Ave., Birmingham 3, Ala.
Moseson, Merland Leonard (J'42) (AES), 4597 Rocky River Dr., Cleveland, Ohio.
Mosher, Fred Donald ('36; '43) (CFMRS), Devel. Engr., Stand. Stoker Co., Inc.; *for mail,* 2613 Atlantic Ave., Erie, Pa.
Mosher, Harold Arthur ('43) (LKCZYD), Supvr. Engr., Kodak Park Works, Eastman Kodak Co., Rochester 13, N.Y.
Moshkoff, Simon V. ('28) (HSMD), Plant Supt., Ga. Power Co., 75 Marietta St., Atlanta 1; *for mail,* Furman Shoals, Milledgeville, Ga.
Moss, Donald Frederick (J'44), Ensign, U.S.N.R., U.S.S. *Power,* DD-839, New York 27, N.Y.; *for mail,* 80 Mt. Vernon St., Melrose 76, Mass.
Moss, Edward Harry, Jr. (J'37) (BEHJKN), Engr., Mech. Design, Engrs. & Fabricators, Inc., Box 7395, Houston 8, Tex.
Moss, Herbert Henry ('17; '35) (BJMDL), Supvr. Process Trng., Linde Air Products Co., 686 Frelinghuysen Ave., Newark 5, N.J.
Moss, Howard Kenneth (J'44) (AHK), Design Aircraft, McDonnell Aircraft Corp., Laclede & Sarah, St. Louis, Mo.; *for mail,* R.F.D. 2, Hillsboro, Ill.
Moss, Sanford Alexander ('03; F'40) (ABEHKS), *Holley Medallist,* '45; Cons. Engr., Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.
Moss, Williamson Wade, Jr. ('45) (JL), East. Editor, *Light Metal Age,* 201 N. Wells St., Chicago 6, Ill.; *for mail,* 5010 Forest Park Ave., Baltimore 7, Md.
Mossberg, Frank ('95), Pres., Mossberg Pressed Steel Corp.; *for mail,* 141 Pleasant St., Attleboro, Mass.
Motch, Edwin Raymond, Jr. ('42) (BHM), Lt. Col., Ord. Dept., Rm. 4C-628, The Pentagon, Washington, D.C.
Motheral, Harry Haviland ('25; '35) (HK), Maint. Supt., W. T. Grant Co., 1441 Broadway, New York, N.Y.
Motherwell, George Washington ('44) (JMCADH), Works Mgr., Wyman-Gordon Co., 105 Madison St., Worcester 1; *for mail,* 10 Aylesbury Rd., Worcester 2, Mass.
Mott, Charles S. ('98; '09), Dir., Gen. Motors Corp., Detroit; *for mail,* 500 Mott Foundation Bldg., Flint 3, Mich.
Mott, Gilbert Culver (J'37) (JCBMN), Lt., U.S.N.R., Engrg. & Inspc. Officer, Naval Ord. Plant, Center Line, Mich.
Mott, Harold Edgar ('42), Pres., c/o H. E. Mott Co. Ltd., Brantford, Ont., Can.
Mouat, Harry George ('23; '35) (CDJMRY), So. Mgr., Whiting Corp., 544 Martin Bldg., Birmingham 3, Ala.
Mould, Arthur Elias (A'21) (ESLC), Sales Engr., Tide Water Associated Oil Co., 17 Battery Pl., New York; *for mail,* 22 Highland Dr., Williamsville 21, N.Y.
Moulenbelt, Robert C. (J'43) (SLEKBH), Mech. Engr., Retort Chem. Div., Cabot Carbon Co., Gainesville, Fla.
Moulton, Clarence F. ('26; '33; '35) (SCFAEO), Supt. of Power, Charge Power Prod., Neb. Power Co., 17th & Harney, Omaha 2; *for mail,* 2316 Fontenelle Blvd., Omaha 3, Neb.
Moulton, Lloyd Jackson (J'43) (OME), R.F.D. 1, Grove Dr., Mentor, Ohio.
Moulton, Rexford Glen (J'43) (KFBM), Asst. Prof. Mech. Engrg., Dartmouth College, Hanover, N.H.

Moultrop, Irving E. ('02; F'36) (EFS), *Manager*, '08-'11, *Vice-President*, '12-'14; Cons. Engr., 28 Adams St., Belmont 78, Mass.
Mounce, Alfred Richard (J'42) (KNJY), Design Engr., Gen. Elec. Co., 1605 Winter St., Ft. Wayne, Ind.
Mount, Ralph H. ('17) (C), Asst. to Pres., Essex Wire Corp., 14310 Woodward Ave., Detroit; *for mail*, 132 Crane Ave., Royal Oak, Mich.
Mousson, Jean Marcus, 2nd ('37) (HSCO), Bartlett Hayward Div., Koppers Co., 200 Scott St., Baltimore 30; *for mail*, 1 E. University Pkwy., Baltimore 18, Md.
Mowat, J. Fred ('16) (OJFSDE), Engr., Chicago Dist., Carnegie-Ill. Steel Corp., 208 S. La Salle St., Chicago 90; *for mail*, 538 S. Park Rd., La Grange, Ill.
Mowat, Magnus ('38; F'38), Brig. Gen., Ebor House, 26 Sheen Gate Gardens, London, S.W. 14, England.
Mowatt, Wm. T. ('28), 2851 Johnson Ave., Alameda, Calif.
Mowlem, Aboudi Robin (J'45) (BCK), Mass. Inst. of Tech. Dormitories, 104 Nichols, Cambridge 39, Mass.
Moxham, Egbert ('06; '12), c/o Brunswick Mar. Constr. Corp., Brunswick, Ga.
Moxley, Stephen Dewey ('24; '26; '30) (BCDJ), Ch. Engr., Am. Cast Iron Pipe Co., Box 2603, Birmingham 2, Ala.
Moxon, Alfred William ('39; '44) (BCM), Engr., Div. Supvr., Eastman Kodak Co., Kodak Park, Rochester 13; *for mail*, 211 Park Rd., Point Pleasant, N.Y.
Moyer, Charles Robert (J'42) (ACBKFO), Asst. Project Engr., Engrg. Planning, Wright Aero. Corp., Paterson; *for mail*, 23 Maple Pl., Clifton, N.J.
Moyer, Malcolm Bridgeman ('28) (BEKLMO), Cons. Engr., 116 S. Salina St., Syracuse 2; *for mail*, 214 Ruhaman Ave., Syracuse 5, N.Y.
Moyer, Robert Eugene, Jr. ('29; '38) (KOLFSH), V.P., Charge Engrg., Heilman Boiler Works, Inc., 132 Linden St., Allentown, Pa.
Moyer, Robert George (J'40) (JM), Research Engr., Pure Oil Co., Box 266, Winnetka; *for mail*, 2237 Foster Ave., Chicago, Ill.
Moyer, Stanley ('41) (BHS), Sr. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Mroz, Edward John (J'43), 59 Fulton St., Lowell, Mass.
Mrvosh, John (J'37), 20 Emory St., Jersey City, N.J.
Mucha, Rudolph S., Jr. (J'46) (LMY), 603—9th St., Charleroi, Pa.
Muchnic, Chas. M. ('27), Suite 2048, 420 Lexington Ave., New York, N.Y.
Mudarri, Edward George (J'45) (MCB), Rm. 928, U.S.N.S.H., Seagate, Brooklyn, N.Y.; *for mail*, 66 Franklin St., Allston, Mass.
Mudd, John Posey ('10; '15; '35) (CG), Personnel Supvr., Midvale Co., Philadelphia 40; *for mail*, 173 Manheim St., Philadelphia 44, Pa.
Mudge, Robt. S. (J'35) (BMS), Pres., Palo-Myers, Inc., 81 Reade St., New York; *for mail*, 856 Park Pl., Brooklyn, N.Y.
Mudge, Sterling W. ('14; '20; '24) (CE), Dir. Trng., Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4; *for mail*, 12 The Place, Glen Cove, L.I., N.Y.
Muehlman, Raymond Louis (J'40) (LMWYS), Power Supt., Belle Alkali Co., Belle; *for mail*, 1116 Highland Rd., Charleston 2, W.Va.
Mueller, Alfred Charles ('45) (BHK), Gp. Leader, Exper. Sta., E. I. du Pont de Nemours & Co., Wilmington, Del.
Mueller, Charles August ('43) (BCDLMN), Mech. Engr., Campbell Soup Co., 100 Market, Camden, N.J.; *for mail*, 102 Dartmouth Rd., Bala-Cynwyd, Pa.
Mueller, Edmund J. (J'41), U.S. Navy; 6412 Park Ave., West New York, N.J.
Mueller, Felix J. ('24) (BJNS), Engr., Ch. Draftsman, Allis-Chalmers Mfg. Co., West Allis; *for mail*, 1558 S. 58th St., West Allis 14, Wis.
Mueller, Frank Hieronymus ('39) (MNHBC), Dir., Research & Devel., Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller, Hans Siegfried (J'42) (ACM), Lt., Army Air Forces; *for mail*, 254 Oelkers Ave., North Tonawanda, N.Y.
Mueller, Herman Gundert ('24) (OMNEHS), Owner, Firm of H.G. Mueller, Engrs., 623-4 Commerce Bldg., Erie, Pa.
Mueller, Lester A. (J'38) (BEF), Asst. Mgr., Mueller Grain & Elevator Co., Roscoe, Tex.
Mueller, Paul Manning ('23; '24; '35) (JC), Mgr., Engrg. & Research, Pratt & Whitney Div., Niles-Bement-Pond Co., West Hartford, Conn.
Mueller, Raymond Albert (J'40) (JSKBNO), Welding Research Engr., Crane Co., 4100 S. Kedzie Ave., Chicago 32, Ill.
Mueller, Robert Karl ('42) (ABZ), Project Engr., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Mueller, Victor H. ('07; '21) (EFHKS), Mech. Engr., Gibbs & Hill, Inc., Pa. Sta., New York, N.Y.; *for mail*, 25 Maple Terrace, East Orange, N.J.
Mueller, William Christian ('28) (BCMN), Mfg. Engr., Hawthorne Plant, West. Elec. Co., Inc., Chicago 23; *for mail*, 2225 Giddings St., Chicago 25, Ill.
Muench, Erwin John (J'42), V.P., Gen. Mch. Co., 398 Market St., Newark 5; *for mail*, 599 Sanford Ave., Newark, N.J.
Muench, Frank Joseph, Jr. (J'40) (CJ), Gen. Mgr., Universal Copper Works, 156 Walnut St., Newark; *for mail*, 171 Sterling Dr. N., Orange, N.J.
Muenger, James Robert (J'41) (EFBAN), Mech. Engr., Beacon Research Labs., Tex. Co., Beacon, N.Y.
Muerle, Richard W. ('35) (BCHJMN), Ch. Product Engr., Hummer Mfg. Co., 1400 S. 9th St.; *for mail*, 1829 Outer Park Dr., Springfield, Ill.
Muessel, Charles Andrew, Jr. ('36; '45) (CSARMK), Asst. Mgr., Spec. Projects Dept., Westinghouse Elec. Internatl. Co., 40 Wall St., New York 5, N.Y.; *for mail*, R.D. 4, Yale Farms, Greenwich, Conn.
Mugfor, Richard J. ('37) (BSK), Ch. Engr., Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, Ohio.
Muir, James Fleming ('20) (CEFGOS), Elec. Properties Consultant, Am. Water Works & Elec. Co., 50 Broad St., New York 4, N.Y.
Muir, John Nimmo (J'44) (DHJMNR), Tool Engr., Natl. Steel Car Corp. Ltd., Kenilworth Ave. N.; *for mail*, 175 Rosslyn Ave. S., Hamilton, Ont., Can.
Muir, Roy Cummings ('17; F'41) (ACS), V.P., Gen. Mgr. Apparatus Dept., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Muir, William P. ('39) (BCJMNY), Mgr., Ch. Engr., Indus. Div., Dominion Engrg. Works Ltd., Box 220, Montreal, Que., Can.
Mukerji, Manik Lal ('45) (COT), Ch. Mech. Engr., Hindustan Constr. Co., Ltd., Ballard Estate, Fort, Bombay; *for mail*, c/o Capt. (Dr.) S. K. Mukerji, Manoramaganj, Indore, Central India.
Mulak, Julian Peter (J'45) (SKBMCD), U.S. Navy, Springfield Rd., Hazardville, Conn.
Mulford, Stewart Furman (J'40) (KRAFE), Ch. Test Engr., S. Wind Div., Stewart-Warner Corp., 1514 Drover St., Indianapolis 7; *for mail*, 51 W. Fall Creek Pkwy., Indianapolis 8, Ind.
Mulheim, Joseph E. (J'43) (BAK), Mech. Design Engr., Westinghouse Elec. Corp., Lima, Ohio.
Mulholland, Roye Artice (A'45) (CHKLST), Mech., Civ. & Elec. Engr., Tex. State Dept. of Health, 410 E. 5th, Austin; *for mail*, 1006 Harper Lane, Austin 22, Tex.
Mullaly, Arthur B. (J'38), Sales Engr., Advance Solvents & Chem. Co., New York, N.Y.; *for mail*, 70 Monmouth Dr., Deal, N.J.
Mullan, Edward (J'42) (BES), Graduate Student, Westinghouse Elec. Corp., East Pittsburgh, Pa.; *for mail*, 29 Bailey Ave., Hillside, N.J.
Mullee, William Robert (J'23) (CDLYMN), Indus. Engr., Am. Hard Rubber Co., 11 Mercer St., New York 13, N.Y.; *for mail*, 125 Pompton Ave., Pompton Lakes, N.J.
Mullen, Bernard J. ('36) (SHCFD), Ch. Plant Engr., Detroit Dept. of Water Supply, 8300 W. Warren, Dearborn; *for mail*, 17534 Birchcrest Dr., Detroit 21, Mich.
Mullen, George Hamilton (A'44) (SEJB), Supvg. Engr., Ocean Accident & Guarantee Corp., Kansas City; *for mail*, 1727 Fed. Reserve Bank Bldg., Kansas City 6, Mo.
Mullen, John Joseph ('44) (DHOMB), Designing Engr., Jones & Henry, Security Bldg., Toledo 4; *for mail*, 2042 Fulton St., Toledo 2, Ohio.
Mullen, John O. (J'40), 444 Ellsworth Ave., New Haven, Conn.
Mullen, John Patrick ('45) (BNKLAF), Sr. Lecturer, Univ. College, Univ. of London, Gower St.; *for mail*, 26 Holmwood Grove, Mill Hill, London, N.W. 7, England.
Mullen, Lester Herbert (J'37) (KHEFBS), Mar. Engr., U.S. Maritime Comm., 14th & Franklin St., Oakland; *for mail*, 863 Dutton Ave., San Leandro, Calif.
Mullen, Robert Marion ('44), Steam & Elec. Engr., Pittsburgh & W. Va. Ry. Co., 410 Liberty Ave., Pittsburgh 22; *for mail*, 3723 Baytree St., Pittsburgh 14, Pa.
Mullen, Thomas Yeatts (J'37) (SEFKO), Project Engr., Gilbert Associates Inc., 412 Washington St., Reading, Pa.
Muller, Daniel Lewis ('34; '34; '35) (ABEKSY), Boiler & Mchy. Dept., Marsh & McLennan Inc., 70 Pine St., New York 5, N.Y.
Muller, Edw. A. ('90; F'36) (BCM), *Manager*, '24-'27, *Vice-President*, '27-'29; Pres., Treas., Gen. Mgr., King Mch. Tool Co., Winton Pl., Cincinnati, Ohio.
Muller, Ellsworth August (J'41) (SHBAKY), Assoc. Mar. Engr., Bur. of Ships, Navy Dept., Code 640, Bldg. T4-2204, Washington, D.C.
Muller, Frank George (J'44) (CJM), Graduate Student, Univ. of Tex., Austin; *for mail*, 3008 N. Guadalupe St., Austin 21, Tex.
Muller, Frans G. D. ('30) (CM), V.P., Plant Mgr., Staytite Co., 3608 Polk Ave., Houston 3; *for mail*, Box 976, La Porte, Tex.
Muller, George John (J'42) (YCBMLN), Dir. of Engrg., Taylor Fibre Co., Norristown; *for mail*, 3344 Ainslie St., Philadelphia 29, Pa.
Muller, Herman Ernest (J'39) (BJNS), Roll Designer, Carnegie-Ill. Steel Corp., Salt Springs Rd.; *for mail*, 131 S. Whitney Ave., Youngstown, Ohio.
Muller, Julius ('27; '28; '35) (BJR), Welding Engr., O. C. Duryea Corp., 30 Rockefeller Plaza, New York, N.Y.; *for mail*, 10743 S. Wood St., Chicago, Ill.
Muller, Otto ('33) (OMJCA), Div. Supt., Am. Optical Co., Mechanic St.; *for mail*, 44 Newell Ave., Southbridge, Mass.
Muller, Richard F. ('21; '29; '35) (HLM), Sales Engr, Allis-Chalmers Mfg. Co., 724 Canal Bldg., New Orleans 12, La.
Muller, Richard John (J'40) (MNACJY), Mech. Engr, David Taylor Model Basin, Washington 7, D.C.; *for mail*, Main St., Bethesda 14, Md.
Muller, Robert Arthur ('24; '37) (CWS), V.P., Dir., Atlas Plywood Corp., 1432 Statler Bldg., Boston 12; *for mail*, 29 Albion Rd., Wellesley Hills 82, Mass.
Muller, William George (J'44), Army Air Forces, 109-27—221st St., Queens Village, L.I., N.Y.
Mullergren, Arthur Leonard ('18; '28) (OSC), Cons. Engr., 202 Fairfax Bldg., Kansas City 6, Mo.
Mullhaupt, Alfred, Jr. (J'11), Engr., Bradford Oil Refining Co.; *for mail*, 25 School St., Bradford, Pa.
Mulligan, James Henry (J'42) (ZFKLCO), Sales Engr., Johnson Serv. Co., 507 E. Michigan St., Milwaukee 2; *for mail*, 118 N. Birdsey St., Columbus 2, Wis.
Mulligan, Paul B. ('35; '42) (ACD), Staff Asst., U.S. Steel Corp., 436—7th Ave., Pittsburgh; *for mail*, 1158 Pinewood Dr., Mt. Lebanon, Pittsburgh 16, Pa.
Mullikin, Harwood Franklin ('30; '41) (FOSCK), *Junior Award*, '36; 409 Weaver St., Larchmont, N.Y.
Mullin, William Howarth (J'44) (KMCDB), Prod. Engr., Philco Corp., Box 453, Tecumseh, Mich.
Mullins, Euthan V. (J'41) (CDS), 1968 Jasmine St., Denver, Colo.
Mullis, Clyde Monroe (J'42) (ABEJNZ), Sr. Test Engr., Wright Aero. Corp., Paterson 3; *for mail*, 470 Prospect St., Ridgewood, N.J.
Multhaup, Robert Hansen (J'44) (BSKCEH), Prod. Supvr., Research & Devel. Dept., Elliott Co., Jeannette, Pa.
Mulveny, Frank, Jr. (J'37) (CFS), Mech. Engr., Edge Moor Iron Works Co., Edge Moor; *for mail*, 9 Polk Rd., Edge Moor Gardens, Wilmington, Del.
Mumford, Albert Russell ('19; '24; '27; F'42) (SFKBCE), Devel. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Mumford, Stephen F. (J'40) (FKS), Mar. Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 191 Ocean Ave., Lynbrook, N.Y.
Mumma, Milton Christian ('42) (EDR), Indus. Lub. Serv. Engr., Gulf Oil Corp., Pittsburgh, Pa.; *for mail*, 5409 Staunton Ave., Charleston 4, W.Va.
Mummery, Charles Rutherford ('30; '35) (BMZCT), Sec. Head, Mech. & Cleaner Research, Hoover Co.; *for mail*, 606 E. Maple St., North Canton, Ohio.
Munck, Ludwig R. ('38) (FS), Sales Engr., Foster Wheeler Corp., 634 Michigan Bldg., Detroit; *for mail*, 14803 Greenview Ave., Detroit 23, Mich.
Mundel, Marvin E. (J'43) (CMWO), Asst. Prof. Indus. Engrg., Purdue Univ., Lafayette, Ind.
Mundy, James T. (J'39) (BCJNTZ), Capt., Ord. Dept., U.S.A., Chief, Admin. Branch, Ord. 4th & 5th Ech., Auto Shop, Atlanta Ord. Depot; *for mail*, 1677 Havilon Dr., S.W., Atlanta, Ga.
Munger, Milton Potter, Jr. (J'37) (ZYHS), Assoc. Mech. Engr., P3, Indus. Lab., Mare Island Naval Shipyard, Vallejo; *for mail*, 2829 Regent St., Berkeley 5, Calif.
Munier, Leon Lucien ('21; '26) (FKO), Pres., Wolff & Munier Inc., 222 E. 41st St., New York 17, N.Y.
Munn, Harvey Timlow ('43) (HSCEGO), Hyd. Engr., Natl. Bd. of Fire Underwriters, 222 W. Adams St., Chicago 6, Ill.
Munn, Hugh Frisbie ('44) (COWK), V.P., Engrg. & Mgmt., Tech. Serv. Co., 501 N. 1st St.; *for mail*, 419 W. Granite Ave., Albuquerque, New Mex.
Munro, Robert W. (J'24) (BEFHKS), Dist. Mgr., Griscom-Russell Co., 1317 Land Title Bldg., Philadelphia 10, Pa.
Munroe, Harold David (J'42) (NMKH), Ord. Engr., Naval Shipyard, Mare Island; *for mail*, 2538 Durant Ave., Berkeley 4, Calif.
Munschauer, Frederick Eugene, Jr. (J'43) (NMJCH), Mgr. of Plant & Indus. & Prod. Engrg., Niagara Mch. & Tool Works, 683 Northland Ave., Buffalo 11, N.Y.
Munsell, David Dows (J'43) (AEM), Ensign, Deer Park Rd., Reisterstown, Md.
Munson, John G. ('17) (J), V.P., U.S. Steel Corp. of Del., 436—7th Ave., Pittsburgh 30, Pa.
Munson, Lloyd Emery (J'41) (ABN), Structures Gp. Engr., Consltd. Vultee Aircraft Corp.; *for mail*, Apt. 156, 3100 Darnell, Ft. Worth, Tex.
Munson, Stanley ('11; '25) (DFLS), Kirkman & Son Div., Colgate-Palmolive-Peet Co., 215 Water St., Brooklyn 1; *for mail*, 89-11—198th St., Hollis 7, N.Y.
Munz, Charles Arthur (J'43) (NMAB), Owner, Ch. Engr., Munz Engrg. Serv., 704 S. Spring St., Los Angeles 14, Calif.
Murayama, Sutemi (J'43) (NLJHDA), Ch. Draftsman, Indus. Washing Mch. Corp.; *for mail*, 145 Welton St., New Brunswick, N.J.
Murphy, Benjamin Stuart ('05; '16) (SFKCR), Asst. Mgr., Elec. Prod. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.; *for mail*, 2600 Boulevard, Jersey City 6, N.J.

Murphy, Benton Franklin, Jr. (J'42) (B), Ensign, O-V(S), U.S.N.R.; *for mail*, Rivermines, Mo.

Murphy, Carroll T. G. (J'45) (BHNOSC), 820 W. 31st St., Wilmington, Del.

Murphy, E. Landry (J'40) (CKBA), Ch. Engr., Mech. Equip. Co., 861 Carondelet St.; *for mail*, 3223 Jena St., New Orleans 15, La.

Murphy, Edward Ross ('45) (BCJMG), Asst. to Ch. Designer, Am. Can Co., 320 Elizabeth Ave., Newark 8; *for mail*, 195 Prospect St., East Orange, N.J.

Murphy, Edward Thomas ('05; '17) (KLT), Sr. V.P., Carrier Corp., 300 S. Geddes St., Syracuse 1, N.Y.

Murphy, Eugene Francis (J'35) (BEKN), Staff Engr., Com. on Prosthetic Devices, Natl. Acad. of Sciences, 355 Northwest. Tech. Inst., Evanston, Ill.

Murphy, Frank L. ('42) (RJBY), Ch. Engr., Charge Prod., Pullman-Stand. Car Mfg. Co., 10909 Cottage Grove Ave., Chicago 28, Ill.

Murphy, George Francis ('26; '35) (ABKNRY), Design Engr., Linde Air Products Co., 086 Frelinghuysen Ave., Newark 5, N.J.

Murphy, George W. (J'41) (SCA), Supvg. Serv. Engr., Westinghouse Elec. Corp., 207 W. 3rd St., Cincinnati 2; *for mail*, 3812 St. John Terrace, Deer Park 13, Ohio.

Murphy, Glenn ('45) (ABH), Prof. Theoretical & Applied Mechanics, Iowa State College, Ames, Iowa.

Murphy, Gordon ('43) (EFB), Ch. Engr., Research Div., Fairbanks, Morse & Co., Lawton Ave., Beloit, Wis.

Murphy, Harold William (J'44) (BKHZAN), Devel. Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1; *for mail*, 413½ Rich St., Syracuse 7, N.Y.

Murphy, Howard Charles ('32) (CEKRTZ), V.P., Am. Air Filter Co., Inc., 215 Central Ave., Louisville 8, Ky.

Murphy, John Vance (J'40) (ABJ), 235 W. Main St., Wilmington, Ohio.

Murphy, John Wilton ('45) (MCJLON), Project Engr., West. Elec. Co., Inc., 300 Central Ave., South Kearny; *for mail*, 272 Chestnut St., Nutley 10, N.J.

Murphy, Marshall Edward (J'42) (CJD), Engr., Job Evaluation, Bridgeport Brass Co., 774 East Main St., Bridgeport; *for mail*, 83 Sands Pl., Stratford, Conn.

Murphy, Moyes Joseph ('45) (EJR), Comdr., U.S.N.R., Project Engr., Bur. of Ord., Navy Dept., Washington, D.C.; *for mail*, 1554 Mount Eagle Pl., Alexandria, Va.

Murphy, Robt. E. ('36) (CDM), Plant Engr., Am. Chicle Co., 30-30 Thomson Ave., Long Island City; *for mail*, 144-46—29th Ave., Flushing, N.Y.

Murphy, Robert Joseph ('27; '43) (FS), Mech. Supt., Consltd. Edison Co. of N.Y., Inc., 201st St. & 9th Ave., New York, N.Y.; *for mail*, 133 Claremont Ave., Jersey City 5, N.J.

Murphy, Thos. R. H. ('11; '16; '35) (CLS), R.F.D. 1, Ridgefield, Conn.

Murphy, Timothy Henry ('42) (ENRFAJ), Supt. of Diesel Power, Am. Loco. Co., Schenectady 5, N.Y.

Murphy, Walter B. ('16; '35) (JCM), Asst. Works Mgr., Messinger Bearings Inc., D St. above Erie Ave., Philadelphia; *for mail*, 4211 Sansom St., Philadelphia 4, Pa.

Murphy, Miss Wilburda Miriam (J'44) (K), Jr. Engr., York Corp., Roosevelt Ave.; *for mail*, 722 Roosevelt Ave., York, Pa.

Murphy, William James ('27) (S), Constr. Supvr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark; *for mail*, 233 Sherman Ave., Glen Ridge, N.J.

Murr, Charles Howard (J'40) (CDJMBN), Methods Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia; *for mail*, 905 Pasadena Ave., Philadelphia 11, Pa.

Murray, Arthur Frederic ('08; '15; '19) (MCD JLN), Works Mgr., Electrolux Corp., Forest Ave., Old Greenwich, Conn.

Murray, Arthur Warden (J'40) (MCD), 1st Lt., Army Air Forces Weather Serv.; *for mail*, 167 High St., Passaic, N.J.

Murray, Aubrey S. (J'41) (CJM), Ch. Draftsman, N. Am. Rayon Corp.; *for mail*, Box 109, Route 4, Elizabethton, Tenn.

Murray, Edwin D. ('32) (MOCS), Comdr., U.S.N.R., 6389 Florio St., Oakland 11, Calif.

Murray, Eugene Walter ('43) (HKS), St. Louis Dist. Mgr., Spencer Turbine Co., 448 New Park Ave., Hartford 6, Conn.; *for mail*, 6224 Magnolia Ave., St. Louis 9, Mo.

Murray, Frederick Forest ('21; '24; '30) (CEM RDJ), Pres., Oil Well Supply Co., Box 478, Dallas 1, Tex.

Murray, Graham Francis James ('44) (EFS), Asst. Ch. Engr., British Coal Utilisation Research Assn., 13 Grosvenor Gardens, London, S.W. 1; *for mail*, 8 Homewood Rd., St. Albans, Herts, England.

Murray, Julian Ronald (J'38) (SEKOFH), Sales Engr., Babcock & Wilcox Co., 105 S. La Salle St., Chicago 3; *for mail*, 511 Lee St., Evanston, Ill.

Murray, Thos. E. ('32), Metro. Engrg. Co., 1250 Atlantic Ave., Brooklyn, N.Y.

Murray, Willard Augustus (J'45) (ABHNCG), Flight Test Engr., Beech Aircraft Corp.; *for mail*, 223 Cleveland Ave., Wichita, Kan.

Murray, William F. ('22; '31) (CLF), Pacific Portland Cement Co., Rm. 9, Manchel Bldg., Grants Pass, Ore.

Murray, William H. G. ('43) (KCSOL), Pres., Smith-Murray Corp., 821 Burnet Ave., Syracuse 3, N.Y.

Murray, William MacGregor ('35; '41) (BA JRO), Assoc. Prof. Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.

Muschamp, George M. ('43) (CZMLBK), V.P., Charge Engrg., Brown Instrument Co., Wayne & Roberts Aves., Philadelphia 44; *for mail*, Rodney St. above Gorgas Lane, Philadelphia, Pa.

Muschenheim, Fred'k A. ('15), Pres., Hotel Astor, Times Sq., New York, N.Y.

Muschett, Wilbery Ramsay (J'44) (EFKRSZ), Mech. Engr., Designer, Detailer, N.Y. Cent. System, Rm. 858, 466 Lexington Ave., New York 17, N.Y.; *for mail*, 1471 Teaneck Rd., West Englewood, N.J.

Muse, Harold Taylor (J'45) (AFS), 2368 Victory Pkwy., Cincinnati 6, Ohio.

Musham, William C. (J'38), Imperial Brass Mfg. Co., 1200 W. Harrison St., Chicago 7, Ill.

Muska, Vojtech Frank (J'45) (BKL), 34-30—28th St., Long Island City 1, N.Y.

Mustin, Gilbert B., Jr. (J'42), Instr., Mech. Engrg. Dept., Swarthmore College, Swarthmore; *for mail*, Herford Pl., Lansdowne, Pa.

Muth, Raymond F. (J'38), Naval Gun Factory, 8th & M Sts., S.E.; *for mail*, 1830 T St., S.E., Washington, D.C.

Mutter, Edward Charles ('42) (SBEM), Pres., Treas., E. C. Mutter Constr. Co., Inc., 316 Turks Head Bldg., Providence 3, R.I.

Muzik, Victor Klifton ('24; '30; 35) (ANORS), Devel. & Analyst Engr., Airquipment Co., Div. of Lockheed Aircraft Corp., 15 E. Verdugo Rd., Burbank; *for mail*, 2011 N. Cheremoya Ave., Hollywood 28, Calif.

Myatt, Donald Joseph (J'46) (CJM), 2368 Tyler Lane, Louisville 5, Ky.

Myatt, Enoch (J'41) (CME), Gen. Mgr., E. Myatt & Co. Ltd., 21 Carlaw Ave., Toronto, Ont., Can.

Myers, Arthur Fred (J'46), 52 Regional Dr., Penns Grove, N.J.

Myers, Arthur Herbert ('40) (CDGM), Asst. Ch., Internatl. Broadcasting Div., Office of Internatl. Information & Cultural Affairs, Dept. of State, 224 W. 57th St., New York 19; *for mail*, 12 E. 61st St., New York 21, N.Y.

Myers, C. O. ('30), Secy., Treas., Natl. Bd. of Boiler & Pressure Vessel Insprs., 145 N. High St., Columbus, Ohio.

Myers, Charles G., Jr. (J'40) (SEYC), Power House Design Engr., Celanese Corp. of Am., 290 Ferry St., Newark 5; *for mail*, 409 Lincoln Park E., Cranford, N.J.

Myers, Charles Louis, Jr. (J'38) (EBM), Fireman, 1st Class, Supply 116, Naval Aviation Tech. Trng. Center, Norman, Okla.; *for mail*, 616 York Ave., Lansdale, Pa.

Myers, Curtis Clark ('05; '20) (ABCJNZ), Res. Dir., Daniel Guggenheim Airship Inst., 1300 Triplett Blvd., Akron 6; *for mail*, 81 Mull Ave., Akron 3, Ohio.

Myers, David Moffat ('07; '12; F'38) (SFK), Partner, Myers, Fuller & Addington, 21 E. 40th St., New York 16, N.Y.

Myers, Elmer Francis (J'42), Lt., 118 Long Hill St., Springfield, Mass.

Myers, Elwood Ray (J'43), Box 201, Chester, Conn.

Myers, Frank M. ('29; '35) (BJS), c/o Supvr. of Shipbldg., U.S.N., 11 Broadway, New York; *for mail*, 39-40—51st St., Woodside, L.I., N.Y.

Myers, Glenn A. ('44) (CSF), Asst. Mgr. Power Prod., Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City 10, Mo.

Myers, Grafton Stuart, Jr. ('39; '89) (SFCB EH), Sales Engr., Jos. W. Eshelman & Co., 1116 Martin Bldg., Birmingham 3, Ala.

Myers, Henry G. ('30; '41) (BEJKL), Foster Wheeler Corp., 165 Broadway, New York; *for mail*, 96 Coolidge St., Malverne, L.I., N.Y.

Myers, Henry Joseph (J'46), 342 E. 67th St., New York, N.Y.

Myers, Jas. (A'26), Secy., Indus. Div., Fed. Council of Churches of Christ in Am., 297—4th Ave., New York, N.Y.

Myers, John Beasley (J'41) (A), Engr., Glenn L. Martin Co., Baltimore 3; *for mail*, Odenton, Md.

Myers, Maurice Earl ('45) (OCWND), Cons. Engr., 1205 Am. Bank Bldg., Portland; *for mail*, 5305 N. Kerby, Portland 7, Ore.

Myers, Oscar Bruce, Jr. (J'45) (HAOMC), Design Engr., with Dalmo Victor, San Carlos; *for mail*, 915 S. Fremont, San Mateo, Calif.

Myers, Phillip Samuel (J'42) (EFACKB), Instr., Mech. Engrg. Dept., Univ. of Wis., 1518 University Ave., Madison 5, Wis.

Myers, Robert Duane ('21; '29) (AFS), Lt. (j.g.), CEC-V(S), U.S.N.R., 67th Naval Constr. Bn., F.P.O., San Francisco, Calif.

Myers, Robert Thomas ('44) (NMCJLK), Ch. Engr., Sprague-Sells Div., Food Mchy. Corp.; *for mail*, 714 S. 4th St., Hoopeston, Ill.

Myers, Thomas Gary ('32) (NBCHYD), V.P., Ch. Engr., U.S. Elec. Motors, Inc., 200 E. Slauson Ave., Los Angeles 54, Calif.

Myers, William Joseph (J'45) (SKENZF), Graduate Mech. Engrg. Student, Graduate House, Mass. Inst. of Tech., Cambridge 39, Mass.

Myers, William Kurtz ('21) (CJR), Pres., Penn Steel Castings Co., Front & Penn Sts., Chester, Pa.

Myhre, Emmett Bernard (J'36) (SHF), Engr., U.S. Bur. of Reclamation; *for mail*, 1216—5th, Boulder City, Nev.

Mylroie, John Ellis ('36; '45) (AC), Field Serv. Engr., Boeing Aircraft Co., Seattle 8; *for mail*, Box 70, Route 2, Renton, Wash.

Mylting, Lauritz Emil ('37) (HMCDN), Ch. Engr., Allen-Sherman-Hoff Co., 225 S. 15th St., Philadelphia, Pa.

Mynderse, Chauncey Nicklin ('25) (BCMZ), V.P., Fulton Sylphon Co., Kingston Pike, Knoxville 4; *for mail*, Box 263, 2008 Terrace Ave., Knoxville 16, Tenn.

N

Nabow, David ('41) (BHS), Ch. Designing Engr., Duke Power Co., 420 S. Church St., Charlotte 1, N.C.

Nace, John Foley (J'41) (SEO), Application Engr., Fed. & Mar. Div., Gen. Elec. Co., Schenectady; *for mail*, 1737 East. Pkwy., Schenectady 8, N.Y.

Nachman, Harry Simon (J'44) (CJK), Propr., H. S. Nachman & Associates, 179 W. Washington St., Chicago 2, Ill.

Nachman, Henry L. ('21) (SKB), Prof. Thermodynamics, Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.

Nacy, Robert James (J'45) (BDJKMS), Plant Engrg. Draftsman, Murray Corp. of Am., 7700 Russell, Detroit 21, Mich.

Nadai, A. L. ('29) (ABJKSY), Cons. Mech. Engr., Westinghouse Research Labs., East Pittsburgh, Pa.

Nadeau, Robert Frederick (J'39) (CMTY), Prod. Engr., Tubbs Cordage Co., 22nd & Iowa Sts., San Francisco 7, Calif.

Nadler, Robert A. (J'42) (NMCDJL), V.P., Charge of Engrg. & Sales, Nadler Fdy. & Mch. Co., Inc., Plaquemine, La.

Naeseth, Rodger Leonard (J'41) (A), Capt., Aero. Research, Langley Memorial Aero. Lab., Natl. Advis. Com. for Aero., Langley Field; *for mail*, 209 Algonquin Rd., Hampton, Va.

Nagai, Gladyes Masao (J'41) (ABM), Draftsman, Houston Fdy. & Mch. Co., 2005 White St., Houston; *for mail*, Box 26, Almeda, Tex.

Nagel, Eric W. ('38) (CDK), Factory Mgr., Wrought Iron Range Co., 5661 Natural Bridge Ave., St. Louis 20, Mo.

Nagel, Theo. ('18) (FKS), Cons. Engr., 76 Remsen St., Brooklyn, N.Y.

Nagle, March Emmett ('42) (BCE), Plant Mgr., Kimble Glass Co., 12th & Arnold Sts., Chicago Heights, Ill.

Nagle, Perry Ira ('44) (BHL), Mgr., Indus. Pumps, Am. Manganese Steel Div., Am. Brake Shoe Co.; *for mail*, 258 W. 16th Pl., Chicago Heights, Ill.

Nagler, Forrest ('18) (AHSK), *Life Member for Distinguished Service*, '21; Ch. Mech. Engr., Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1, Wis.

Nagler, Larry H. ('45) (MCFYB), Tech. Editor, *Motor Magazine*, 572 Madison Ave., New York 22, N.Y.; *for mail*, 16628 Pinehurst, Detroit 21, Mich.

Naiman, Robert (J'42) (JMNYZ), Electronics Technician's Mate, 2nd Class, U.S.N.R., U.S.S. *Portsmouth*, CL-102; *for mail*, 8 Haley St., Roxbury 19, Mass.

Nairne, Clayton Ludlow ('43) (CKSFAY), Asst. to Exec. V.P., New Orleans Pub. Serv., Inc., 317 Baronne St., New Orleans 12, La.

Nalbandian, Haig Sarkis (J'44) (AHBS), Asst. Engr., Cent. Tech. Dept., Shipbldg. Div., Bethlehem Steel Co., 97 E. Howard St., Quincy 69; *for mail*, 86 S. Crescent Circuit, Brighton 35, Mass.

Näll, Robert Lauren ('43) (LDM), Plant Engr., Flintkote Co., 4500 N. Galvez St., New Orleans 4; *for mail*, 1139 State St., New Orleans 18, La.

Nalven, Robert M. (J'40) (KSHL), Mech. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 8267 Austin St., Kew Gardens, N.Y.

Nankivell, John Elbert (J'43) (EHS), 201 Hackensack St., East Rutherford, N.J.

Nantz, Merle Edward ('43) (HBNJYL), Hydroelec. Engr., Charge Hydroelec. Design, U.S. Engr. Office, 1709 Jackson St.; *for mail*, Box 869, Omaha, Neb.

Napier, Alex E. (J'28), Bldg. Secy., Pur. & Maint., Y.M.C.A., 167 Sands St., Brooklyn, N.Y.

Napier, David Jamieson ('45) (FKS), Canadian Rep. & Supvg. Inspr., Mutual Boiler Ins. Co. of Boston, 607 Sterling Tower, 372 Bay St., Toronto; *for mail*, 85 Esgore Dr., Toronto 12, Ont., Can.

Napier, E. T. (J'37) (BKS), Lt., U.S.N., Naval Boiler & Turbine Lab., Naval Shipyard, Philadelphia, Pa.

Napp, Anthony Edward (J'43) (BAHJS), Product & Design Engr., Arma Corp., 254—36th St., Brooklyn; *for mail*, 101 Nassau Ave., Brooklyn 22, N.Y.

Nardelli, Dominick (J'45), 576 E. 181st St., Bronx 57, N.Y.
Nardone, Pio (J'43) (SHME), Mech. Engr., Research Dept., Worthington Pump & Mchy. Corp., Worthington Ave., Harrison, N.J.; *for mail*, 1578—85th St., Brooklyn 28, N.Y.
Nardone, Romeo Morton ('34; '35) (ABMNHY), Ch. Engr., Victory Engrg. Corp., 744 Broad St., Newark 2; *for mail*, 744 River Rd., Teaneck, N.J.
Narmore, Phil Blasier ('44) (BJYCAN), Exec. Dean, Ga. Sch. of Tech., Atlanta, Ga.
Nascenzi, Frank Lawrence (J'42) (MCYJNB), Mfg. Engr., Westinghouse Elec. Corp., Bloomfield; *for mail*, 50 Morton Pl., East Orange, N.J.
Nash, Charles Whitwell (J'42) (ACHB), Pilot Officer, Royal Canadian Air Force; 1940 Greatford Pl., Victoria, B.C., Can.
Nash, Douglas E. ('17; '35), Treas., Nash Engrg. Co., Wilson Ave., South Norwalk, Conn.
Nash, Richard Leland (J'34) (CBENRS), Asst. Ch. Engr., Chicago Pneumatic Tool Co.; *for mail*, 1043 Elk St., Franklin, Pa.
Nasmyth, P. H. (J'46), 129 Adolphus St., Cornwall, Ont., Can.
Nass, Lawrence (J'41) (NJBAGY), Mech. Designer, Mch. & Tool Design Co. of N.Y., 200 Madison Ave., New York; *for mail*, 1670 E. 24th St., Brooklyn, N.Y.
Nass, Walter Robert (J'40) (BNZJH), Major, U.S.A.; *for mail*, 2033 W. 78th Pl., Los Angeles, Calif.
Nathan, Henry H. (J'34) (BSK), Assoc. Mar. Engr., Naval Shipyard, Charleston; *for mail*, 20 Glebe, Charleston 6, S.C.
Nathan, Walter (J'44) (MNC), Electronics Technician's Mate, 2nd Class, U.S.N.R., Radio Instr., Naval Trng. Sta., Navy Pier, 4th Phase, Chicago; *for mail*, 740 Bittersweet Pl., Chicago 13, Ill.
Nathanson, Max ('45), Propr., Mgr., Canadian Armature Works, 6595 St. Urbain St.; *for mail*, 5457 Victoria Ave., Montreal, Que., Can.
Natkins, Ephraim Nathaniel (J'44) (BHJ), Gries Reproducer Corp., 780 E. 133rd St., New York; *for mail*, 939 Woodycrest Ave., Bronx, N.Y.
Nau, Henry A. ('40) (MNORTY), 90 West St., New York 6, N.Y.
Nau, Paul Russell (J'38) (EFZ), Asst. Exper. Engr., Wis. Motor Corp., 1910 S. 53rd St.; *for mail*, 2164 S. 60th St., West Allis 14, Wis.
Naughton, Frank U., Jr. (J'45) (BCT), Sales Engr., Hyatt Bearings Div., Gen. Motor Sales Corp., Harrison, N.J.; *for mail*, 983 Pleasant St., Worcester, Mass.
Naughton, John Francis (J'45) (ABJ), Asst. Metallurgist, Wright Aero. Corp., Paterson; *for mail*, 292 Edgewood Ave., West Englewood, N.J.
Naugle, John J. ('24), 120 Wall St., New York, N.Y.
Naujoks, Waldemar ('44) (JCANML), Ch. Engr., Steel Improvement & Forge Co., 970 E. 64th St., Cleveland 14; *for mail*, 3804 Severn Rd., Cleveland Heights 18, Ohio.
Naulin, Donald B. (J'43) (ECMSFA), Jr. Mech. Engr., Diesel Div., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7; *for mail*, 4787 N. Idlewild Ave., Milwaukee 11, Wis.
Naumburg, Robert Elkan ('17; '24) (TLM), Mech. Engr., Jonas & Naumburg Corp., 516 W. 35th St., New York 1, N.Y.
Naylor, Franklin Llewellyn, Jr. (J'31) (CAMRJN), Co-ordinator, Charge of Interchangeability, Lockheed Aircraft Corp. & Instr., Burbank Unified Sch. Dist., Burbank; *for mail*, 1324 N. Pacific Ave., Glendale 2, Calif.
Naylor, George M. ('20) (CMDWSO), Pres., Gen. Mgr., Fairbanks Co., 393 Lafayette St., New York 8, N.Y.; *for mail*, 88 Warren Pl., Montclair, N.J.
Naylor, Henry Alan, Jr. ('41; '45) (OSHKED), Mech. Engr., Whitman, Requardt & Associates, 1304 St. Paul St., Baltimore 2; *for mail*, 5104 Wetheredville Rd., Dickeyville, Baltimore 7, Md.
Nazzaro, Armand Louis (J'43) (TDLNZO), Lt. (j.g.), U.S.N.R., Exec. Officer, LSM-555, Fla. Gp., 16th Fleet, c/o Postmaster, Green Cove Springs, Fla.; *for mail*, 150 Greenway Terrace, Forest Hills, N.Y.
Nazzaro, Rocco Michael ('45) (CDKSTZ), Owner, Am. Textile Processing Co., 18 Market St., Paterson 1, N.J.
Nazzer, Don Barkley (J'41) (NMB), Asst. Research Engr., Natl. Research Council, Montreal Rd., Ottawa, Ont., Can.
Neal, George Alfred ('21; '35) (CES), Pres., Sioux City Gas & Elec. Co., 515—5th Ave., Sioux City, Iowa.
Neal, Henry Percy (J'36) (NS), Prof., Engrg. Drawing, Miss. State College, Box 74, State College, Miss.
Neal, Kenneth Gerow (A'43) (CMD), Partner, Charge of Cons. Servs., Neal, Sawyer & Co., 299 Madison Ave., New York 17, N.Y.
Neal, Robert Abbott ('45) (SHRACO), V.P., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30, Pa.
Neal, Robert S. ('22; 35) (EHY), Sales Engr., Tex. Co., 332 S. Michigan Ave., Chicago 4, Ill.; *for mail*, 5412 Cabanne Ave., St. Louis 12, Mo.
Neal, Stanford (J'36) (SKE), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 21 N. Ferry St., Schenectady 5, N.Y.
Neal, William (J'44) (OCFSMJ), Jr. Engr., Trainee, Creole Petroleum Corp., Apartado 172, Maracaibo, Venezuela, S.A.
Neale, Douglas Frederick (J'45), Ensign, U.S.N.R.; 1102 Oakmont Rd., Charleston, W.Va.
Neale, John A. ('25) (CBHLAY), Ch. Engr., Underwriters Labs. Inc., 207 E. Ohio St., Chicago 11, Ill.
Near, Lloyd Burton ('38) (DSBA), Div. Mgr., Aircraft Div., Los Angeles By-Products Co., 639 S. San Pedro St., Los Angeles 14; *for mail* 4416 New York Ave., La Crescenta, Calif.
Nearing, Dudley W. ('32), Sales Engr., New Departure Div., Gen. Motors Corp., Bristol, Conn.; *for mail*, 3457 Manor Hill Dr., Cincinnati, Ohio.
Neave, Pierson M. ('03; '26) (CKT), Secy., N.Y. Electrical Society, Inc., 29 W. 39th St., New York 18, N.Y.; *for mail*, 46 Highland Ave., Glen Ridge, N.J.
Nebelsiek, Hilbert John (J'43) (BNWEHK), Engrg. Inspr., Stand. Oil Co. of Ind., 119th & Front Sts., Whiting, Ind.; *for mail*, 723 E. 90th Pl., Chicago 19, Ill.
Nebesar, Robert J. ('28; '35), Ch. Engr., Bristol Aircraft Div., Universal Moulded Products Corp.; *for mail*, 725 Arlington Ave., Bristol, Va.
Neblett, Robert Stratton ('39) (SEC), Mgr., Fed. & Mar. Div., Gen. Elec. Co., 17 River Rd., Schenectady 5, N.Y.
Nechine, Leonard Myron ('44) (HMNCBJ), Mgr., Prod. & Engrg., Chicago Pump Co., 2336 Wolfram St., Chicago 18, Ill.
Nederman, Malte R. ('25; '31; '35) (CNMJ), Ch. Draftsman, Admin. Engr., Am. Mch. & Fdy. Co., 5502—2nd Ave., Brooklyn 20, N.Y.
Nee, Raymond Moore ('41) (SFCEKD), Power Engr., Am. Cyanamid Co., 30 Rockefeller Plaza, New York 20, N.Y.
Needham, Howard Henry ('21; '35) (JKL), Product Mgr., Vessel Div., A. O. Smith Corp., Milwaukee; *for mail*, 1806 N. 74th St., Wauwatosa 13, Wis.
Needham, Paul Emerson ('42) (AJR), Gen. Engr., N.Y. Air Brake Co., Starbuck Ave., Watertown, N.Y.
Needham, Robert James ('20) (RBDFSK), Mech. & Elec. Engr., Canadian Natl. Rys., 1 Simcoe St., Toronto; *for mail*, 73 Rivercrest Rd., Toronto 9, Ont., Can.
Needs, Sydney J. ('26; '33), Serv. Mgr., Kingsbury Mch. Works, Inc., 4324 Tackawanna St.; *for mail*, The Fairfax, 43rd & Locust Sts., Philadelphia, Pa.
Needy, John Alfred ('19; '35) (BEFKNS), Dir. of Technology & Engrg. Education, Evansville College, 1600-1900 Lincoln Ave., Evansville 4; *for mail*, 2311 E. Mulberry St., Evansville 14, Ind.
Neely, Frank H. ('08; '14) (CTYMDG), Exec. V.P., Secy., Rich's, Inc., 45 Broad St., S.W., Atlanta 2, Ga.
Neely, William Jack ('28) (C), Plant Engr., W. T. Rawleigh Co., 101 Liberty; *for mail*, 1426 W. Lincoln Blvd., Freeport, Ill.
Neff, John P. ('13) (RA), Retired; 106 S. Harrison St., East Orange, N. J.
Neff, John Wallace ('44), Ch. Engr., Binney & Smith Co., Bushkill Dr., Easton, Pa.
Neff, Robert Leslie (J'38) (CWJ), Mech. Engr., Brunswick-Balke-Collender Co., 1700 Messler St., Muskegon; *for mail*, 1142 W. Summit Ave., Muskegon Heights, Mich.
Negrin, Miguel (J'45) (BCAJSN), Pvt., 80th QM. Trng. Co., 14th QM. Trng. Bn., Camp Lee, Va.
Negulici, Charles (J'43) (AKM), 4422 W. 66th St., Cleveland 9, Ohio.
Neidel, Edward Charles, Jr. (J'44), Radio Technician, 3rd Class, Class 79B, Bks. 17, Radio Materiel Sch., Treasure Island, Calif.
Neifert, Harry Ralph ('43) (BRYJM), Research Engr., Supvr. R.R. Research Dept., Timken Roller Bearing Co., 1835 Deuber Ave., S.W., Canton 6, Ohio.
Neil, Edmund B. ('15; '20; '28) (EAMJNO), Cons. Engr., 547 E. Broad St., Columbus 15, Ohio.
Neill, Frank Henry (J'43) (BHJ), Tenn. Eastman Corp.; *for mail*, 139 Henley Rd., Oak Ridge, Tenn.
Neill, William A. ('18) (CMDNBS), Retired; 1273 North Ave., New Rochelle, N.Y.
Neilon, Claude Russell ('45) (BNMCJY), Mech. Engr., Houston Plant, Natl. Supply Co., Box 2616, Houston, Tex.
Neilson, Edward J. (J'45) (DL), Neilson Bros., 232-238—39th St., Brooklyn, N.Y.
Neilson, George Edward (J'42) (SKFED), Pvt., 20th Co., Separation Center, Ft. Dix, N.J.; *for mail*, 4714 McPherson St., St. Louis 8, Mo.
Neiman, Charles Herman, Jr. ('43; '46) (KFBNSZ), Asst. Ch. Engr., York-Shipley, Inc., Jessop Place & Pa. R.R.; *for mail*, 330 W. Jackson St., York, Pa.
Neithercut, Donald Charles (J'45) (CJM), Mgr., Am. Steel & Iron Works, Denver; *for mail*, 3600 Garfield, Denver 5, Colo.
Neiva, Rubens V. (J'36) (BKS), Lt. Comdr., Naval Shipyard, Philadelphia, Pa.
Nelden, Richard Marine (J'36) (HNSEAB), Research Engrg. Supvr., Hyd. Coupling Sec., Am. Blower Corp., Box 58, Roosevelt Park Annex, Detroit 32, Mich.
Nelden, William A. (J'37) (AKDLM), Branch Mgr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., 918 Spitzer Bldg., Toledo 4, Ohio.
Nelgner, Harry DeWitt (J'43), 6709 Leona, St. Louis, Mo.
Nelidov, Ivan Manuel ('44), Sr. Engr., U.S. Engr. Dept., 210 P.O. Bldg., Sacramento; *for mail*, 1109 Contra Costa Dr., El Cerrito, Calif.
Nelis, Joseph J. ('14), V.P., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Nelles, Maurice ('44) (CLAWJB), Asst. to Research Dir., Lockheed Aircraft Corp., Burbank; *for mail*, 3713 Laurel Canyon, North Hollywood, Calif.
Nelligan, Thomas Paul ('46) (CHO), Partner, Thomas P. Nelligan & Associates, 53 W. Jackson Blvd., Chicago 4; *for mail*, 631 Blackthorn Rd., Winnetka, Ill.
Nelson, Albert Walter (J'38) (BAHNMJ), Lt., U.S.N.R., Research & Devel., Launcher Sec., Naval Ord. Test Sta., Inyokern, Calif.
Nelson, Arthur L. ('28), Sr. Partner, Arthur L. Nelson, Engrs., 31 St. James Ave., Boston, Mass.
Nelson, Bernard Stanley ('10; '17; '22) (HESKBW), Ch. Engr., A. M. Lockett & Co. Ltd., 808 Whitney Bank Bldg., New Orleans, La.
Nelson, Carl Willard (J'37) (BRJZSE), Draftsman, David S. Castle Co., 1082½ N. 1st St., Abilene; *for mail*, 2101 Layton Ave., Ft. Worth, Tex.
Nelson, Christian B. ('39), Mfr., C. B. Nelson & Co., 727 S. Dearborn St.; *for mail*, 6223 Lundy Ave., Edgebrook, Chicago, Ill.
Nelson, David Bruce (J'41) (SAJNKH), Mech. Engr., Design, Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 218 Union St., Schenectady 5, N.Y.
Nelson, Delmar W. ('23; '30) (EKS), Assoc. Prof. Mech. Engrg., College of Engrg., Mech. Engrg. Bldg., Univ. of Wis., Madison 6, Wis.
Nelson, Donald Arthur (J'41) (SBFK), 215 N. 32nd St., Lincoln 3, Neb.
Nelson, Edward John (J'42) (JMLNCB), Clerk, Prod. Office, Am. Mch. & Fdy. Co., 5502—2nd Ave., Brooklyn; *for mail*, 151 E. 81st St., New York 28, N.Y.
Nelson, Elmer Charles (J'40) (MDLER), Tool Designer, Buda Co., 154th St. & Commercial Ave., Harvey; *for mail*, 2524 W. Vermont St., Blue Island, Ill.
Nelson, Emil Andrew ('13) (BAHM), Owner, Nelson Engrg., Ardmore Park, St. Clair Shores, Mich.
Nelson, Eric H. ('09) (CDKM), V.P., Charge of Mfg., Griscom-Russell Co., Massillon, Ohio.
Nelson, Eric William ('38; '43) (MCNBJ), Product Engr., West. Elec. Co., Inc., 395 Hudson St., New York 14, N.Y.; *for mail*, 356 Millburn Ave., Millburn, N.J.
Nelson, Francis Perin (J'44) (KS), Mech. Designer, Austin Co., 19 Rector St., New York; *for mail*, 364 Lincoln Pl., Brooklyn 17, N.Y.
Nelson, Glenn Wilford (J'39) (FMR), Asst. Terminal Foreman, Gardenville Eng. House, N.Y. Cent. R.R., Gardenville; *for mail*, 18 Janet St., Buffalo 15, N.Y.
Nelson, Harry L. N. (J'43) (MLCJOB), Jr. Product Engr., Eversharp Inc., 1800 W. Roscoe St.; *for mail*, 1322 Cornelia Ave., Chicago 13, Ill.
Nelson, Henry Drexel (J'46) (CKM), 10-12—154th St., Beechhurst, L.I., N.Y.
Nelson, James Foster (J'46), Downs Crane & Hoist Co., 540 W. Vernon Ave., Los Angeles 37; *for mail*, 1049 Elden Ave., Los Angeles 6, Calif.
Nelson, James Torrance (J'45) (BESNJH), Jr. Engr., Vancouver Iron Works, Ltd., 1155 W. 6th, Vancouver; *for mail*, 1529 Chesterfield Ave., North Vancouver, B.C., Can.
Nelson, Kenneth Albert (J'46), Norris Stamping Co., Los Angeles; *for mail*, Box 200, Route 1, Garden Grove, Calif.
Nelson, Kenneth Edward (J'45), U.S. Navy, Stevens Inst. of Tech., Hoboken, N.J.
Nelson, Laurence Kitchener ('44) (SEOHKL), Assoc. Engr., with James M. Todd, 217 N. Peters St., New Orleans 16; *for mail*, 2502 Palmer Ave., New Orleans 15, La.
Nelson, Lloyd Stanley (J'37) (EBYHFN), Capt., Proof Officer, Combat Vehicle Branch, Automotive Div., Ord. Research & Devel. Center, Aberdeen Proving Ground, Md.
Nelson, Lorraine Russell ('43) (SFCKD), Supt., Power Plant, Keokuk Electro Metals Co.; *for mail*, 206 Concert St., Keokuk, Iowa.
Nelson, Manley R. (J'25), Gen. Supt., Gen. Iron Works Co., 600 W. Bates St.; *for mail*, 911 S. Columbine, Denver, Colo.
Nelson, Morgan (J'41) (ACMY), Maj., Army Air Forces, Engrg. Officer, 43rd Air Depot Gp., A.P.O. 149, New York, N.Y.; *for mail*, Box 140, Route 2, Roswell, New Mex.
Nelson, Norman Travers (J'41) (CJML), Lt. (j.g.), U.S.N.R., Staff Communications Officer, LST Gp. 107, F.P.O., San Francisco, Calif.; *for mail*, 27 Murray Hill Rd., Baltimore 12, Md.
Nelson, Olaf (J'38) (BSD), 1st Lt., 456th Engr. Depot Co., 2nd Platoon, A.P.O. 709, San Francisco, Calif.; *for mail*, 3422 S.E. 16th Ave., Portland 2, Ore.
Nelson, Paul Howard (J'40) (MNYFCJ), 18 Rahway Rd., Millburn, N.J.
Nelson, Prentiss Carl (J'43) (EKFS), Asst. Prof., Engrg. Dept., Univ. of Calif., Berkeley 4, Calif.
Nelson, Richard Lewis (J'39), 712 W. Portland St., Phoenix, Ariz.
Nelson, Robert Lee (J'44), Lt. (j.g.), U.S.N.R.; *for mail*, 426 E. 107th St., Los Angeles 3, Calif.

Nelson, Seddon Cabell (J'34) (BGATYC), Tech. Asst. to Mfg. Mgr., Am. Viscose Corp., Front Royal, Va.
Nelson, Stanley Clifford (J'39) (SFC), Mech. Engr., Plant Engrg. Dept., Hawthorne Sta., West. Elec. Co., Inc., Chicago; *for mail,* 836 N. Massasoit Ave., Chicago 51, Ill.
Nelson, Waldemar Stanley (J'36) (OESALW), Partner, Engr., V. J. Bedell Co., 840 Union St., New Orleans 12; *for mail,* 322 Hillary St., New Orleans 18, La.
Nelson, Warren Roy (J'41) (SOKLFB), Dist. Steam Engr., Westinghouse Elec. Corp., 20 N. Wacker Dr., Chicago 6; *for mail,* 2009 Warner Ave., Chicago 18, Ill.
Nelson, William Andrew ('44), 37-28—85th St., Jackson Heights, N.Y.
Nelson, William Richard (J'46), 154-28—11th Ave., Beechhurst, N.Y.
Nenninger, L. F. ('16; '25; '31) (CNMJHB), Works Mgr., Ch. Engr., Cincinnati Milling Mch. Co., Oakley, Cincinnati 9; *for mail,* 3592 Raymar Dr., Cincinnati 8, Ohio.
Neperud, Wilson F. (J'40) (FKS), Box 1, Guild, Tenn.
Nerad, Anthony J. ('30; '35), Research Engr., Gen. Elec. Co., Schenectady; *for mail,* Alplaus Ave., Alplaus, N.Y.
Nesbitt, Hugh ('34) (BLMZ), Supt., Engrg. Dept., N.Y. Office, Mass. Bonding & Ins. Co., 130 William St., New York 7, N.Y.
Nesbitt, James Wilbur ('42), Retired; Box 73, Mullet Lake, Mich.
Nesgoda, Leonard A. (J'41) (ACS), 1151 Lake Ave., Detroit Lakes, Minn.
Ness, Irving Stanley (J'43) (YTMC), Asst. Engr., Chicopee Mfg. Co., Milltown; *for mail,* 50 University Pl., Princeton, N.J.
Nesson, Israel (J'42), 12 Nichols St., Lynn, Mass.
Nestler, Donald Eugene (J'45) (BJNSKH), Student Test Engr., Philadelphia Works, Gen. Elec. Co., 6901 Elmwood St., Philadelphia; *for mail,* 3703 Chestnut St., Philadelphia 4, Pa.
Nestler, Peter Joseph ('16; '21) (SFHJEA), V.P., Sales, Vincent-Gilson Engrg. Co., 30 Church St., New York 7, N.Y.; *for mail,* 146 Monte Vista Ave., Ridgewood, N.J.
Netherwood, Joseph Stanley ('14; '21; '30) (REJKDF), Supt., M.P. & Equip., Tex. & New Orleans R.R. Co., P.O. Box 219, Houston 1, Tex.
Nettleton, George H., III ('38) (ACK), Yale Club, 50 Vanderbilt Ave., New York, N.Y.
Neubauer, Emil T. P. ('37; '40) (BNJKC), Mech. Engr., Hawley Plant, Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Neubert, Ralph L. (J'43) (ACLOSB), Ensign, U.S.N.R., 25 Austin Pl., Kirkwood 22, Mo.
Neugroschl, Seth Gerson (J'41) (CLTWY), Partner, Internatl. Indus. Associates, 225 Lafayette St., New York; *for mail,* 230 Riverside Dr., New York 25, N.Y.
Neuhaus, Fritz ('98; '06), 33 Pariserstrasse, Berlin, W. 15, Germany.
Neuhaus, Ralph ('22; '37) (CM), Plant Mgr., Aircraft Strut Div., Hughes Tool Co., Box 9188, Central Park Sta., Houston 11, Tex.
Neuhoff, Justin (J'37) (KCBL), Engr., Charge of Air Conditioning, Worthington Pump & Mchy. Corp., Holyoke; *for mail,* 55 Northwood Ave., West Springfield, Mass.
Neumann, Arthur (J'34) (ZNJBG), Ord. Engr., Field Serv., Ord. Dept., Raritan Arsenal, Metuchen; *for mail,* Apt. 3-B, 122 Harper St., Highland Park, N.J.
Neumann, Ernest Paul (J'46) (AEK), Asst. Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge; *for mail,* 30 Louise Rd., Belmont, Mass.
Neumunz, Martin ('27) (HDS), Mech. Engr., Mfrs. Rep., Rm. 811, 90 West St., New York 4, N.Y.
Neunes, Rolf William (J'43), Ensign, U.S.N.R., Exec. Officer, LSM-453, F.P.O., San Francisco Calif.; *for mail,* 146 Melrose St., Rochester 11, N.Y.
Nevard, Maurice Sydney ('44) (CLY), Mech. Engr., Micanite & Insulators Co. Ltd.; *for mail,* 15, The Meadway, Roebuck Lane, Buckhurst Hill, Essex, England.
Nevell, Thomas George William (A'44) (CMYNG), Partner, Charge of Design & Engrg., Cushing & Nevell, 101 Park Ave., New York 17, N.Y.; *for mail,* Box 167-A, R.D. 1, Field Rd., Riverside, Conn.
Nevells, Irving Leslie ('37) (FS), Equip. Engr., Stoker Dept., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* Moylan, Pa.
Nevill, Gale E. (J'28) (BJMH), Southwest Co., 3106 Navigation Blvd., Houston 11; *for mail,* 2224 Herman Dr., Houston 4, Tex.
Nevins, Ennis ('25) Mech. Engr., El Paso Water Works, El Paso, Tex.
New, Winston Randolph ('41) (AESBK), Mgr., Aviation Gas Turbine Lab., Westinghouse Elec. Corp., Lester P.O., Philadelphia, Pa.
Newbegin, John Lander ('43; '45) (TNZLHD), Devel. Engr., Research Div., Chicopee Mfg. Corp., Chicopee Falls, Mass.
Newberg, Eric George, Jr. (J'41) (SFMZBK), Lt. Comdr., U.S.N.R., Naval Ord. Plant, Forest Park, Ill.
Newbern, David Pelle (J'42) (BHDK), Mech. Engr., E. I. du Pont de Nemours & Co., Box 993, Charleston, W.Va.
Newbold, James S. (J'40), Jr. Engr., Va. Pub. Serv. Co., Charlottesville, Va.
Newbury, Frank Davies ('44) (CM), V.P., Dir., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30, Pa.
Newby, William Murray (J'45) (LSBC), H. G. Acres & Co., 2135 Culp St., Niagara Falls, Ont., Can.
Newcomb, Edw. C. ('13) (ABE), Retired; Country Way, North Scituate, Mass.
Newcomb, Franklin LeRoy ('14; '21; '27) (JLY), Sr. Specialist, Stand. Oil Devel. Co., Box 37, Elizabeth, N.J.
Newcomb, Robert E. ('07; '16; '18) (MLCJHB), Cons. Engr., Walsh Holyoke Steam Boiler Works, 110 Appleton St.; *for mail,* 80 N. Pleasant St., Holyoke, Mass.
Newcomb, Robert Scott ('09; '25) (OFHKS), Partner, Newcomb & Boyd, 615 Trust Co. of Ga. Bldg., Atlanta 3, Ga.
Newcomb, Wallace Kempfer ('39; '45) (BNEK), Mech. Engr., Ingersoll-Rand Co.; *for mail,* 140 Hamilton Circle, Painted Post, N.Y.
Newell, Harry B. ('20), Exec. V.P., Ohio Forge & Mch. Corp., 3010 Woodhill Rd., Cleveland, Ohio.
Newell, John B. (J'43), 1746 E. Beardsley Ave., Elkhart, Ind.
Newell, Randall L. ('45), 106 Riley St., Charlestown, Ind.
Newell, Sidney Wynne (J'38) (BCEMN), Asst. Gen. Mgr., Union Diesel Eng. Co., 2200 E. 7th St., Oakland 6, Calif.
Newell, Thomas A. (J'34) (CDL), Seaford, Del.
Newell, W. L. ('14), Lowman Bldg., Seattle, Wash.
Newell, William Stark ('44), Pres., Bath Iron Works Corp., Bath, Me.
Newhall, Donald Harwood (J'37) (ZHLMBJ), Research Engr., Foxboro Co., Foxboro; *for mail,* 129 Common St., Walpole, Mass.
Newhouse, Charles ('42) (BS), Sales Engr., Coon-De Visser Co., 2051 W. Lafayette Blvd.; *for mail,* 14053 Northlawn, Detroit, Mich.
Newhouse, Ray C. ('20) (BFKOMY), Cons. Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 7006 Milwaukee Ave., Milwaukee 13, Wis.
Newkirk, Burt Leroy ('26) (AB), Prof. Vibration Theory & Practice, Rensselaer Poly. Inst., 8th St., Troy, N.Y.
Newlove, Benjamin Carter (J'42) (BHM), 391 Del Rey, Pasadena 8, Calif.
Newman, Albert Augustus (J'44) (MNBZ), Aviation Electronics Technician, U.S.N.R., U.S. Naval Hospital, Great Lakes, Ill.; *for mail,* 168 Elm Ave., Hackensack, N.J.
Newman, Ernest George (J'42) (BSJMHA), U.S. Navy; c/o Fred V. Gordon, 56 W. 76th St., New York, N.Y.
Newman, Louis Benjamin ('44) (KCHZYG), Comdr., Medical Corps, U.S.N.R., Head, Dept. of Physical Medicine, Naval Hospital, Seattle 4, Wash.; *for mail,* 5501 Washington Blvd., Chicago 44, Ill.
Newman, Samuel Frantz, ('19) (BCHJMN), Pres., Landis Tool Co., Ringgold & 6th Sts., Waynesboro, Pa.
Newman, William Arthur ('46), Mgr., Research Dept., Canadian Pacific Ry., 803 Windsor Sta.; *for mail,* 4049 Gage Rd., Montreal, Que., Can.
Newport, Victor George ('23; '36) (BN), Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 85-80—67th Ave., Forest Hills, N.Y.
Newstrom, Carl Leonard ('43; '46) (BHWYKN), Sr. Mech. Engr., Supvr., Test & Trials Unit, Puget Sound Naval Shipyard, Bremerton; *for mail,* Box 337, Harper, Wash.
Newton, Alwin B. (J'31) (ABKCFZ), Ch. Engr., Airtemp Div., Chrysler Corp., 1119 E. Leo St., Dayton 1; *for mail,* 905 E. Schantz Ave., Dayton 9, Ohio.
Newton, C. Arthur ('40) (LCTA), Div. Pur. Agt., Am. Viscose Co., Box 468, Wilmington 99, Del.; *for mail,* 810—15th Ave., Prospect Park, Pa.
Newton, Don L. (J'40) (CDJ), 20107 Libby Rd., Maple Heights, Ohio.
Newton, Evans Kendrick ('30) (LHKOSD), Asst. Ch. Engr., Hooker Electrochem. Co., 47th & Buffalo Ave.; *for mail,* 1837 Niagara Ave., Niagara Falls, N.Y.
Newton, George Henry ('45) (BKS), Advisory Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 4014 Berry Ave., Drexel Hill, Pa.
Newton, John Skillman ('45), Asst. Mgr. Engrg., Steam Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 101 Gladstone Rd., Lansdowne, Pa.
Newton, Raymond Congdon ('44), Pres., Firm of Raymond C. Newton, Babson Park 57, Mass.
Newton, Richard Woodhull (J'44), (ABJMFK), Prod. Engr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* Portion Rd., Lake Ronkonkoma, N.Y.
Newton, Robert E. (J'38) (BAO), *Junior Award,* '40; Asst. Prof. Applied Math., Wash. Univ., St. Louis; *for mail,* 254 S. Brentwood, Clayton 5, Mo.
Newton, Robert George (J'40) (KBZNF), Test Engr., Pittsburgh Testing Lab., 1330 Locust St., Pittsburgh 19; *for mail,* 5230 Westminster Pl., Pittsburgh 6, Pa.
Newton, Ross Buchanan (J'46), Hyde Park, Vt.
Newton, Wm. Geo. ('27), Supt., Peck Bros. & Co., 127 Chestnut St.; *for mail,* 809 Townsend Ave., New Haven, Conn.
Nexsen, Randolph H. ('45) (CEFKS), Ch. Power Engr., Pub. Serv. Comm., State of N.Y., 233 Broadway, New York 7, N.Y.
Nibecker, Karl ('08; '14; '35) (GJC), 8400 Aramingo Ave., Philadelphia 34, Pa.
Nicastro, George Joseph ('25; '36) (CSF), Rep., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Nice, Harold E. (J'44) (JYNKA), Metal Parts Engr., B. F. Goodrich Co., 500 S. Main, Akron; *for mail,* Box 135, R.D. 1, Barberton, Ohio.
Nicholas, Serafim (J'36), 1600 Metropolitan Ave., New York, N.Y.
Nicholls, Malcolm Bancroft ('46) (CLO), 302 Strand Bldg., Cincinnati 2, Ohio.
Nichols, Anthony (J'43) (CLMTS), Project Engr., Johnson & Johnson, George St., New Brunswick; *for mail,* 30 Rose St., Kearny, N.J.
Nichols, Charles Raymond, Jr. ('39) (WMDBL), Dir. of Engrg., Joseph Dixon Crucible Co., 167 Wayne St., Jersey City 3, N.J.
Nichols, Clayton Frank (J'46) (CLW), Mech. Engr. (Fire Protection), Washington Surveying & Rating Bur., Seattle; *for mail,* 34 E. Jones Dr., Bremerton, Wash.
Nichols, David Kierstad (J'42), 622 Prospect St., Nutley 10, N.J.
Nichols, DeOwen, Jr. ('36; '46) (EFS), Research Engr., Research Labs. Div., Gen. Motors Corp., 485 W. Milwaukee Ave., Detroit; *for mail,* 1605 Webster Ave., Birmingham, Mich.
Nichols, Donald Angus (J'41) (RHBJYC), Ensign, U.S.N.R.; 805 N. Audubon Rd., Indianapolis 1, Ind.
Nichols, Edgar Byron ('45) (YZMNLC), Pres., Bio-Plastics Research, Inc. & Nichols Prod. Co., 325 W. Main St.; *for mail,* 19 Colonial Ave., Moorestown, N.J.
Nichols, George Edwin, Jr. (J'44), Engr., Bird Mchy. Co., South Walpole; *for mail,* 35 Dutcher St., Hopedale, Mass.
Nichols, Harold Joseph ('28; '35) (EFSZ), Serv. Engr., Bird-Archer Co. of Calif., 245 Fries Ave., Wilmington; *for mail,* 536 W. 7th St., Downey, Calif.
Nichols, Howard Ralph ('45) (SEKOCF), Engr., Grudem Bros. Co., Inc., 2645 University Ave., St. Paul 4; *for mail,* 5416 Park Pl., Minneapolis 10, Minn.
Nichols, Louis Albert, Jr. ('44) (BETNHK), Mech. Engr., Designer, E. I. du Pont de Nemours & Co.; *for mail,* 1310 Birdsall, Old Hickory, Tenn.
Nichols, Paul Hooper ('42) (SLK), Owner, P. H. Nichols & Co., 903 Whitehead Bldg., Atlanta 3, Ga.
Nichols, Robert Lorain ('45) (ZFHLOS), Instrument Foreman, Magnolia Petroleum Co., Box 3311, Beaumont, Tex.
Nichols, Victor Louis (J'42) (BAJ), Tutor, Sch. of Tech., College of the City of N.Y., 140th St. & Amsterdam Ave., New York; *for mail,* 1326 Riverside Dr., New York 33, N.Y.
Nichols, Wallace Hunter (J'44) (CMYAB), V.P., Ch. Engr., Ring Grip Corp., Cross Highway, Westport, Conn.
Nichols, William Martin ('25; '30) (EFORAJ), Cons. Engr., Am. Loco. Co., Schenectady 5; *for mail,* 1549 Wyoming Ave., Schenectady 8, N.Y.
Nichols, William Wallace ('99) (CHMARS), 116 Edgemont Rd., Scarsdale, N.Y.
Nicholson, Ezra K. ('24; '34) (CLSDKB), Ch. Engr., Charles Lennig & Co., Inc., 5000 Richmond St., Philadelphia 37; *for mail,* 1618 Latimer St., Philadelphia 3, Pa.
Nicholson, James Morton ('40) (RCFJ), Asst. to V.P., Atchison, Topeka & Santa Fe Ry. Co., 80 E. Jackson St., Chicago 4, Ill.
Nicholson, James Robert ('45) (SARHGC), Supvg. Engr., Gen. Elec. Co., 570 Lexington Ave.; *for mail,* Apt. 509, 5 Prospect Pl., New York 17, N.Y.
Nicholson, Lawrence E. (J'41) (KLS), Box 64, Waynesboro, Va.
Nicholson, Richard Frederick, Jr. (J'35) (CH), Engr., Secy., Dir., Nicholson Sprinkler Corp., 111 John St., New York 7, N.Y.
Nicholson, Ross Samuel Lewis (J'45) (ESACLR), Owner, Nicholson Labs., 181 College St., Toronto; *for mail,* 17 Alhambra Ave., Toronto 3, Ont., Can.
Nicholson, Samuel T. ('04), Pres., W. N. Nicholson & Co., Wilkes-Barre, Pa.
Nick, Edwin William ('45), Pres., No. Equip. Co., Erie, Pa.; Bd. Chmn., Copes Regulators, Ltd., London, England; *for mail,* 2020 S. Shore Dr., Erie, Pa.
Nickel, Alfred A. (J'35) (CDMZL), Procedure Supvr., Columbia Steel Co., 235 Montgomery St., San Francisco 6, Calif.
Nickelsen, Charles J. (J'45), U.S.N.R.; *for mail,* 225 Sip Ave., Jersey City, N.J.
Nickerson, Alvano Thomas ('01) Retired; 144-65—87th Ave., Jamaica, N.Y.
Nickerson, John Winslow ('11; '20) (C), Dir. Indus. Relations, Bigelow, Kent, Willard & Co., 75 Federal St., Boston, Mass.; *for mail,* 46 Mountain View Dr., West Hartford 7, Conn.
Nickerson, Robert David (J'44) (BAEL), Lt., Box 75, Hdq., Engr. Sch., Ft. Belvoir, Va.

Nicklas, Donald Louis (J'44) 2nd Lt., Hdq. Co., 3rd Bn., 24th Inf., A.P.O. 331, c/o Postmaster, San Francisco, Calif.; *for mail,* 5 Coffin St., Newburyport, Mass.
Niclaus, William Ludwig (J'43) (Y), Partner, Engr., Wilpet Engrg. & Mfg. Co., 41 King St., Belleville 9, N.J.
Nicol, Geo. A., Jr. ('14; '35) (CLR), Retired; 1127 Lake Way, Palm Beach, Fla.
Nicol, Herbert Erskine ('28) (MNLKBJ), Research Engr., 2417 E. Newton Ave., Milwaukee 11, Wis.
Nicol, Norman C. ('19; '35), Sales Engr., Natl. Tube Co., 71 Broadway, New York, N.Y.
Nicolai, Arduo Lewis ('36; '43) (BEFKS), Mech. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Nicolson, Garth Fancher (J'42) (NHBJ), Ch. Engr., Oil Tool Div., Byron Jackson Co., Box 2017, Terminal Annex, Los Angeles 54, Calif.
Niebanck, Richard John ('23; '28; '35) (CDNMLH), Engrg. & Devel. Coordinator, Natl. Carbide Corp., 60 E. 42nd St., New York 17; *for mail,* 114 Farrington Ave., Philipse Manor, North Tarrytown, N.Y.
Niedhammer, Herbert Albert (J'42) (KBJ), Res. Engr., Charge of Branch Engrg. Office, DeVilbiss Co., 111—8th Ave., New York 11, N.Y.; *for mail,* 680 Linden Ave., Rahway, N.J.
Niekerk, Leo Jacob ('42) (EFKS), Asst. Ch. Engr., Scovill Mfg. Co., 99 Mill St., Waterbury 91; *for mail,* R.F.D. 3, Middlebury Rd., Waterbury 83, Conn.
Nielsen, Donald M. (J'38) (ZLK), V.P., Nielsen & Fryer, Inc., 152 E. Superior St., Chicago 11, Ill.
Nielsen, Hans Kristian ('31) (CLODFB), Mech. Engr., Ch. Draftsman, F. L. Smidth & Co., 11 W. 42nd St., New York 18, N.Y.
Nielsen, Howard L. (J'44), Pres., Ervin A. Rice Co., 2230 S. La Salle St., Chicago 16, Ill.
Nielsen, Jack Norman (J'41) (KBA), Box 46, Star Route, Petaluma, Calif.
Nielsen, M. ('38) (CB), Supt., Mar. Erection Dept., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Nielsen, Svend G. (J'30), 810 Kerper St., Philadelphia, Pa.
Nieman, Howard Clifford (J'43) (COLJD), Lt. (j.g.), U.S.N.R., Asst. Personnel Relations Officer, Naval Am. Depot, Earle, N.J.; *for mail,* 1160 York St., Pittsburg, Calif.
Niemeier, Bernard August (J'44) (NBDZO), Instr. in Mech. Engrg., Rensselaer Poly. Inst.; *for mail,* 46—2nd St., Troy, N.Y.
Niemeyer, Gerald Walter (J'44) (A), Pvt., 62nd QM. Trng. Co., Camp Lee, Va.; *for mail,* 4322 Holly Hills Blvd., St. Louis 16, Mo.
Niemi, Leonard S. ('29; '35) (SFEBK), Engr., U.S. Maritime Comm., Commerce Bldg., Washington; *for mail,* 2309—25th St., S.E., Washington 20, D.C.
Niems, Lee H. (J'41) (CJM), U.S. Navy; 2801 E. 77th St., Chicago, Ill.
Nigh, George W. ('14; '25) (LKEFJS), Asst. Mech. Supt., Tide Water Associated Oil Co.; *for mail,* 930 Hudson Blvd., Bayonne, N.J.
Nihlen, Arvid C. K. (J'40) (CKM), Mfg. Engr., Am. Heat Reclaiming Corp., 1270—6th Ave., New York 20, N.Y.; *for mail,* 908 Irving Rd., Birmingham 9, Ala.
Nikoloff, Subbo ('29), V.P., Leland-Gifford Co., Worcester 1, Mass.; *for mail,* 488 Pomfret St., Putnam, Conn.
Nikonow, John Paul ('19) (B), Pat. Atty., 551—5th Ave., New York 17, N.Y.
Niland, Patrick Barry (J'43) (CMDAJ), Lt. (j.g.), U.S.N.R., Staff Engr., LCI (L) Gp. 108, F.P.O., San Francisco, Calif.
Niles, Alfred S. ('27) (BA), Prof. Aero. Engrg., Stanford Univ., Stanford University, Calif.
Nilges, William C. (J'41), 4209 W. 189th St., Fairview Village, Cleveland, Ohio.
Nilsen, Laurits (J'43), Lt., U.S.N.R., U.S.S. *Puget Sound,* CVE-113, F.P.O., San Francisco, Calif.; *for mail,* Box 63, 79 Dosoris Lane, Glen Cove, N.Y.
Nimbkar, Vishnu (J'36) (B), Mgr., Engr., Consltd. Pneumatic Tool Co., Ltd., P.O. Box 335, Bombay, India.
Niper, Louis S. (J'25) (CGMOW), Firm of Louis S. Niper, Real Estate, Box 103; *for mail,* Ridgewood, Clinton, Conn.
Nitka, William Alfred (J'43) (RC), Mech. Engr., Plant Engrg., Dept. 6002, West. Elec. Co., Inc., Baltimore; *for mail,* 1403 Dundalk Ave., Baltimore 22, Md.
Nitze, Carl Victor (J'46) (BKS), Ensign, U.S.N.R., U.S.S. *Denver,* CL-58, F.P.O., New York, N.Y.; *for mail,* 905 St. George's Rd., Roland Park, Baltimore 10, Md.
Nixon, Boris C. ('46) (LSY), Designer, Dept. 110-X, Goodyear Tire & Rubber Co.; *for mail,* 86½ Broad St., Akron, Ohio.
Nixon, James A. (J'40) (AEJ), 520 W. 3rd St., Eureka, Kan.
Nixon, Stuart ('44) (EAJBCM), Sales Engr., Sealed Power Corp., Muskegon Heights, Mich.
Nixon, William ('30; '42) (KMZDCB), Mech. Engr., Tenn. Eastman Corp., Box 590; *for mail,* Westwood Dr., Knoxville, Tenn.

Noble, Eugene Emmor ('46), Asst. Ch. Engr., Gen. Foods Corp., 250 Park Ave., New York; *for mail,* 40 Algonquin Rd., Tuckahoe, N.Y.
Noble, John Joseph (J'45) (CRABYO), Asst. Test Engr., Am. Steel Fdys., 410 N. Michigan Ave., Chicago 11; *for mail,* 2440 S. Euclid Ave., Berwyn, Ill.
Noble, K. B. ('43) (BJMAZL), Pres., K. B. Noble Co., 100 Pearl St., Hartford, Conn.
Noble, William John, Jr. (J'42) (N), Tool Designer, Stanley Works, 195 Lake St.; *for mail,* 42 Black Rock Ave., New Britain, Conn.
Nobles, Elon John (J'39) (LCOSZE), Engr., Chem. Constr. Corp., 350—5th Ave., New York; *for mail,* 120 Brambach Rd., Scarsdale, N.Y.
Nodelyk, John Cornelius ('45) (BS), Mech. Engr., West. Cartridge Co., East Alton; *for mail,* 3630 Aberdeen, Alton, Ill.
Noe, Robert Clark (J'45), U.S.N.R., 331 E. Sullivan, Gallup, New Mex.
Noell, Milton James (J'35) (EDNSAM), Mgr., Engrg. Dept., Oil Well Supply Co., 2001 N. Lamar St., Dallas 1; *for mail,* 4153 Travis St., Dallas 4, Tex.
Nofsinger, Charles William ('38) (CFKLMO), Cons. (Oil Refining), 36 Bedford Rd., Summit, N.J.
Nohse, William R. E. (J'21) (RSC), Gen. Mech. Inspr., Cent. R.R. of N.J., Elizabethport, N.J.; *for mail,* 630 W. 139th St., New York 31, N.Y.
Nolan, Hubert L. (J'40) (CKS), 2nd Lt., Ord. Dept., 722nd Ord. Co., Aviation (AB), Orlando Air Base, Orlando, Fla.
Nolan, John B. (J'37) (RMCDEJ), Prod. Engr., Am. Loco. Co., Schenectady 5, N.Y.
Noland, Solomon Calhoon ('43) (ACGHLM), So. Mgr., Automatic Sprinkler Corp. of Am., 161 Spring St., N.W., Atlanta, Ga.
Nolin, Charles Albert (J'44) (AJYMBZ), Engr., Lodington-Griswold Inc., Old Saybrook; *for mail,* Westbrook, Conn.
Nolte, Frederic Stuart (J'42) (FKS), 124 Pierrepont St., Brooklyn 2, N.Y.
Nolte, Richard B. (J'41) (ACS), Lt., U.S.N.; 2328 E. 70th St., Chicago, Ill.
Nonemaker, Frank, Jr. (J'42) (JMBN), Plant Engr., Milton Mfg. Co., Lower Market St.; *for mail,* 30—1st St., Milton, Pa.
Nones, Lynn W. ('31) (CSJEF), V.P., Diamond Power Specialty Corp., Suite 1215, 37 W. 43rd St., New York 18, N.Y.
Nonnenbruch, Otto ('22) (ERYH), Sales Engr., Baldwin Loco. Works, 120 Broadway, New York 5; *for mail,* 310 E. 75th St., New York 21, N.Y.
Nooger, Samuel (J'38) (AMBCGE), Ch. Engr., Aero Manuscripts, Inc., 101 Park Ave., New York 17, N.Y.
Noon, Alonzo William (J'42) (BAJ), Mech. Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 2137 Campbell Ave., Schenectady 6, N.Y.
Noonan, Edward Francis, Jr. (J'43) (BESKNO), Partner, Mar. Engr., Tomalin & Noonan, 302 Bond Bldg., Washington 5, D.C.; *for mail,* 25 Darwin Ave., Takoma Park 12, Md.
Noonan, John D. ('35) (JMS), Lt. Comdr., U.S.N.R., Office of Supvr. of Shipbldg., U.S.N., Fore River Plant, Shipbldg. Div., Bethlehem Steel Co., Quincy; *for mail,* 12 Chesterfield Rd., East Milton, Mass.
Noor, Robert A. ('41) (LO), Mech. Engr., Day & Zimmermann, Inc., 15th & Chestnut Sts., Philadelphia 2; *for mail,* 1107 Allston Rd., Brookline, Upper Darby, Pa.
Nopper, Richard Earl (J'39) (CJS), 740 Edgewood St., Baltimore 29, Md.
Norcross, Adrian Moulton (J'46), 10 Colonial Rd., Worcester 2, Mass.
Norcross, Gilbert Floyd (J'46), Design Engr., McGlynn, Hays & Co., 71 Rutgers St., Belleville; *for mail,* 102 Ravine Ave., West Caldwell, N.J.
Nordberg, Bruno Victor Edward ('21) (ESBFH), Exec. Engr., Nordberg Mfg. Co., 3073 S. Chase Ave.; *for mail,* 2940 S. Logan Ave., Milwaukee 7, Wis.
Norden, Carl Lucas (Non-Member), *Holley Medallist,* '44; V.P., Carl L. Norden Inc., 50 Varick St.; *for mail,* Hotel Fairfax, 116 E. 56th St., New York, N.Y.
Norden, Henry Frederic ('20; '24; '30) (CJY), Mar. Engr., Foreign Repairs & Salvage Div., War Shipping Admin., Commerce Bldg., Washington 25, D.C.; *for mail,* 616 Reiss Pl., New York 67, N.Y.
Nordenholt, George Fred ('26) (ABJMRY), Editor, *Product Engineering,* McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.
Nordheimer, Albert ('16; '24; '35) (SERMJO), War Assets Admin., 70 Pine St., New York 5; *for mail,* 75 E. 21st St., Brooklyn 26, N.Y.
Nordin, Obert Layne (J'42) (BJNKEF), Mech. Engr., Thornhill-Craver Co., Box 1184, Houston 1; *for mail,* 7126 Palmetto, Houston 12, Tex.
Nordlie, Frederick Robert (J'41) (NSHEJW), Designing Engr., Gardner Mch. Co.; *for mail,* 318½ Adams St., Beloit, Wis.
Nordstrom, Reginald Fred (J'26) (CM), Prod. Mgr., Ford Instrument Co., 31-10 Thompson Ave., Long Island City 1; *for mail,* 29—1st St., Lynbrook, N.Y.
Nordt, Paul Winfried, Jr. ('35; '45) (CJM), Sr. Mech. Engr., Charge of Mech. Engrg., Allen B. DuMont Labs., Inc., 2 Main Ave., Passaic, N.J.
Norgren, Carl A. ('41) (BCHMN), Gen. Partner, Gen. Mgr., C. A. Norgren Co., 222 Santa Fe Dr., Denver 9, Colo.

Norgren, Duane Uriel (J'45), Ensign, Civil Engr. Corps, U.S.N.R.; R.R. 2, Sequim, Wash.
Norling, Bert S. ('36) (SFHEJK), Power Design Engr., E. I. du Pont de Nemours & Co., Wilmington; *for mail,* 600 S. Bancroft Pkwy., Wilmington 157, Del.
Norman, Benjamin Franklin, Jr. ('37; '41) (SCZFL), 1224 W. Broad St., Freeport, Tex.
Norman, Carl Adolph ('18) (EN), Prof. Mch. Design, Mech. Engrg. Dept., Ohio State Univ., Columbus 10, Ohio.
Norman, Earl Edward ('30) (HSCF), Mgr., City Light & Water Utilities, City Hall, Kalamazoo 9, Mich.
Norman, John Lawrence (J'40) (MNCEBA), Lt. Comdr., U.S.N.R., U.S. Naval Separation Center, Camp Wallace; *for mail,* 2733 Rosedale Ave., Dallas 5, Tex.
Norman, Melvin H. (J'39) (NBJMEH), Design Engr., Ralph N. Brodie Co., 61st & Lowell Sts., Oakland; *for mail,* 4274 Coolidge Ave., Oakland 2, Calif.
Norris, Alfred George ('45) (COMJ), Cons. Research Engr., 2430 Piedmont Ave., Berkeley 4, Calif.
Norris, Charles Brazer ('26) (WBAYZL), Engr. (P-6), U.S. Forest Products Lab., Madison 5, Wis.
Norris, Charles H. ('13) (ERS), Mech. Engr., 43 Ten Acre Rd., New Britain, Conn.
Norris, Clifton Dwight ('24; '35) (EFCD), Supvg. Engr., Refining Dept., Engrg. Div., Tex. Co., 205 E. 42nd St., New York 17, N.Y.; *for mail,* 412 Franklin Ave., Hasbrouck Heights, N.J.
Norris, Earle B. ('15) (BSCELM), Dean of Engrg., Va. Poly. Inst.; *for mail,* Box 26, Blacksburg, Va.
Norris, Edson R. ('06), 542 Braddock Ave., Pittsburgh, Pa.
Norris, Edward W. ('12; '21; '21) (BLS), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Norris, H. Lee, Jr. (J'37) (ALS), Mech. Engr., Dow Chem. Co.; *for mail,* 323 E. Broad St., Freeport, Tex.
Norris, Henry Lee ('04; '21; '21) (EFO), Cons. Engr., 600 W. 116th St., New York 27, N.Y.
Norris, R. Hosmer ('35; '46) (KAB), *Pi Tau Sigma Medallist,* '41; Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1385 Regent St., Schenectady 8, N.Y.
North, John Walton ('37; '41) (KLCWSF), Partner, North Bros., Box 252, Atlanta 1, Ga.
North, Richard A. ('21; '39) (C), Exec. V.P., Republic Industries, Inc., 25 Broad St., New York 4, N.Y.
Northam, Carleton D. (J'24) (SCFK), Asst. Supt. Steam Power, Wis. Steel Div., Internatl. Harvester Co., 2701 E. 106th St., Chicago 17; *for mail,* 2142 W. 110th St., Chicago 43, Ill.
Northrup, Francis Benjamin, Jr. (J'38) (HOSCWY), Engr., Ingersoll-Rand Co., 224 E. 9th St., Cincinnati 2; *for mail,* R.R. 2, Tipp City, Ohio.
Northrup, Francis Benjamin, Sr. ('05; '16) (YNBDA), Designer, Howe Mchy. Co. Inc., 30 Gregory Ave., Passaic; *for mail,* 58 Raymond Ave., Nutley 10, N.J.
Northrup, Milton George ('36; '46) (HSB), Engr., Draftsman, Kelco Co., 530 Broadway, San Diego 1; *for mail,* 5448—3rd Ave., Los Angeles 43, Calif.
Northrup, Robert Wright (J'46) (CEF), Ensign, U.S.N.R.; 81 Virginia Ave., Clifton, N.J.
Norton, Augustus Peter, Jr. (J'44), 3170—17th St., N.W.; Washington, D.C.
Norton, C. Palmer (J'41) (BCK), Jr. Mar. Engr., U.S. Maritime Comm., 348 Baronne St.; *for mail,* 2508½ Peniston St., New Orleans, La.
Norton, Eric Holt (J'43) (SEFBMN), Engr. Officer in Command of Mchy., Royal Canadian Navy (Reserve), H.M.C.S. *Petrolia,* Halifax N.S.; *for mail,* 2049 W. 37th Ave., Vancouver, B.C., Can.
Norton, Frederick Howard (A'44) (RCJ), V.P., Sales, Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.
Norton, Paul Thornley, Jr. ('33) (CWJDML), Prof. Indus. Engrg., Va. Poly. Inst.; *for mail,* Box 183, Blacksburg, Va.
Norton, William Mear, Jr. (J'45) (AC), Staff Asst., Pur. Director's Staff, Consltd. Vultee Aircraft Corp., 3302 Pacific Highway, San Diego 12; *for mail,* 444 Retaheim Way, La Jolla, Calif.
Norvig, Johan ('13) (DFSL), Gen. Supt., Pa. Dixie Cement Corp., Nazareth, Pa.
Norville, Richard G. (J'40) (CDMJLO), Mech. Engr., Reduction Div., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19; *for mail,* 403 Groveland St., Pittsburgh 27, Pa.
Nothmann, Gerhard Adolf (J'42) (BNKG), Radio Technician, 3rd Class, U.S.N.R., Radio Materiel Sch., Naval Trng. Sta., Billet 209-127, Navy Pier, Chicago; *for mail,* 5968 W. Erie St., Chicago 44, Ill.
Nott, Albin J. ('10; '17) (SAHZLE), Engr., Colgate-Palmolive-Peet Co., 1 Newark Ave., Jersey City, N.J.; *for mail,* 16 Strathmore Rd., Great Neck, L.I., N.Y.
Nottage, Herbert B. (J'37) (KBL), Research Assoc., Research Lab., Am. Soc. of Htg. & Vent. Engrs., 7218 Euclid Ave., Cleveland 3; *for mail,* 5 Douglas Rd., Wickliffe, Ohio.
Nottberg, Henry, Jr. (J'38) (CEFKOS), Secy., U.S. Engrg. Co., 914 Campbell St., Kansas City, Mo.

Nottelmann, John F. (J'40) (NBFLCS), Mech. Engr., Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.
Nourse, Chester Linwood ('39) (CKLM), Indus. Engr., H. P. Hood & Sons, Inc., 500 Rutherford Ave., Boston 29, Mass.
Novak, Howard Miroslav (J'44) (KSHFBN), Jr. Engr., Foster Wheeler Corp., Dansville, N.Y.
Noveck, Solomon Jacob (J'42) (AMDEC), Sr. Detailer, Arma Corp., 254—36th St., Brooklyn; *for mail,* 1168—52nd St., Brooklyn 19, N.Y.
Novikoff, Igor A. (J'28), Mech. Engr., Solvay Process Co.; *for mail,* 101 Crescent Ave., Hopewell, Va.
Novitzky, Paul George (J'44) (KHFA), Serv. Engr., Foster Wheeler Corp., 165 Broadway, New York 6; *for mail,* 148-19—12th Ave., Whitestone, N.Y.
Nowacki, Leo Matthew (A'44) (BJKND), Asst. Sec. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 83 Snowden Ave., Schenectady 4, N.Y.
Nowak, Richard Charles (J'45) (NBEAKH), Physicist, Mech. Engr., Naval Research Lab., Washington 25, D.C.; *for mail,* 58 Plymouth St., New Hyde Park, L.I., N.Y.
Noxon, Elmer W. ('39) (LCD), Ch. Engr., Mgr., Engrg. Dept., Ralston Purina Co., St. Louis 2; *for mail,* 415 Yorkshire Dr., Webster Groves 19, Mo.
Noy, Jack M. ('45) (LJDKZS), Elec. Engr., Climax Molybdenum Co. of Pa.; *for mail,* 516 W. 5th Ave., Langeloth, Pa.
Noyes, George Edward ('31) (RBSJZF), Ch. Chemist, So. Pacific Co.; *for mail,* 2775—21st St., Sacramento 14, Calif.
Noyes, Jonathan Alvan ('16; '25) (CEFMNJ), *Vice-President,* '43-'45; Dist. Mgr., Sullivan Div., Joy Mfg. Co., 6522 Hines Blvd., Dallas 9, Tex.
Noyes, Mason Shaw ('33; '35) (SBJFH), Asst. Ch. Engr., Lunkenheimer Co., Box 360, Annex Sta., Cincinnati 14; *for mail,* 3430 Morrison Pl., Cincinnati 20, Ohio.
Noyes, Richard Roe (J'36) (KSWLON), Sales Engr., Canadian Sirocco Co. Ltd., 630 Dorchester St. W., Montreal, Que., Can.
Noyes, Richard W. (J'28) (JCBKMG), Chief, Engrg. Dept., Oneida Ltd., Oneida, N.Y.
Noyes, William ('31) (CNBJYR), Asst. Ch. Designer, Armstrong Cork Co.; *for mail,* 226 E. Orange St., Lancaster, Pa.
Nugent, Corliss Daniel ('29) (ESR), Pres., Ch. Engr., William W. Nugent & Co., 410 N. Hermitage Ave., Chicago 22; *for mail,* 26 Crescent Dr., Glencoe, Ill.
Nugent, Edward Leo ('36) (SKFDNC), Asst. Engr., Mech. Design, Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Nugent, John Berchmans (J'37) (BDM), Mech. Engr., Arthur D. Little, Inc., 30 Memorial Dr., Cambridge 42; *for mail,* 11 Bartlett Pkwy., Winthrop 52, Mass.
Nugey, Anthony L. ('34; '35), 2 Pierpont St., Rahway, N.J.
Nulle, J. Howard (J'36) (CJM), Inspr., Munitions Div., Am. Type Founders, Inc., Elizabeth; *for mail,* 50 Garden Dr., Roselle, N.J.
Nulsen, John C. ('28) (ACJ), Indus. Engrg., 915 Moulton Rd., North Muskegon, Mich.
Nulsen, Marvin E. ('21; '27) (OHAJBY), Mfrs.' Rep., 333 Kenyon Ave., Indianapolis 1, Ind.
Nunnally, Major Perkins ('23; '31) (RCBNS), Mech. Engr., St. Louis, Southwest. Ry. Lines, Pine Bluff, Ark.
Nunne, Frederick Charles ('44) (CDMN), Ch. Engr., Simplicity System Co., Riverside Dr. & Sholar Ave., Chattanooga 6, Tenn.
Nunnelee, Harold Bernard (J'43), Graduate Trng. Course, Allis-Chalmers Mfg. Co., 1126 S. 70th, Milwaukee 1; *for mail,* 1460 S. 108th St., West Allis 14, Wis.
Nunns, Edgar Donald (J'42) (ABM), Stress Engr., Glenn L. Martin-Neb. Co., Omaha; *for mail,* 7932 State St., Ralston, Neb.
Nurnberger, Gunther Rudolf (J'42) (NMCZJ), Mech. Engr., Sylvania Elec. Products Inc., 83—30 Kew Gardens Rd., Kew Gardens, L.I., N.Y.
Nusbaum, Lee ('17), Propr., Pa. Engrg. Co., 1119-21 N. Howard St.; *for mail,* 315 Carpenter Lane, Germantown, Philadelphia, Pa.
Nusim, Melach Jos. ('12) (SBH), Cons. Engr., Roots-Connersville Blower Corp., Connersville, Ind.; *for mail,* 416 Cattell, Easton, Pa.
Nute, Edwin L. ('11; '26) (BCFKSY), Plant Engr., Converse Rubber Co., Pearl St., Malden 48; *for mail,* 12 Wave Ave., Wakefield, Mass.
Nutt, John Gordon (J'37) (LKW), Designer, Potash Co. of Am.; *for mail,* Thayer Apts., Carlsbad, New Mex.
Nutt, Robert E. (J'39), 1st Lt., Ord. Dept., Savanna Ord. Depot, Proving Ground, Ill.
Nutting, Earle McNeil ('37) (BL), 3441 W. 2nd St., Los Angeles 4, Calif.
Nutting, Willard H. (J'38) (BGK), 555 Farlson Sq., Pittsburgh 6, Pa.
Nydegger, Paul Fred ('21) (CDJMTW), Factory Supt., Singer Mfg. Co., Trumball St., Elizabeth 1, N.J.
Nye, Edwin Packard (J'41) (SEA), Instr. in Mech. Engrg., Univ. of N.H., Durham, N.H.; *for mail,* 706 Huntington Ave., Boston, Mass.
Nye, Robert G. ('12) (CHMNSZ), Edinboro, Pa.
Nyffeler, Otto W. (J'31), Locust Ave., Route 3, Peekskill, N.Y.
Nygren, John ('42) (DBNJKL), Mech. Engr., Natl. Products Refining Co., 902 Garfield Ave., Jersey City 5, N.J.; *for mail,* 76 Eltingville Blvd., Staten Island 12, N.Y.
Nyquist, David Frederick (J'42) (CDS), 45 Raddin Grove Ave., Lynn, Mass.
Nystel, Archie C. (J'46), 2935 Flower St., Walnut Park, Huntington Park, Calif.
Nystrom, Karl F. ('21; '25; F'45) (ERS), Ch. Mech. Officer, Chicago, Milwaukee, St. Paul & Pac. R.R. Co., 321 W. Everett St., Milwaukee 3, Wis.
Nystrom, Karl T. ('42; '42) (RC), Asst. V.P., Stand. Ry. Equip. Co., 310 S. Michigan Ave., Chicago 4, Ill.

O

Oakes, Wm. H. ('16), Pres., Buerkel & Co., Inc., 22 Union Park St., Boston, Mass.
Oakey, William Edgar, Jr. (J'45) (NJM), Devel. Engr., Am. Emblem Co., Inc., Box 116, Utica, N.Y.
Oakhill, Frederic Emery ('43) (CMNLDB), V.P., Charge of Mfg., O-Cedar Corp., 2246 W. 49th St., Chicago 9; *for mail,* 711 Linden Ave., Wilmette, Ill.
Oakley, Alfred William ('15; '25) (CDKLZ), V.P., Gen. Mgr., Hudson Refrig. Co., 150—15th St., Jersey City 2, N.J.
Oakley, Harold Clifford (J'41) (SCLJ), Ensign, U.S.N.R., U.S.S. *Nashville,* CL-43, F.P.O., New York, N.Y.; *for mail,* 60 Rhoda Ave., Nutley 10, N.J.
Oakley, Walter Weckerle ('45) (FKDCLM), Ch. Engr., Corning Glass Works; *for mail,* 31 E. 4th St., Corning, N.Y.
Oakley, Warren Briggs, Jr. (J'35) (EFCB), Sr. Project Engr., Detroit Diesel Eng. Div., Gen. Motors Corp., 13400 W. Outer Dr., Detroit 23; *for mail,* 15820 Wisconsin Ave., Detroit 21, Mich.
Oates, Frank Richardson ('41) (DKOS), Engr., Constr. Div., Am. Sugar Refining Co., 120 Wall St., New York 5; *for mail,* 81 Columbia Heights, Brooklyn 2, N.Y.
Oatley, Henry Bigelow ('10; F'45) (RKSF), V.P., Cons. Engr., Superheater Co., 60 E. 42nd St., New York 17, N.Y.
Oatley, Henry Clay ('30; '37) (EFMOS), Ch. Boiler Room Engr., Consltd. Edison Co. of New York, Inc., 666—1st Ave., New York; *for mail,* 241-19—87th Ave., Bellerose 6, L.I., N.Y.
O'Bannon, Lester S. ('22; '35) (LSFEZ), Research Engr., Agric. Exper. Sta., Univ. of Ky., Lexington 29, Ky.
Ober, George C. (J'40) (ABE), Erieside, Ohio.
Ober, Philip LeRoy (J'36) (NCBZ), Engr., Design, Applied Physics Lab., Johns Hopkins Univ., 8621 Georgia Ave., Silver Spring; *for mail,* 16 Cleveland Ave., Takoma Park, Md.
Ober, Theodore M. ('34; '38) (CAHOSM), 805 N.E. Portland Blvd., Portland 11, Ore.
Oberg, David Potter (J'44) (CJMD), Supvr. of Planning, Am. Can Co., 499 Alabama St., San Francisco; *for mail,* 147 Birch St., Redwood City, Calif.
Oberg, Erik ('13; '19) (CJM), *Treasurer,* '25-'35; Editor, *Machinery,* Indus. Press, 148 Lafayette St., New York 13, N.Y.
Oberg, Henry V. ('30; '40) (CDMN), Mgmt. Engr., Robert Heller & Associates, Inc., 1052 Union Commerce Bldg., Cleveland 14; *for mail,* 16630 Kenyon Rd., Shaker Heights 20, Ohio.
Obergfell, Howard Herbert (J'41) (MDLNJC), Jr. Test Engr., Automatic Elec. Co., 1033 W. Van Buren St., Chicago 7; *for mail,* 617 Clinton Pl., River Forest, Ill.
Oberhauser, Louis G. ('38) (S), Plant Supt., Consumers Power Co., Box 126, Essexville, Mich.
Oberholtzer, Robert Everett ('37; '40) (TCSDMN), Ch. Engr., James Lees & Sons Co. & Lees-Cochrane Co. Inc., Bridgeport; *for mail,* 8008 Brierdale Rd., Philadelphia 28, Pa.
Oberhuber, Wm. F. ('22; '35) (BJ), Supt. of Maint., Philadelphia Elec. Co., 27th & Christian St., Philadelphia; *for mail,* 4103 Chestnut St., Philadelphia 4, Pa.
Obermanns, Henry E. (J'30) (GLW), Paper Mill Research, Hammermill Paper Co.; *for mail,* 1938 E. Lakeside Dr., Erie, Pa.
Oberst, Donald Anthony ('34; '44) (NJDOBC), Mech. Engr., Design & Drafting, Libbey-Owens-Ford Glass Co., Toledo; *for mail,* 3708 Revere Dr., Toledo 12, Ohio.
Oberst, Harry E. (J'41) (CJS), Technician, 5th Grade, 13714 Chatauqua Ave., Cleveland, Ohio.
Obert, Casin W. ('04; '14) (SLMREB), Cons. Engr., Linde Air Products Co., 300 Madison Ave., New York 17; *for mail,* 122 N. Columbus Ave., Mt. Vernon, N.Y.
Obert, Edward Fredric ('38; '45) (EBFKN), Assoc. Prof. Mech. Engrg., Northwest. Tech. Inst.; *for mail,* 1204 Noyes St., Evanston, Ill.
Obrant, Leon Joseph (J'44), 5428 Woodland Ave., Philadelphia 43, Pa.
O'Brien, Charles Francis, Jr. (J'43) (CMO), 1st Lt., Army Serv. Forces, Redistribution & Salvage Officer, Chickasaw Ord. Works, Millington, Tenn.
O'Brien, Donald G. (J'40) (ABS), Capt., 233 LeCato, Audubon, N.J.
O'Brien, Eugene William ('21; '27; '31; F'46) (SECLO), *Manager,* '31-'34, *Vice-President,* '34-'36; V.P., W.R.C. Smith Publ. Co., 1020 Grant Bldg., Atlanta 3, Ga.
O'Brien, Francis R. (J'35) (FKS), 110 W. Magnolia, Oak Ridge, Tenn.
O'Brien, Frank L., Jr. (J'31) (SOM), Co-Owner, O'Brien Mchy. Co., 113 N. 3rd St., Philadelphia 6; *for mail,* Ithan Ave., Rosemont, Pa.
O'Brien, George William (J'44), 2840 Castlegate Ave., Pittsburgh 26, Pa.
O'Brien, Isaac Kell ('21; '35), Pur. Dept. Engr., Atlas Powder Co., Delaware Trust Bldg., Wilmington 99, Del.; *for mail,* 40 W. Ridley Ave., Ridley Park, Pa.
O'Brien, James K. ('44), 101 City Line Ave., Bala-Cynwyd, Pa.
O'Brien, James Walter (J'40) (DBS), Partner, Firm of C. Walker, Contractor, Box 258, Route 2, Germantown, Tenn.
O'Brien, John E. ('15), Mech. Rep., Association of American Railroads, 59 E. Van Buren St.; *for mail,* 7000 S. Shore Dr., Chicago, Ill.
O'Brien, John Edward (J'44) (JPS), Asst. Engr., Pacific Gas & Elec. Co., 245 Market St., San Francisco; *for mail,* 5491 Kales Ave., Oakland, Calif.
O'Brien, Jos., Jr. (J'38), 122 Highland Ave., Jersey City, N.J.
O'Brien, M. F. (J'36), 827 Tower Bldg., 14th & K Sts., N.W., Washington, D.C.
O'Brien, Morrough Parker ('29; '35) (BHK), Dean, College of Engrg., Univ. of Calif., Berkeley 4, Calif.
O'Brien, Patrick J. (J'41) (BJM), 9 Kendall Ave., Rutland, Vt.
Obrig, Adolph ('14) (DH), Sales Engr., Otis Elev. Co., 1375 E. 6th St., Cleveland 14; *for mail,* 3047 Edgehill Rd., Cleveland Heights, Ohio.
Obst, Charles Vincent ('39; '41) (NMBJSW), Engr., Apparatus Devel., Bell Tel. Labs., Inc., 463 West St., New York; *for mail,* 109-31—135th St., Richmond Hill, N.Y.
O'Callaghan, John (J'25) (DMZK), Plant Supt., Lone Star Cement Corp., 1812 Comer Bldg., Birmingham 3, Ala.
Ochiltree, Ned A. (J'42) (BNAZC), Research Engr., Research Labs., Gen. Motors Corp., 485 W. Milwaukee Ave., Detroit 2; *for mail,* 8325 E. Jefferson Ave., Detroit 14, Mich.
Ochtman, Leonard, Jr. ('16; '22; '29) (AEJBC), Mech. Engr., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 53 N. Monroe St., Ridgewood, N.J.
O'Connell, Raymond Griffin (J'41) (ME), c/o Torrington Co., Torrington, Conn.
O'Connor, William Donald ('41) (CHMN), Pat. Atty., Kearney & Trecker Corp., 6784 W. National Ave., West Allis 14; *for mail,* 2517 N. 85th St., Wauwatosa 13, Wis.
O'Connor, William John (J'41) (CR), 115 Prospect Pl., Rutherford, N.J.
O'Conor, Frank ('33; '46) (BCHLNT), Ch. Engr., Tolhurst Centrifugals Div., Am. Mch. & Metals, Inc., East Moline; *for mail,* 1194—27th St., Moline, Ill.
Odbert, John T. (J'28) (FKS), Asst. Ch. Engr., Centrifix Corp., 3029 Prospect Ave., Cleveland; *for mail,* 3562 Riedham Rd., Shaker Heights 20, Ohio.
Ode, Randolph Theodore ('01; '08) (ABCG), Pres., Secy., Providence Lithograph Co., Box 1266, 353 Prairie Ave., Providence, R.I.
Odegaard, Eugene Arthur (J'42) (ASFCBK), Analytical Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford 8, Conn.; *for mail,* 114 N. 5th St., Madison 4, Wis.
Odell, I. Newton ('43), Gen. Supt., Defender Photo Supply Co., Inc., Rochester 3; *for mail,* 229 Cobbs Hill Dr., Rochester 10, N.Y.
Odell, LeRoy L. ('28), Cons. Engr., Lincoln Bldg., 60 E. 42nd St., New York 17, N.Y.
Odell, Malcolm Jamieson (J'41) (MYWDRA), Dewey & Almy Chem. Co., Cambridge 40, Mass.
Oden, Curtis Grant (J'38) (CODMW), Engr., J. A. Jones Constr. Co., Inc., 209 W. 4th St., Charlotte, N.C.
Odenbach, Robert Charles (J'41) (OMCJBD), Treas., Ch. Engr., Odenbach Shipbldg. Corp., 4800 Dewey Ave., Rochester 12; *for mail,* 99 Thorncliffe Dr., Rochester 5, N.Y.
Odenweller, Hugo Franz (J'41) (BNO), Mech. Engr., Selas Corp. of Am., Erie & D St., Philadelphia 34; *for mail,* 601 Providence Rd., Clifton Heights, Pa.
Odom, Guy Thaine (J'44) (CMDJOW), Mech. Engr., Prod., H. P. Smith Paper Co., 5001 W. 66th St., Chicago 38; *for mail,* 156 N. Oak Park Ave., Oak Park, Ill.
O'Donoghue, John K. (J'41) (ESH), 227 Audubon Ave., New York 33, N.Y.
O'Donoghue, Mrs. Victoria (J'41) (SKH), Mech. Engr., with J. K. Blum, 205 E. 42nd St., New York; *for mail,* 227 Audubon Ave., New York 33, N.Y.
Oehrig, Henry Basil ('37), Mech. & Elec. Engr., Kahn & Jacobs, Archs., 2 Park Ave., New York, N.Y.
Oergel, Charles T. (J'29) (ABH), Devel. Engrg., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 135 Marble St., Stoneham, Mass.
Oesterle, Arthur Leo ('45) (CSJKRD), Gen. Supt., Holston Ord. Works, Army Serv. Forces, Kingsport, Tenn.
Oesterle, Paul Daniel ('29; '37) (SDFH), Asst. Mgr., Apparatus Sec., Dravo Corp., 300 Penn Ave., Pittsburgh 22; *for mail,* 135 Marlin Dr. E., Pittsburgh 16, Pa.
Oestnaes, Victor L. ('27), Ch. Engr., Am. Gas Accumulator Co., Elizabeth; *for mail,* 905 Summit Ave., Westfield, N.J.

Oetinger, George, Jr. (J'26) (DMZ), Maj., Ord. Dept., U.S.A.; *for mail,* 200 S. Huntington Ave., Boston, Mass.
Officer, Wesley J. (M'28) (CM), V.P., Blake Mfg. Corp., Green St., Clinton, Mass.
Offner, Walter W. ('45) (HSOA), Managing Engr., Partner, Indus. X-Ray Engrs., 420 Market St., San Francisco 11, Calif.
Ofner, Frank Robert (J'41) (EKCFMZ), Indus. Salesman, Tex. Co., 3640 N.W. St. Helena Rd., Portland; *for mail,* 5125 S.W. 49th Dr., Portland 1, Ore.
O'Gara, Edward Francis, Jr. (J'44) (FS), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* Main Rd., Tiverton, R.I.
Ogden, Nelson ('14; '35) (CESB), Gen. Mgr., Kingsbury Mch. Works, Inc., 4324 Tackawanna St., Philadelphia 24, Pa.
Ogden, Paul (J'43) (JBMNKZ), Sr. Engr., Phillips Petroleum Co., Bartlesville, Okla.
Ogden, Walter Headden ('44) (EJCKF), Engr., Phillips Petroleum Co., 135 S. La Salle St., Chicago 3, Ill.
Oge, George W. (J'40) (AH), Ensign, U.S.N.R.; *for mail,* 208 Morton St., San Antonio, Tex.
Ogg, Donald C. (J'40) (CDM), Capt., 1416 Court St., Adel, Iowa.
Ogle, Emerson Des Forges (J'40) (ABJC), Industry Sec. Supvr., SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34; *for mail,* 7009 McCallum St., Philadelphia 19, Pa.
Ogur, E. ('29; '35) (EFS), Cons. Mech. & Elec. Engr., 38 Crescent Terrace, Belleville, N.J.
O'Gureck, Richard Michael (J'44) (BCM), 686 Forest St., Kearny, N.J.
O'Hagan, Rae Selwyn (J'45) (CMY), Co. Mgr. & Cons. Engr., Plastic & Diecasting, Ltd., 14 Hazeldean Rd., Christchurch, New Zealand.
Ohart, Theodore Charles ('29; '35; '37) (CO), Dist. Engr., Lamp Dept., Gen. Elec. Co., 901 Genesee Bldg., Buffalo 2; *for mail,* 250 Bernhardt Dr., Buffalo 21, N.Y.
Ohgren, Carl Warren (J'43) (BCJMNY), Mech. Engr., Swain Nelson Co., Glenview; *for mail,* 4954 N. Kilpatrick Ave., Chicago 30, Ill.
Ohler, Ralph Edwin ('44) (CHM), Prod. Mgr., Constr. Mchy. Co.; *for mail,* 1105 Steely St., Waterloo, Iowa.
Ohren, George Albert (A'14) (ASE), Asst. to John E. Mackel, Struc. Engr., 234 N. Main St., Los Angeles 12; *for mail,* 5289 Coliseum St., Los Angeles 16, Calif.
O'Keefe, Philip ('42) (FSEH), V.P., Byrne Associates Inc., 80 Wall St., New York 5; *for mail,* 33 Haven Esplanade, Staten Island 1, N.Y.
Okey, Perry ('16), Owner, Mgr., Okey Mfg. Co., 562 Dennison Ave., Columbus 8; *for mail,* 765 Bryden Rd., Columbus 5, Ohio.
Okner, Bernard S. (34) (ABCNRS), Pres., Indus. & Automotive Bearing Co., 1313 N. Clybourn Ave., Chicago 10, Ill.
Oladko, Miss Helen (J'43) (HA), Devel. Engr., Purolator Products Inc., 91 Empire St., Newark 5, N.J.
Oldacre, William H. ('36) (BCDJMN), Pres., Gen. Mgr., D. A. Stuart Oil Co., 2727 S. Troy St., Chicago 23, Ill.
Oldenkamp, Henry A. ('43) (JBCNTH), Devel. Engr., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland; *for mail,* R.F.D. 2, 3360 Sprague Rd., Cleveland 9, Ohio.
Oldham, Edward Lindsay ('19; '35) (ABCGN), Adv. Mgr., Cleveland Pneumatic Tool Co., 3781 E. 77th St., Cleveland 5; *for mail,* 1512 E. 118th St., Cleveland 6, Ohio.
Oldham, Percy Talbert ('30) (JY), Mgr., Mar. Div., By-Products Steel Corp., Coatesville, Pa.
Olditch, Frederick William ('17) (SDOH), Retired; Estancia Las Higueritas, Alta Gracia, F.C.C.A., Argentina, S.A.
Oldright, William (J'41) (AJM), Power Plant Design Unit, Boeing Aircraft Co.; *for mail,* 1500 Westlake Ave. N., Seattle, Wash.
Oldson, Norman P. (J'40) (KSA), Assoc. Engr., (Mar.), Bur. of Ships, Navy Dept., Washington 25; *for mail,* 1412 Whittier St., N.W., Washington 12, D.C.
O'Leary, John J. ('30) (CLMBS), Regional Mgr., Atmospheric Nitrogen Corp., Ironton, Ohio.
O'Lenick, Anthony John (J'45) (NCDBMJ), Designer, Conmar Products Corp., 140 Thomas St., Newark 1; *for mail,* 162 W. 49th St., Bayonne, N.J.
Oles, Herbert E. (J'40) (HMS), P. R. Mallory & Co., Inc.; *for mail,* 4923 Graceland, Indianapolis, Ind.
Olgardt, Jack Paul ('42) (CM), 280 Menlo Oak Dr., Menlo Park, Calif.
Olin, F. W. (A'95), Pres., West. Cartridge Co., East Alton, Ill.
Olin, Spencer Truman ('26; '28; '35) (CMDJ), 1st V.P., Olin Industries, Inc., East Alton, Ill.
Olive, R. W. ('29; '35), Test Engr., Duke Power Co., Charlotte, N.C.
Olive, Theodore Ristine ('31; '45) (LZYKDT), Assoc. Editor, *Chemical & Metallurgical Engineering,* McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.
Olivenstein, Martin N. (J'41) (AEB), 100 Van Cortlandt Pk. S., Bronx 63, N.Y.
Oliver, Charles Estes (J'39) (SBJ), Mech. Engr., Westcott & Mapes, Inc., Archs. & Engrs., 109 Church St., New Haven; *for mail,* 53 Haverford St., Hamden, Conn.
Oliver, Ellis William (J'41) (WJE), Lt., U.S.N.R., 710 J St., Centralia, Wash.
Oliver, Frank J. ('22; '28; '35) (NYZMJ), Editor, *Electrical Manufacturing,* Gage Publ. Co., 1250 Ave. of the Americas, New York 20, N.Y.
Oliver, Horace G., Jr. (J'35) (CJL), Lt., Asst. to Constr. QM., Office of Constr. QM., Pine Camp, Great Bend, N.Y.
Oliver, Owen Norman (J'45) (MNJY), Equip. Design Engr., Halliburton Oil Well Cementing Co.; *for mail,* 711 N. 12th St., Duncan, Okla.
Olivetti, Adriano (J'26), Ivrea, Italy.
Olivetti, Dino (J'41) (MSA), Mech. Engr., Research & Devel., Diamond Instrument Co., Wakefield; *for mail,* 54 Abbot St., Andover, Mass.
Olmstead, Arthur E. (J'38) (SKF), Mar. Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail,* 34 Park St., Pulaski, N.Y.
Olofson, E. Clifford (J'35) (BCJ), Lt., U.S.N.R., 59 Shadylawn Dr., Madison, N.J.
O'Loughlin, William Henry (J'46) (CDM), 215 Douglas St., Stratford, Ont., Can.
Olowinski, Edward J. (J'45) (SKB), Research Asst., Engrg. Exper. Sta., Pa. State College, State College, Pa.
Olschner, Heinz J. (J'42) (ABFKNS), Mech. Engr., 10 Fairway Ave., Belleville 9, N.J.
Olsen, Clyde Russell (J'45) (MBNOSA), 231 N. 1st St. W., Ephraim, Utah.
Olsen, Gustav Edwin (J'20) (KMCDF), V.P., Mgr. of Sales, Fitzgibbons Boiler Co. Inc., 101 Park Ave., New York 17; *for mail,* 68-09 Beach Channel Dr., Arverne, N.Y.
Olsen, Joseph O. ('45) (NOHMBC), Pres., Productioneering Associates, 526 W. State St., Rockford, Ill.
Olsen, Leroy (J'38), Dept. of Mech. Engrg., Carnegie Inst. of Tech., Pittsburgh, Pa.
Olsen, Olaf LaCour ('24; '41) (EHKL), Owner, Firm of O. L. Olsen, 610—2nd Natl. Bank Bldg., Houston 2, Tex.
Olsen, Robert Arnold (J'42), 333 W. Stadium Ave., West Lafayette, Ind.
Olsen, Robert Carlton (J'44) (CES), 309 Sherwood Pl., Stratford, Conn.
Olson, Carlton Tropp (J'41) (EFZAM), Automotive Engr., Research & Devel. Lab., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail,* 341 Roosevelt Ave., Pitman, N.J.
Olson, Donald Richard (J'45) (KSNFAB), Instr., Mech. Engrg. Dept., Yale Univ., 400 Temple St., New Haven, Conn.
Olson, Emil (J'29) (MSCF), Supt. of Prod., Hawaiian Elec. Co., Ltd., 900 Richards St., Honolulu, T.H.
Olson, Ernest William ('29; '42) (YNLCMS), Ch. Engr., Firestone Rubber & Latex Co., 1 Firestone Ave., Fall River; *for mail,* Pearse Rd., Swansea, Mass.
Olson, F. Stuart (J'40), 6 McGrady St., Glen Cove, L.I., N.Y.
Olson, Foster A. (J'38) (CJL), Oper. Engr., N.J. Zinc Co.; *for mail,* Horse Head Inn, Palmerton, Pa.
Olson, Geo. D. ('36) (SCF), Ch. Engr., Commonwealth Edison Co., 3400 N. California Ave., Chicago 18, Ill.
Olson, Gordon Glenn (J'43) (SKE), Application Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 210 N. 74th St., Milwaukee 13, Wis.
Olson, Harold Thornton ('44) (CNZM), Engr., Remington Rand, Inc., 67—35th St., Brooklyn 32, N.Y.
Olson, Malven Leonard ('45) (FLCBK), Sr. Engr., Equip. Devel., Prest-O-Lite Co., Inc., Box 1478, Indianapolis 6, Ind.
Olson, Martin L. ('14; '19) (ACM), Retired; 81 West St., Hyde Park 36, Boston, Mass.
Olson, Mauritz M. (J'40) (NABEJ), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail,* 239 Raleigh Ave., Hampton, Va.
Olson, Peter G., Jr. (J'45) Foreman, Large Turbine Erection, Westinghouse Elec. Corp., Lester; *for mail,* R.F.D. 2, Media, Pa.
Olson, Raymond G., Jr. (J'44), 707 York Rd., Glenview, Ill.
Olson, Reuben M. (J'41) (SBKW), Mech. Engr., Naval Ord. Lab., Naval Gun Factory, Washington 25, D.C.; *for mail,* 6614 Hillandale Rd., Chevy Chase 15, Md.
Olson, Robert George ('43) (ESHA), East. Mgr., Hyd. Coupling Div., Am. Blower Corp., 50 W. 40th St., New York 18; *for mail,* 108 E. 38th St., New York 16, N.Y.
Olson, Robert V. ('29; '35) (CMBTJA), Pres., Gen. Mgr., Mossberg Pressed Steel Corp., West St., Attleboro, Mass.
Olson, Theodor M. (J'41) (BEM), Lt. (j.g.), U.S.N.R.; 165 Laurel Ave., North Palo Alto, Calif.
Olson, William Harry Milton (J'38) (CMDN EA), Spec. Assignments, Factory Layout, Turbine Mfg. Div., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 1958 Wabash Ave., Schenectady 6, N.Y.
Olsson, Charles David (J'41) (MNJY), Sales Engr., Olsson & Pierce Co., 111 Broadway, New York 6, N.Y.
Olsson, T. Karl (J'41) (BHJ), Test Engr., Ford Instrument Co., 31-10 Thompson Ave., Long Island City; *for mail,* 186-05 Ilion Ave., St. Albans 12, N.Y.
Olt, Richard Greve ('44; '44) (ZMLCJK), Devel. Engr., Optron Lab., 2655 Salem Ave.; *for mail,* 1819 Ravenwood Ave., Dayton 6, Ohio.
Olvany, William James, Jr. (J'43) (EFKS), 109-40—71st Rd., Forest Hills, N.Y.
Olver, Albert Stephen (J'29) (APR), Gen. Sales Dept., Imperial Oil, Ltd., Imperial Oil Bldg., Toronto 1, Ont., Can.
Olver, Frank Monroe (J'42) (ACJ), Asst. Stress Analyst, Glenn L. Martin Co., Baltimore; *for mail,* Box 513, Cent. Y.M.C.A., Franklin at Cathedral St., Baltimore 1, Md.
O'Mara, Edward Andrew (J'38), Lt., U.S.N., Washington, D.C.; *for mail,* 78 Central Ave., Montclair, N.J.
O'Mara, Richard F. ('28; '37) (LSCF), Sales Mgr., West. Precipitation Corp., 1016 W. 9th St., Los Angeles 15, Calif.
Omelianoff, Geo. M. (J'38) (NHACDG), Estimating Engr., Progressive Welder Co., 3050 E. Outer Dr., Detroit 12; *for mail,* 15 Bartlett St., Detroit 3, Mich.
Onderdonk, Paul Truman ('27; '35) (SHC), Inspc. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 204 Kensington Rd., Garden City, N.Y.
Ondr, Joe Oliver, Jr. (J'44) (AW), 67 Ridgemoor Dr., Clayton 5, Mo.
O'Neil, Frederick William ('01; '08), V.P. & Ch. Engr., Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.; *for mail,* Oklawaha, Marion Co., Fla.
O'Neil, John Francis (J'41) (ABC), Sales Engr., Busch-Sulzer Bros.-Diesel Eng. Co., 3300 S. 2nd, St. Louis; *for mail,* 819 S. Gray Ave., Webster Groves 12, Mo.
O'Neil, Robert Dravo ('27; '36) (EFZOHK), Engr., Am. Meter Co., Inc., 1104 Watts Bldg., Birmingham 3, Ala.
O'Neill, George W. ('28; '34) (DFS), Maj., Petawawa Military Camp, Petawawa, Ont., Can.
O'Neill, James Lawrence ('42) (CMOD), Indus. Engr., Geo. Scher Engrg. Co., 115—1st St., Newark 7; *for mail,* 78 Pennington Ave., Passaic, N.J.
O'Neill, Theodore (J'43) (YDC), Field Engr., Goodyear Tire & Rubber Co. of Can., Ltd., New Toronto; *for mail,* 69 Canning Ave., Islington, Ont., Can.
O'Neill, William Charles, III (J'43) (CRS), U.S. Navy; Huntington Valley, Pa.
Onsrud, Rudolph F. ('35) (WNJCA), Pres., Gen. Mgr., Onsrud Mch. Works, Inc., 3900 Palmer St., Chicago 47, Ill.
Onuf, Bronis R. (J'44) (KEBS), Instr., Yale Univ., 400 Temple St., New Haven 11, Conn.
Oonk, Rowland George (J'45), Ford Motor Co., 5410 Orchard St., Dearborn; *for mail,* 11832 Ohio, Detroit 4, Mich.
Oonk, William Joseph ('22; '37) (SWKDRL), Dist. Application Engr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., 411 N. 7th St., St. Louis 1; *for mail,* 4548 Red Bud Ave., St. Louis 15, Mo.
Oosten, Louis Stanley (J'44) (BHK), Asst. Ch. Engr., Bell & Gossett Co., Morton Grove; *for mail,* 725 Case St., Evanston, Ill.
Opatowski, I. ('40) (BWAKHJ), Research Staff Member, Univ. of Chicago, 5822 Drexel Ave., Chicago 37; *for mail,* 1047 W. North Shore Ave., Chicago 26, Ill.
Openshaw, Arthur Benson ('46) (FKS), Dist. Mgr., Combustion Engrg. Co., Inc., 2330 Bankers Bldg., Chicago 3, Ill.
Ophuls, Fred ('14) (EFKOS), Owner, Fred Ophuls & Associates, 112 W. 42nd St., New York 18, N.Y.
Opila, Francis Anthony (J'43) (BCZJMR), Asst. Supt. of Southwark Div., Baldwin Loco. Works, Eddystone; *for mail,* 101 Gladstone Rd., Lansdowne, Pa.
Opkins, Jerold Stanley (J'43) (ASNMER), 1st Lt., Corps of Engrs., U.S.A., Supply Officer, 496th Engr. Heavy Shop Co., Gp. III, Army Serv. Forces Trng. Center, Ft. Belvoir, Va.
Opp, George C. A. ('42) (SCOMFR), Safety Consultant, Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 20144 Renfrew Rd., Detroit 21, Mich.
Oppenheimer, Arthur C., II (J'41) (BCE), Lt. (j.g.), Ord. Officer, c/o Asst. to Indus. Mgr., U.S.N., Box 1423, Mobile; *for mail,* Spring Hill, Ala.
Oppenheimer, Edwin Adam ('28; '34; '35) (SF LDKC), M.M., Penick & Ford Ltd., Inc., Cedar Rapids, Iowa.
Oppenheimer, Philip H. ('27; '35) (FKLOSZ), Cons. Engr., 225 E. Broad St., Westfield, N.J.
Opsahl, Emil ('42; '45) (DNCS), Mech. Engr., Link-Belt Co., Ltd., 791 Eastern Ave., Toronto, Ont., Can.
Oquendo, Raul (J'41) (CHS), Ulacia, S.A., Central Ulacia, Rodrigo, Provincia de las Villas, Cuba.
Oran, Samuel (J'45) (BMO), Mech. Designer, Barnes & Reinecke, 230 E. Ohio St., Chicago 11; *for mail,* 5804 W. Catalpa, Chicago 30, Ill.
Orban, Alex Richard (J'42) (ABJ), Mech. Engr., Dowmetal Engrg. Lab., Dow Chem. Co.; *for mail,* 322 N. Lincoln Ave., Bay City, Mich.
Orbeck, Einar Melvin ('27; '37) (MNACJ), Mech. Designer, Hamilton Stand. Propellers Div., United Aircraft Corp., East Hartford 8; *for mail,* 9 Linnard Rd., West Hartford 7, Conn.
Orcutt, Arthur Hitchcock ('36) (NHC), Managing Dir., Gear Grinding Co. Ltd., Cranmore Blvd., Shirley, Birmingham; *for mail,* Rowington Hall, Nr. Warwick, England.

Ordway, Earl P. ('14; '17), Pres., Union Steam Pump Co.; *for mail,* 151 Emmett St., Battle Creek, Mich.
O'Regan, William Frederick ('42) (SFEG), Mech. Inspr., c/o Supvr. of Shipbldg., U.S.N., Quincy; *for mail,* 23 Crockett Ave., Dorchester 22, Mass.
Oren, Jess William, III (J'45), Research Asst., Pa. State College, Box 436, State College; *for mail,* 307 N. George St., Pottsville, Pa.
Orenstein, Hartwig ('22) (OMCDJW), Cons. Engr., 140 Queen's Drive, Glasgow, S. 2, Scotland.
Orent, Edward (J'44), Layout Draftsman, Detroit Transmission Div., Gen. Motors Corp., 5140 Riopelle, Detroit; *for mail,* 523 Crosby St., Grand Rapids 4, Mich.
Oreskovich, Peter Joe (J'41) (ONBJEM), Pres., Oreskovich & Co., Louisville Trust Bldg., Louisville 2; *for mail,* 1000 Oakwood Ave., Louisville 8, Ky.
Orlando, Patrick Joseph (J'46) (BFK), 2209 Dietz Pl., North Bergen, N.J.
Orloff, Serge ('42; '44) (NBMC), Ch. Design Engr., Am. Chain & Cable Co. Inc., 1575 S. 6th St., Camden, N.J.
Ormond, Alex C. (J'40) (OSACMJ), Draftsman, Union Bag & Paper Corp.; *for mail,* 102 E. 44th St., Savannah, Ga.
Ormond, Alex. M. ('23) (CFL), Engr., Savannah Sugar Refining Corp.; *for mail,* 102 E. 44th St., Savannah, Ga.
Ormondroyd, Jesse ('31) (BEHASR), 2104 Copley Ave., Ann Arbor, Mich.
Ormston, Alfred J. (J'39) (AKS), Jr. Engr., Plant Constr. Dept., Fla. Power Corp.; *for mail,* 337-8th St. N., St. Petersburg, Fla.
Orning, Arthur Alonzo ('45) (FKSE), Mem. Staff, Coal Research Lab., Carnegie Inst. of Tech., Schenley Park, Pittsburgh 13, Pa.
Orno, Knud E. ('27) (CLNT), Mgr., Engrg. Sec., Rayon Div., E. I. du Pont de Nemours & Co., Wilmington; *for mail,* Apt. C-205, 1300 Harrison St., Wilmington 12, Del.
O'Rourke, Hugh Dominic, Jr. (J'38) (KS), B. F. Sturtevant Div., Westinghouse Elec. Corp., 812 Mich. Theatre Bldg., Detroit 26, Mich.
O'Rourke, John Thomas (J'46) (GJ), 530 Vidal Dr., Parkmerced, San Francisco, Calif.
O'Rourke, Patrick Edward ('41) (ABCES), Pres., O'Rourke & Royer Inc., 740 Gas & Elec. Bldg., Denver 2, Colo.
Orr, Alexander Macklin ('98; '23), Retired; Box 311, Edgartown, Mass.
Orr, Carol Lister ('21; '23; '29) (CLSFKH), Power Plant Mgr., Tex. Gulf Sulphur Co., Newgulf, Tex.
Orr, Claud Henry ('19; '26) (CMD), Supt., Natl. Carbon Co. Inc., Box 749, Charlotte 1, N.C.
Orr, Fred B. ('21) (FSYAE), Retired; 1618 Sunset Plaza Dr., Los Angeles 46, Calif.
Orr, Howard S. (J'32) (JB), Turn Foreman, Carnegie-Ill. Steel Corp., N. Buchanan; *for mail,* 444 Marshall St., Gary, Ind.
Orr, J. L. ('41), 920 E. Haines St., Philadelphia 38, Pa.
Orr, John ('09), Prof. Engrg., 14 Hyde Park Mansions, Park Lane, Parktown, Johannesburg, S. Africa.
Orr, Leighton ('44) (ABK), Research Engr., Research Lab., Pittsburgh Plate Glass Co., Creighton; *for mail,* 1116 Cambridge St., Tarentum, Pa.
Orr, Wesley Leland ('44) (BONCH), Assoc. Prof. Engrg., Univ. of Calif. at Los Angeles, 405 Hilgard Ave., Los Angeles 25; *for mail,* 728—23rd St., Santa Monica, Calif.
Orr, William Austin (J'42) (AJSNMC), Engr., U. S. Maritime Serv., War Shipping Admin., Washington, D.C.; *for mail,* 3914 Lindell Blvd., St. Louis 8, Mo.
Orr, William R. (J'41) (ACJ), N.A.T.S.-Pacific, Navy 29, F.P.O., San Francisco, Calif.
Orrell, John E. ('34; '43) (MESZJC), Mech. Engr., Shell Oil Co., Inc., Box 2010, Wichita Falls, Tex.
Orrok, George A., Jr. ('31; '39) (SKBFHJ), Acting Asst. Supt. of Engrg., Boston Edison Co., 39 Boylston St., Boston 12; *for mail,* 5 Cleveland St., Cambridge 38, Mass.
Orsini, Anthony (J'42) (MLAS), Devel. Engr., Walter Kidde & Co., 675 Main St., Belleville 9; *for mail,* 122—65th St., West New York, N.J.
Orskog, Arthur Gordon (J'45) (FKO), Canadian Ice Mch. Co., 65 Villiers St., Toronto 2, Ont., Can.
Orth, Harry R. (J'46) (SYZ), 3131 Forest, Kansas City, Mo.
Ortla, Frederick L. ('27), 25 Belvoir Rd., Milton, Mass.
Ortman, Hadar ('30; '35) (CG), V.P., McClure, Hadden & Ortman, Inc., 111 W. Washington St., Chicago, Ill.
Ortner, Louis ('13), Ch. Engr., Dept. of Hospitals, City of New York, 10th Floor, Mun. Bldg., New York; *for mail,* 644—77th St., Brooklyn, N.Y.
Orton, Albert William ('42) (ZCLM), Factory Mgr., Pur. Agt., Partlow Corp., 2 Campion Rd., New Hartford; *for mail,* 39 Parkside Court, Utica 3, N.Y.
Orton, Raymond E. ('40) (CBMJ), Ch. Engr., Mall Tool Co., 7740 S. Chicago Ave., Chicago 19; *for mail,* 8055 Paxton Ave., Chicago 17, Ill.
Orvis, Ward Charles (J'43) (EHFKBC), Asst. Distribution Design Engr., So. Counties Gas Co. of Calif., 810 S. Flower St., Los Angeles 14, Calif.
Osborn, Oliver (J'45) (JZ), Supvr., Electro-Chem. Engrg. Dept., Dow Chem. Co.; *for mail,* 615 W. 4th St., Freeport, Tex.
Osborne, Bodwell Doe (J'41) (BLSC), Lt., U.S.N.R., 133 Woodland Rd., Grove Park, Asheville, N.C.
Osborne, Harry Joseph ('44) (MNCBJ), Ch. Engr., Ill. Watch Case Co., 853 Dundee Ave.; *for mail,* 170 Grand Blvd., Elgin, Ill.
Osborne, Latham Edgar ('43) (CM), V.P., Westinghouse Elec. Corp., Box 1017, 306—4th Ave., Pittsburgh 30, Pa.
Osborne, William Carle (J'42) (HSKBEZ), Instr. in Mech. Engrg., Rensselaer Poly. Inst., Troy, N.Y.; *for mail,* 284 Ridge St., Newark 4, N.J.
Osbourne, William A. ('30; '35) (CSFM), V.P., Babcock-Wilcox & Goldie-McCulloch, Galt, Ont., Can.
Oschwald, Arthur, Jr. (J'44) (EHK), Exper. Test Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail,* 495 Main St., Orange, N.J.
Osgood, Carol Eugene ('39) (NGABO), Design Engr., Douglas Aircraft Co., Inc., El Segundo; *for mail,* 3271 Malcolm Ave., Los Angeles 34, Calif.
Osgood, Charles Frederick (A'45) (NHDBMJ), Pat. Atty., Pat. Dept., Sullivan Div., Joy Mfg. Co., Claremont, N.H.
O'Shea, Daniel W. ('27) (FHS), Buckingham, Que., Can.
Osipovich, A. A. ('36) (AHS), Mech. Engr., Asst. Ch., Transmission Design, Bonneville Power Admin., U.S. Dept. of Interior, 1300 N.E. Union Ave., Portland; *for mail,* Apt. 3, 5565 N.E. Sandy Crest Terrace, Portland 13, Ore.
Oskochil, Joseph William (J'42) (CM), Mfg., Wage Rate & Time Study, Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 11 Rockland St., Swampscott, Mass.
Osmundsen, Norman Kristian (J'44), 7010 Churchland St., Pittsburgh 6, Pa.
Osojnak, Boris Mladen (J'44) (BANEJH), Analytical Engr., Lycoming Div., Aviation Corp., Williamsport; *for mail,* 1804 Memorial Ave., Williamsport 38, Pa.
Ossman, Ernest H. (J'44) (CJW), Ave. 16 de Septiembre 6-301, Mexico, D.F., Mex.
Ostborg, John (J'40) (BEM), R.R. 6, Redmond Rd., Springfield, Ohio.
Oster, Eugene Arthur ('17; '20; '29) (FKBDZ), Mech. Engr., French Bauer Inc., Plum & Central Pkwy., Cincinnati 2; *for mail,* 3913 Regent Ave., Norwood Sta., Cincinnati 12, Ohio.
Osterman, Philip Campbell ('12) (CFJT), V.P., Gen. Mgr., Am. Gas Furnace Co., Spring & Lafayette Sts., Elizabeth B; *for mail,* 19 Stiles St., Elizabeth 3, N.J.
Ostermann, Rudolf Maximilian ('13; F'46) (RS FEK), V.P., Superheater Co., 122 S. Michigan Ave., Chicago 3, Ill.
Ostrach, Simon (J'44) (ABKHS), Mech. Engr., Analyst, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 12010 Saywell Ave., Cleveland 8, Ohio.
Ostrognai, Allen Gilbert (J'42) (BKHN), Engr., Fan Engrg. Dept., Gen. Elec. Co., 1285 Boston Ave., Bridgeport 2; *for mail,* Apt. 103, Door 118, Bldg. 17, Court G, Yellow Mill Village, Bridgeport 8, Conn.
Ostrom, Ernest Arthur ('43) (JMWDCH), Prod. Mgr., Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15; *for mail,* 1112 E. 78th St., Kansas City 5, Mo.
Ostwald, Richard ('39; '45) (NZBJKC), Devel. Engr., Ingersoll-Rand Co., Phillipsburg, N.J.; *for mail,* 1458 Northampton St., Easton, Pa.
Osuch, Eugene B. (J'41) (CFS), Ensign, U.S.N.R.; 826 Warren St., Utica, N.Y.
Oswald, Gilbert Leverich (J'42) (CMDLSZ), 130 Pine St., Shelton, Wash.
Othmer, Donald Frederick ('45) (LWKYB), Prof. Chem. Engrg., Head of Dept., Poly. Inst. of Brooklyn, 99 Livingston St., Brooklyn 2, N.Y.
Othus, James Carey ('28) (BNDCMW), Engr., West. Mchy. Corp., 816 S.E. Taylor St., Portland 14; *for mail,* 3842 N.E. 32nd Pl., Portland 12, Ore.
Otis, William Grady (J'45) (EHS), Great Lakes Pipelines, Kansas City; *for mail,* 908 Benton Blvd., Kansas City 1, Mo.
Otocka, Edward Andrew (J'37) (CML), Mech. Engr., Am. Home Products Inc., 22 E. 40th St., New York, N.Y.; *for mail,* 242 Kent Pl., Summit, N.J.
O'Toole, Charles Bartholomew (J'43) (NGB), Mech. Engr., G. W. Klages & Son Inc., 804 Sara St., S.S., Pittsburgh; *for mail,* 151 Chesterfield Rd., Pittsburgh 13, Pa.
O'Toole, Joseph M. (J'37) (BAJKN), Matls. & Processes Engr., Aircraft Gas Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn 3, Mass.
Otrembiak, John Joseph (J'41) (NBHLJK), Engr., Designing, Acme Steel Co., 134th St. & Perry Ave., Riverdale Sta., Chicago; *for mail,* 11825 Eggleston Ave., Chicago 28, Ill.
Ott, Arnold ('40) (NMYOAC), Design Engr., Armament Div., Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21; *for mail,* 4961 McPherson Ave., St. Louis 8, Mo.
Ott, George Edwin, Jr. (J'40) (CEMD), Indus. Engr., Cleveland Equip. Works, Gen. Elec. Co., 1175 E. 152nd St., Cleveland; *for mail,* R.F.D. 3, Willoughby, Ohio.
Otte, Arthur F. (J'34) (SEFRZC), Instr., Power Plants, Detroit Pub. Schs., 4333—12th St., Detroit 8, Mich.
Otte, Karl Henry ('26; '37) (CLM), Indus. Engr., Purity Bakeries Serv. Corp., 919 N. Michigan Ave., Chicago 11; *for mail,* 1320 Rosedale Ave., Chicago 40, Ill.
Otter, George Edward (J'38) (ABJL), Ch. Engr., Fleet Aircraft Ltd., Gilmore Rd., Ft. Erie, Ont., Can.
Otterbacher, Elliott H. (J'38), Lt. Comdr., Naval Trng. Sch., Princeton Univ., Princeton, N.J.
Otterbein, Mark Edward (J'35) (BJMN), Exper. Engr., Hyatt Bearings Div., Gen. Motors Corp., Harrison; *for mail,* 5 Mayfair Rd., Morris Plains, N.J.
Otto, Carl A. ('38) (ZMKAJY), Ch. Engr., Johnson Serv. Co., 507 E. Michigan St., Milwaukee 2, Wis.
Otto, Charles Raymond ('38; '46) (ZJND), Instr., Engrg. Matls. Dept., Cornell Univ., Ithaca, N.Y.
Otto, Harold M. (J'32) (BKS), Mech. Engr., Design, Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Otto, Henry Chester Linton (J'41) (CDM), 150 Secatogue Ave., Farmingdale, L.I., N.Y.
Otto, Herbert George (J'44) (MHKNY), Plant Engr., Die Cast Div., Elec. Auto-Lite Co.; *for mail,* 425 N. Hoy St., Woodstock, Ill.
Otto, Herbert Robert, Jr. (J'38) (MYCHLD), Lt. (j.g.), U.S.N.R., Navy Yard Annex, Tompkinsville, S.I., N.Y.; *for mail,* 123 James St., Hillside 5, N.J.
Otto, William Henry (J'39) (CMLD), Scheduling Engr., Wilmington Shops, E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington 97; *for mail,* 18 Boxwood Rd., Wilmington 177, Del.
Otzmann, Henry, Jr. ('44) (BJM), Mech. Design Engr., Westinghouse Elec. Corp., 95 Orange St., Newark, N.J.
Ouzts, John Andrew, Jr. (J'41) (CJ), Co-Owner, Southern Welding Co., 623 Ellis; *for mail,* 1017 Adrian St., Augusta, Ga.
Ovaitt, David Wheeler ('42) (CMW), El-Cid Apts., West Palm Beach, Fla.
Overbagh, John S. (J'41), Fraser Paper, Ltd., Madawaska, Me.
Overman, Harold S. (J'38) (NBOZM), 5802—16th St. N., Arlington, Va.
Overton, David Ayward (J'45) (CMORTL), Asst. Tech. Engr. (Indus.), Carbide & Carbon Chemicals Corp., Box P, Oak Ridge; *for mail,* 501 E. Oak Hill, Knoxville 17, Tenn.
Overton, Ralph M. ('24; '30) (CSFA), U.S. Steel Corp., of Del., 436—7th Ave., Pittsburgh 30; *for mail,* 2006 Duquesne Ave., McKeesport, Pa.
Overton, William J. ('30), Ch. Engr., Deerfield Packing Corp., Seabrook Farms, Bridgeton; *for mail,* R.F.D. 1, Elmer, N.J.
Ovesen, Henrik ('14; '22) (DJLORY), Cons. Engr., Box 14, Poland, Ohio.
Ovestrud, Melvin (J'17), V.P., Supt., Pioneer Gravel Equip. Mfg. Co., 1515 Central Ave.; *for mail,* 3825 Chowen Ave. S., Minneapolis, Minn.
Ovregaard, Olav ('45), Naval Arch., c/o Asst. to Indus. Mgr., U.S.N., Ferry Bldg., San Francisco; *for mail,* 505 Haddon Rd., Oakland 6, Calif.
Owen, A. S. H. A. ('37), Lecturer in Engrg., South-East Essex Tech. College, Dagenham, Essex; Inspc. & Cons. Engr., Robt. Bruce & Sons, Moorgate Hall, London; *for mail,* 58 Dominion Dr., Romford, Essex, England.
Owen, Ernest V. ('24) (EKLSFH), Chem. Engr., E. B. Badger & Sons Co., 75 Pitts St., Boston 14, Mass.
Owen, Harry Scheu (J'42) (MDNEZJ), Engr., Reynolds Metals Co., Grand Rapids, Mich.; *for mail,* 411—11th St., N.E., Canton 4, Ohio.
Owen, John Malcolm ('45) (OCDM), Mech. Engr., Corning Glass Works; *for mail,* 119 E. Pulteney St., Corning, N.Y.
Owens, Charles Trump ('12; '20) Retired; Box 326, Chestertown, Md.
Owens, Frederick Richard ('44) (OSCKFJ), Secy., Serv. Div., Cyrus Wm. Rice & Co., Inc., 15 Noble Ave., Pittsburgh 5, Pa.
Owens, Herbert Malcolm (J'38) (BHS), Federalsburg, Md.
Owens, Jas. Whitfield ('29), Dir. of Welding, Fairbanks, Morse & Co., Beloit, Wis.
Owens, William Bruckner (J'40) (FNWY), Automotive Engr., Ga. Power Co., 75 Marietta St., Atlanta; *for mail,* 805 Church St., Decatur, Ga.
Owsley, William David ('42) (CNMHJ), Ch. Engr., Halliburton Oil Well Cementing Co., Duncan, Okla.
Ozley, Geo. R. ('28; '34), Mech. Engr., Ala. By-Products Corp., Box 7, Tarrant Branch, Birmingham, Ala.

P

Paca, John Philemon ('42) (JNCMBY), V.P., Charge Product Engrg., Diebold Inc., 818 Mulberry Rd., Canton 2; *for mail,* 810—25th St., N.W., Canton 3, Ohio.
Pace, Dominick Martin (J'45) (ABNDHS), Mech. Designer, Semet-Solvay Engrg. Corp., 44 Wall St., New York; *for mail,* 3488 Seymour Ave., Bronx 67, N.Y.

Pace, Edwin Lancaster (J'40) (SBK), Mech. Engr., Auxiliary Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 400 Puritan Rd., Swampscott, Mass.
Pacey, John J. (J'46) (ACM), Engr., Aro Equip. Corp.; *for mail,* Box 387, Bryan, Ohio.
Pach, Leo (J'41) (ESB), Pvt., Co. C., 5th Bn., Army Serv. Forces Trng. Center, Ft. Belvoir, Va.; *for mail,* 72 Lincoln Ave., Tuckahoe, N.Y.
Pach, Stewart Warner (J'42) (CMAD), Student, Sch. of Business, Columbia Univ., New York; *for mail,* 98 Joralemon St., Brooklyn 2, N.Y.
Pacher, Andrew Joseph (J'42) (ARS), Devel. Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* Beagle Manor, Cloutman's Lane, Marblehead, Mass.
Pachl, Pierson Russell (J'38) (ALJC), Lt. (j.g.), U.S.N.R., Office of Representative of Bur. of Aero., Navy Dept., Ryan Aero. Co.; *for mail,* 2843 E St., San Diego, Calif.
Packard, Horace Nelson ('11; '19) (ZCMJ), V.P., Treas., Cambridge Instrument Co., Inc., 73 Spring St., Ossining, N.Y.
Packard, Kenneth Adkins (J'36) (SHK), Dist. Serv. Engr., Westinghouse Elec. Corp., 540 Grant St., Bridgeport 8, Conn.
Packard, Roland Alfred ('18) (ACFMNS), Ch. Engr., Smith Paper Co. Inc.; *for mail,* Whiteholme Rd., Lee, Mass.
Packer, John B., Jr. (J'37), Crown Can Co., G & Erie Ave., Philadelphia; *for mail,* 427 Owen Rd., Wynnewood, Pa.
Packer, Leo S. (J'41) (MJCB), Lt., Hdq., 1180th Engr. Constr. Gp., A.P.O. 994, c/o Postmaster, San Francisco, Calif.
Packham, E. Thornton ('37; '38) (MCJ), Planning Engr., West. Elec. Co., Inc., Baltimore; *for mail,* 5316 Plymouth Rd., Baltimore 14, Md.
Packman, Ian Buchanan ('44) (CLDFSO), Engr., Borden Cheese Co., 350 Madison Ave., New York 17, N.Y.
Paddock, Charles B. ('21), Mgr., Hartford Steam Boiler Inspc. & Ins. Co., 144 Sansome St., San Francisco; *for mail,* 424 Terrace Way, San Mateo, Calif.
Paddock, Russell Gurney ('21; '25; '35) (SFE), Prof. & Head, Dept. of Mech. Engrg., Univ. of Ark.; *for mail,* 1400 Cleveland Ave., Fayetteville, Ark.
Paddock, Stephen Gorham (J'43) (EKA), Indus. Engr., Burgess-Manning Co., 2815 W. Roscoe St., Chicago 18; *for mail,* 814 Center Ave., Arlington Heights, Ill.
Paech, Erik G. ('30; '35) (CKLMNZ), Mgr. of Mfg., Braun Corp., 2260 E. 15th St., Los Angeles 21, Calif.
Paetz, George Albert (J'40) (CMAJ), Lt. (j.g.), O-(V)S, U.S.N.R., 1721 Preston Rd., Alexandria, Va.
Pafenbach, Charles Henry ('43) (ND), Tool Div., Willys-Overland Motors, Inc., Walcott Blvd., Toledo; *for mail,* 4224 N. Haven St., Toledo 12, Ohio.
Paffen, Paul J. ('16) (EFS), Cons. Indus. Engr., 331 W. Water St.; *for mail,* Hotel York, New Haven, Conn.
Paffenbarger, Ralph Seal ('45), Prof. & Chmn., Dept. of Engrg. Drawing, Rm. 218, Brown Hall, Ohio State Univ., Columbus 10, Ohio.
Page, Cazwell J., Jr. (J'46) (CJM), 210 Fordyce St., Pittsburgh, Pa.
Page, Harold Dean ('41) (RYBJF), V.P., Waugh Equip. Co., 420 Lexington Ave., New York 17, N.Y.
Page, John Fisher (J'43), Mech. Engr., Apparatus Design & Specifications, Fed. Tel. & Radio Corp., 591 Broad St., Newark; *for mail,* 15 Cathedral Ave., Nutley, N.J.
Page, Kingman W. (J'40) (BE), Test Engr., Gen. Elec. Co., River Works, West Lynn, Mass.; *for mail,* 135 S. Union St., Olean, N.Y.
Page, Raymond Leon (J'42) (NBCYA), Ch. Designer, Hydrodynamics Lab., Calif. Inst. of Tech., 1201 E. California, Pasadena; *for mail,* Apt. B, 2014 Meridian, South Pasadena, Calif.
Page, Schuyler C. ('30; '35) (ELSFYK), 349 Richmond Ave., Buffalo 13, N.Y.
Page, Stanley Erskine (J'42), 50 Follen St., Cambridge, Mass.
Page, Stanley H. ('29) (EAZ), Owner, Dynamometer Lab., Box 404, Los Gatos, Calif.
Pahmeyer, Fred O. ('23) (CMSDF), Works Mgr., Heine Boiler Div., Combustion Engrg. Co., Inc., 5319 Shreve Ave., St. Louis 15, Mo.
Pailin, George ('29), Dir., Ferguson, Pailin Ltd., Higher Openshaw, Manchester 11, England.
Paine, Arthur Preston ('20; '35) (GTCMN), Dir. of Pats., Mergenthaler Linotype Co., 29 Ryerson St., Brooklyn 5; *for mail,* 160 E. 48th St., New York 17, N.Y.
Paine, Walter Sayer ('18; '21) (BCMY), Mgr., Engrg. & Inspc. Dept., Aetna Casualty & Surety Co., 151 Farmington Ave., Hartford 15, Conn.
Pais, Wilson John (J'37) (SEFA), Combustion Engr., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.; *for mail,* Box 191, Pocahontas, Va.
Palchik, Edward H. (J'37) (BKS), Asst. Mar. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 63-109 Saunders St., Forest Hills, N.Y.
Palladino, Nunzio J. (J'38) (BRS), Design Engr., Westinghouse Elec. Corp.; *for mail,* 354 Clearbrook Ave., Drexel Plaza, Lansdowne, Pa.
Palley, Julius Arnold (J'46), 17 Havelock Rd., Worcester, Mass.
Palm, Axel Edgar ('42) (FS), Ch. Engr., Internatl. Smelting & Refining Co., Elm St., Perth Amboy; *for mail,* 25 Lakeside Dr., Matawan, N.J.
Palm, Bernhard Nelson ('38; '42) (BCM), Ch. Mech. Engr., Sterling Elec. Motors, Inc., 5401 Telegraph Rd., Los Angeles 22; *for mail,* 2570 Lorain Rd., San Marino 9, Calif.
Palm, Edgar Holt (J'44), 1400 E. 53rd St., Chicago, Ill.
Palmatier, Everett Perry ('45) (ABK), Admin. Engr., Carrier Corp., Syracuse; *for mail,* 106 Wynthrop Rd., Solvay, N.Y.
Palme, Richard Baumann (J'44) (NCYJHK), Asst. Stands. Engr., Gleason Works, 1000 University Ave.; *for mail,* Y.M.C.A., 100 Gibbs St., Rochester, N.Y.
Palmer, Albert ('45) (CTMNB), V.P., Dir. of Engrg., Crompton & Knowles Loom Works, 93 Grand St., Worcester 1, Mass.
Palmer, Charles Sumner, Jr. (J'40) (CMALDR), 1232 N. Lake Ave., Pasadena 6, Calif.
Palmer, Delos Marquis ('31; '35) (KNOSYZ), Cons. Engr., 783 Nicholas Bldg., Toledo 4; *for mail,* 4112 Douglas Rd., Toledo 12, Ohio.
Palmer, Elbridge Woodman (A'27) (BCDGMY), Pres., Kingsport Press, Inc., Reedy St., Kingsport, Tenn.
Palmer, Henry O. ('20) (CS), Contract Renegotiator, Army Air Forces, 67 Broad St., New York 4; *for mail,* 11 Waverly Pl., New York 3, N.Y.
Palmer, John Hambleton (J'35) (ZMKJCN), Asst. Engr., Advance Engrg. Sec., A. & M. Dept., Gen. Elec. Co., 1285 Boston Ave., Bridgeport 2; *for mail,* 50 Wilbar Ave., Milford, Conn.
Palmer, Lubin, Jr. (J'38) (MCJD), V.P., Treas., Engr., J. P. Glasby Mfg. Co., Inc., 1 Montgomery St., Belleville 9; *for mail,* 2 Club Rd., Upper Montclair, N.J.
Palmer, Paul Augustus (J'44) (ABK), Engrg. Draftsman, Douglas Aircraft Co., Inc., Santa Monica; *for mail,* 2062 Sawtelle Blvd., Los Angeles 25, Calif.
Palmer, Ralph Mallory ('33) (SHJKLM), Pres., Ferro-Nil Corp., 381—4th Ave., New York 16, N.Y.
Palmer, Shepard Brown, Jr. (J'34) (OH), Partner, Chandler & Palmer, 114-16 Thayer Bldg., Norwich, Conn.
Palmer, Virgil M., Jr. (J'43) (MCA), Indus. Engr., Crucible Steel Co. of Am., Harrison; *for mail,* 8 Enclosure, Nutley 10, N.J.
Palmer, William Cecil ('22; '35) (ABJMNS), Mgr., Mfg. & Repair Dept., Westinghouse Elec. Corp., 5757 Trumbull Ave., Detroit 8, Mich.
Palmer, William John Douglas (J'28) (FZEKCJ), Asst. Research Officer, Australian Gas Light Co., 477-487 Pitt St., Sydney, N.S.W., Australia.
Palmgren, Stig George (J'42) (MCLJD), Prod. Engr., E. I. du Pont de Nemours & Co., Wilmington 55; *for mail,* 410 Claymont Garden Apts., Claymont, Del.
Palsgrove, Grant Knauer ('18; '21) (HSBRZ), Prof. Hyd. & Mech. Engrg., Rensselaer Poly. Inst.; *for mail,* 1514 Sage Ave., Troy, N.Y.
Palumbo, Mario Domenic (J'41) (CJM), Ensign, O-V(S), U.S.N.R., A.G.O. Sch., Naval Aviation Tech. Trng. Center, Jacksonville, Fla.; *for mail,* 2 Granite St., Waterbury 34, Conn.
Pamphilon, G. Maxwell ('36; '41) (BCM), Ch. Engr., Johnson Gear & Mfg. Co., Ltd., 8th & Parker, Berkeley 2; *for mail,* 207 Colgate Ave., Berkeley 8, Calif.
Panabaker, David Deane (J'40) (CMJND), Mgmt. Engr., Stevenson & Kellogg Ltd., Can. Permanent Bldg., Toronto; *for mail,* 33 Sutherland Dr., Leaside, Ont., Can.
Panagos, Peter (J'43) (HRS), 3 Palmer Sq., Princeton, N.J.
Panak, Leon P. (J'36) (CLM), 1964 Grand Concourse, New York, N.Y.
Panczner, Erwin Herman (A'44) (CHL), Div. Mgr., P. R. Mallory & Co., Inc., 3029 E. Washington, Indianapolis 6, Ind.
Panitz, Adolf (J'39) (CMO), Instr., East N.Y. Vocational High Sch., Founten & Welles Ave., Brooklyn; *for mail,* 83-46—118th St., Kew Gardens 15, N.Y.
Panitz, Karl A. ('32; '46) (CBMNDJ), Supt., Arthur Colton Co., 2600 Jefferson E., Detroit 7; *for mail,* 10107 Grayton, Detroit 24, Mich.
Pankey, Thomas Lorton ('42) (SZL), Sales Engr., Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Pansky, Emil John (J'42) (ACM), 162-12 Horace Harding Blvd., Flushing, N.Y.
Panuska, Frank C. ('14; '21; '26), Admin. Asst., Head, Mech. Drawing Dept., Stuyvesant High Sch., 345 E. 15th St., New York; *for mail,* 3346—98th St., Corona, L.I., N.Y.
Pape, Paul F. ('23; '35), 30 Corell Rd., Scarsdale, N.Y.
Papenfuss, Charles August ('30; '31; '35) (CS), Ch. Engr., Fed. Reserve Bank of N.Y., 33 Liberty St., New York 8, N.Y.
Pappas, James Nicholas (J'45) (NDOGMC), Mech. Engr., Frank Tracy Inc., 202 Centre St., New York 13, N.Y.
Paque, E. J. ('20; '21) (CNMLJF), Works Mgr., Ridgewood Steel Co., Box 5, Elmwood, Cincinnati; *for mail,* 3303 Beredith Pl., Cincinnati 13, Ohio.
Paradiso, Sam (J'40) (CS), U.S.N.R., 3050 R St., Lincoln 3, Neb.
Paraso, Benefico F. ('45) (SFHMNK), Mech. Engr., P-4, U.S.N., Pub. Works Design Sec., Pearl Harbor, T.H.
Parce, Joseph Y. (J'25) (SFECK), Engr., Power Dept., Stearns-Roger Mfg. Co., 1720 California St., Denver 2; *for mail,* 2705 Tennyson St., Denver 12, Colo.
Parcels, Charles Franklin ('44) (CFKS), Supt. Power, Del. Power & Light Co., 600 Market St.; *for mail,* Gen. Delivery, Main P.O., Wilmington, Del.
Pardo, Vincent Anthony (J'42) (FHZ), Pvt., 1st Class, Corps of Engrs., U.S.A., Chief, Instrument Research Lab., Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.; *for mail,* 910 Montgomery Ave., Narberth, Pa.
Parent, Donald F. (J'42) (SKEBJ), Contract Sec., Turbine Engrg. Dept., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 232 Jackson Ave., Schenectady 7, N.Y.
Parish, Joseph Mack, Jr. ('45), Tech. Expert, Dow Chem. Co.; *for mail,* 1615 W. 4th St., Freeport, Tex.
Park, Frank Carleton ('45), Supvr. of Engrg., Safe Cabinet Div., Remington Rand, Inc.; *for mail,* 608 Washington St., Marietta, Ohio.
Park, Phocion Samuel (J'43) (BASJ), Ensign, U.S.N.R., 1405 Hartford Rd., Austin 21, Tex.
Parke, Peter ('14), Ch. Engr., Design & Constr., Pullman Co., 79 E. Adams St.; *for mail,* 1725 E. 53rd St., Chicago, Ill.
Parken, Edward Arthur (J'35) (JLM), Mech. Engr., Natl. Carbon Co., Niagara Falls, N.Y.
Parker, Arthur Rowley ('17; '35) (SFKBED), Mech. Engr., Gen. Engrg. Dept., Cincinnati Gas & Elec. Co., 323 Plum St., Cincinnati 2, Ohio.
Parker, Austin Wilbur ('92; '01) (EGH), Retired; 262 Sylvan St., Rutherford, N.J.
Parker, Charles George (J'42) (SKHOF), Cons. & Application Engr., Steam Div., Westinghouse Elec. Corp., 10 High St., Boston 10, Mass.
Parker, Clarence Colcord (J'41) (MCZJD), Asst. Mech. Engr., Army Serv. Forces, Birmingham Ord. Dist., 700 Frank Nelson Bldg., Birmingham 1; *for mail,* 1717—11th Ave. S., Birmingham, Ala.
Parker, Darrell Smith ('44) (LCSDMA), Valuation Engr., Bur. of Internal Revenue, U.S. Treasury Dept., Internal Revenue Bldg., Constitution Ave., Washington 25, D.C.; *for mail,* Park Fairfax, 3201 Valley Dr., Alexandria, Va.
Parker, E. B. ('34) (JLMWYN), Acting Dir. of Indus. Research, State College of Wash.; *for mail,* 207 College Ave., Pullman, Wash.
Parker, E. H. (J'39), Engr., Okla. Natural Gas Co., Tulsa, Okla.
Parker, Earl William Edward (J'46), 27 Cedar St., Hamilton, Ont., Can.
Parker, Edwin Ellison ('35; '42) (SBKE), Turbine Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.
Parker, Francis A. ('29; '35) (ABCMN), Partner, Firm of Frank A. Parker, 19 Oakland Ave., Auburndale 66, Mass.
Paker, Francis Warner, III (J'38) (NMDC), Engr., Designer, Kimberly-Clark Corp.; *for mail,* 510 Oak St., Neenah, Wis.
Parker, Frederick Leon (J'40) (AKEF), Installation Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St.; *for mail,* 38 Porter St., East Hartford 8, Conn.
Parker, George Adelbert ('27) (SEFHKB), Asst. Mech. Engr., Control Power Sta., Utah Copper Co., Kearns Bldg., Salt Lake City 1; *for mail,* 422 Coatsville Ave., Salt Lake City 4, Utah.
Parker, George Clifford ('38) (BJM), Ch. Estimator, Taft-Peirce Mfg. Co., Mechanics Ave.; *for mail,* Box 1261, Woonsocket, R.I.
Parker, George H. (J'41) (BC), Asst. Maint. Engr., Container Corp. of Am., 1301 W. 35th St.; *for mail,* 4228 Washington Blvd., Chicago, Ill.
Parker, Graham Woodward ('43) (MDC), Partner, Parker & Strackbein, 230 Park Ave., New York 17, N.Y.; *for mail,* Comstock Hill, Norwalk, Conn.
Parker, H. Sterling ('22; '27; '35), Supvr. of Bldgs., Harry Thoens & Co., Inc., 366—5th Ave., New York; *for mail,* 295 Pennsylvania Ave., Crestwood, N.Y.
Parker, Harry M. ('30; '35) (ACJM), Exec. Engr., Pur. Agt., Wayne Mfg. Corp.; *for mail,* Box 676, Waynesboro, Va.
Parker, Henry William, Jr. (J'38) (BJYH), Mech. Engr., U.S. Naval Base; *for mail,* 429 Richards Ave., Portsmouth, N.H.
Parker, Humphrey Francis ('30) (NDJ), Ch. Engr., Columbus McKinnon Chain Corp., Tonawanda; *for mail,* 778 Lafayette Ave., Buffalo 9, N.Y.
Parker, James W. ('13; '25) (SCKFB), *Manager,* '35-'38, *Vice-President,* '38-'40, *President,* '42; Pres., Gen. Mgr., Dir., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Parker, John (J'45) (HKS), Design Draftsman, Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* 234 Sharon Ave., Collingdale, Pa.
Parker, John ('21) (JBC), Sales Engr., F. H. Crawford & Co., Inc., 30 Church St., New York 7, N.Y.; *for mail,* 212 William St., East Orange, N.J.
Parker, John C., III (J'43) (ACMJ), 85 Remsen St., Brooklyn, N.Y.
Parker, John Castlereagh ('05; '09; F'41) (FKS), V.P., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Parker, John Clinton ('05), Pres., Lefax, Inc., 9th & Sansom Sts., Philadelphia, Pa.

Parker, John Wesley, Jr. (J'32) (KN), Sales Engr., Leo S. Bosarge Co., 315 Spring St., N.W., Atlanta 3; *for mail*, 273 Rumson Rd., Atlanta, Ga.
Parker, John William ('41) (ZMB), Mech. Engr., Brown & Sharpe Mfg. Co., Promenade St., Providence; *for mail*, Box 126, West Barrington, R.I.
Parker, Karr ('18; '20; '24) (BCH), Pres., Buffalo Elec. Co., 75-81 W. Mohawk St., Buffalo 2, N.Y.
Parker, Leland L. (J'43) (BNSWEA), Lt., Ord. Dept., U.S.A., Product, Design & Devel. Officer, Research & Devel. Serv. Sub-Office, Aberdeen Proving Ground, Md.; *for mail*, 3828—2nd St., S.E., Washington 20, D.C.
Parker, Luther Milton ('36; '45) (SKC), Asst. Serv. Mgr., Worthington Pump & Mchy. Corp., Coats St.; *for mail*, 9 W. State St., Wellsville, N.Y.
Parker, McRea ('22; '35) (SLF), 2214 Delamere Dr., Cleveland Heights, Cleveland, Ohio.
Parker, Milton Ellsworth ('44) (CLMK), Counselor in Food Prod. Devel., 308 W. Washington St., Chicago 6, Ill.
Parker, Norman Alwyn ('41) (EASF), Prof. & Head, Dept. of Aero. & Mech. Engrg., Univ. of Colo., Boulder, Colo.
Parker, R. Starr ('32; '38) (ELS), H. H. Meyer Packing Co., 2115 Linn St., Cincinnati 14, Ohio.
Parker, Raymond Easton (J'42), Student Engr., Testing Dept., Gen. Elec. Co., River Works, West Lynn; *for mail*, 12 Rowell Ave., East Lynn, Mass.
Parker, Rey L. ('22; '29) (CM), Comdr., U.S.N.R., c/o Wilbur C. Parker, 45 Rolph St., San Francisco, Calif.
Parker, Richard Ernest Barry (J'45) (KSREFZ), 491 E. Sanger St., Philadelphia 20, Pa.
Parker, Robert Beirne ('45) (TNMOBD), Owner, Parker Engrg. Co., 54 Custom House, Providence 3; *for mail*, 76 Ferncrest Ave., Edgewood 5, R.I.
Parker, Selsor S. ('41) (YHAEK), Ch. Draftsman, Ashland Oil & Refining Co.; *for mail*, 2801 Main St., Ashland, Ky.
Parker, Warde L. (J'40), Engr., Wagner Morehouse, Inc., 2371 E. 51st St.; *for mail*, 1915 W. 108th St., Los Angeles, Calif.
Parker, William Elbridge, Jr. (J'41) (MNCB), Designer, Warren City Mfg. Co., Griswold St.; *for mail*, 978 East Ave., S.E., Warren, Ohio.
Parker, William T., Jr. (J'39) (ADC), Lt. Comdr., U.S.N.R., 66 Rivermont Dr., Hilton Village, Va.
Parkhurst, E. Raymond ('44), V.P., Sales Mgr., Rensselaer Valve Co.; *for mail*, 143—2nd Ave., Troy, N.Y.
Parkin, John Hamilton ('19; '24) (ABHFY), Dir., Div. of Mech. Engrg., Natl. Research Council of Can., Ottawa, Ont., Can.
Parkin, Robert Edward (J'44) (ZLBMJY), Mech. Engr., Design, Taylor Instrument Cos., 95 Ames St., Rochester; *for mail*, 27 Woodstock Rd., Rochester 9, N.Y.
Parkinson, Russell William ('26; '38) (SFKOHA), Asst. Ch. Mech. Engr., Commonwealth & So. Corp., 212 W. Michigan Ave., Jackson, Mich.
Parks, George Urbane ('30) (SFCJKE), Gen. Mgr., Montaup Elec. Co., Box 391, Fall River, Mass.
Parks, Henry Schumacher (J'43) (BOS), Constr. Supt., with Slack Winburn, Arch., 618 Beason Bldg., Salt Lake City 1; *for mail*, 1431 E. 21st St., South Salt Lake City 5, Utah.
Parks, Joseph Aloysius, Jr. ('41) (CJM), Columbia Mch. Works, Inc., 255 Chestnut St., Brooklyn 8; *for mail*, 347 Madison Ave., New York, N.Y.
Parks, Paul Harold (J'43) (AEF), Engr., Exper. Test Dept., Continental Motors Corp., Muskegon; *for mail*, 315 Grand Ave., Grand Haven, Mich.
Parks, Wilbur H. (J'42) (BMNHK), Asst. Prof. Mech. Engrg., Univ. of Denver, Denver 10, Colo.
Parlett, Raymond Coleman ('21; '25; '35) (SKFEL), Mech. Engr., Gen. Hdq., Johns-Manville Corp., 22 E. 40th St., New York 16, N.Y.
Parlette, Harry Leslie, Jr. ('39; '41) (SCLEOD), Asst. to V.P., L. I. Lighting Co., 250 Old Country Rd., Mineola, N.Y.
Parlini, Alexander C. ('41) (AMNZ), Works Mgr., Partner, Empire State Labs., 161 Maiden Lane, New York 7, N.Y.
Parlon, William L. (J'40) (ACL), Ensign, U.S.N.R., 321 Sunset Court, Rochester 10, N.Y.
Parma, Edward James (J'41) (JLC), 414 Nelson Ave., Cliffside Park 10, N.J.
Parmely, J. Clyde ('14; '21) (SFEDCL), Matl. Control, Midwest Piping & Supply Co., 1450 S. 2nd St., St. Louis 4, Mo.
Parmley, Seba M. ('20) (DFJ), Asst. Ch. Engr., Pittsburgh Coal Co., Oliver Bldg., Pittsburgh; *for mail*, 210 Castle Shannon Rd., Pittsburgh 16, Pa.
Parr, Garner C. ('43) (SZFLTC), Mgr., Sales Engr., Ash Engrg. Co., 225 S. 15th St., Philadelphia 2, Pa.
Parr, Harry L. ('10) (HBASZ), Stevens Prof. Mech. Engrg., Columbia Univ., New York 27, N.Y.
Parris, Gordon Charles (J'44) (KAFSBH), Flying Officer, Special Engrg. Duties, Tech. Branch, Royal Air Force, c/o Rolls Royce Ltd., Clitheroe, Lancaster; *for mail*, 20 Alhambra Rd., Southsea, Hampshire, England.
Parris, Joseph N. (J'43) (BNZH), 1210 Voscamp St., Pittsburgh 12, Pa.
Parrish, Donald Jardine (J'43), Lecturer, Univ. of Toronto; *for mail*, Apt. 403, 26 St. Joseph St., Toronto, Ont., Can.
Parrish, Joseph R. (J'38) (SHKFEJ), Mech. Engr., Tenn. Valley Authority, 205 Union Bldg., Knoxville 15, Tenn.
Parrish, Vernon MacLeod (J'38) (SFZELK), Sales Serv. Engr., Bailey Meter Co. Ltd., 906 McArthur Bldg., Winnipeg, Man., Can.
Parrish, William Clarence (J'45) (EKAB), Devel. Engr., Gas Turbines, Airesearch Mfg. Co., 9851 Sepulveda Blvd., Los Angeles 45; *for mail*, 214 W. Ellis Ave., Inglewood, Calif.
Parsell, Roy L. ('27) (CMNKB), Pat. Atty., Winchester Repeating Arms Co., New Haven 4, Conn.
Parsons, Charles Sumner ('45) (TCB), Supt., Hathaway Mfg. Co., 97 Cove St., New Bedford; *for mail*, 10 Lafayette St., Fairlawn, Mass.
Parsons, Charles Warren ('30; '35) (ZSF), Asst. to V.P., Republic Flow Meters Co., 2240 Diversey Pkwy., Chicago 47, Ill.
Parsons, Fred'k A. ('14; '21), 1506 Mariner Tower, Milwaukee, Wis.
Parsons, G. Chauncey ('43) (ONC), Cons. Engr., 327 W. 7th St.; *for mail*, Box 454, San Pedro, Calif.
Parson, George K. ('11) (CMYDLJ), Cons. Mgmt. Engr., 158 Alta Ave., Yonkers 5, N.Y.
Parsons, Harold Kent (A'42) (NMC), Res. Atty. & Exec., Cincinnati Milling Mch. Co., 4701 Marburg St.; *for mail*, Box 83, Oakley, Cincinnati 9, Ohio.
Parsons, Harry N. ('12; '19; '24) (BNJMEC), Ch. Engr., Internatl. Harvester Co., 1015 W. 120th St., Chicago 43, Ill.
Parsons, Herbert Leland (J'41) (CEJMNA), Asst. Project Engr., Chambers Works, E. I. du Pont de Nemours & Co., Deepwater, N. J.; *for mail*, 208 W. 21st St., Wilmington, Del.
Parsons, James Lowell (J'40) (LKO), Indus. Engr., Research Devel., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail*, 5 Ave. I Ext., Carneys Point, Penns Grove, N.J.
Parsons, L. D., Jr. (J'36) (FKS), Passman Equip. Co., 114-20 Walnut St.; *for mail*, 304 McGuire St., Monroe, La.
Parsons, Richard Lewis (J'40) (HKB), Petroleum Engr., Stand. Oil Co. of Calif., Box 397, La Habra, Calif.
Parsons, Winchell M. (J'36) (KL), 1502½ Bank St., South Pasadena, Calif.
Parthesius, H. J. ('15; '25) (BFJKNS), 122 Hillcrest Ave., Manhasset, L.I., N.Y.
Partington, James ('17) (ERS), Mgr., Engrg. Dept., Am. Loco. Co., 30 Church St., New York, N.Y.
Partlow, James Edward (J'42) (JHB), Lt., U.S.N.R.; 9045 S. Carpenter St., Chicago 20, Ill.
Partridge, Harry Ewart ('25) (DEFKST), Sr. Partner, Partridge, Earp & Partners, 7 Coates Crescent, Edinburgh 3; *for mail*, 10 Gillespie Rd., Colinton, Edinburgh 13, Scotland.
Partridge, Robert Green (J'44), Lt. (j.g.), U.S.N.R., U.S.S. *Fox*, AG-85, F.P.O., San Francisco, Calif.
Parzick, Henry Anthony (J'43) (SKCEJF), Steam Serv. Engr., Westinghouse Elec. Corp., 420 S. San Pedro, Los Angeles 13; *for mail*, 3135 E. Ocean Ave., Long Beach 3, Calif.
Pascaloff, Carl John (J'44) (JMYFNR), Lt. (j.g.), U.S.N.R., U.S.S. *Dortch*, DD-670, F.P.O., New York, N.Y.; *for mail*, 4608 Los Feliz Blvd., Los Angeles 27, Calif.
Paschall, A. L. ('23) (GNEWC), Mem. Engrg. Staff, Charge Publications, Hobart Mfg. Co., Pennsylvania Ave.; *for mail*, 606 Ridge Ave., Troy, Ohio.
Pascual, Santiago J. ('44) (SLF), Buell Engrg. Co., Inc., 70 Pine St., New York 5; *for mail*, 864—67th St., Brooklyn 20, N.Y.
Pasick, Julian Morgan (J'41) (BCKMRS), Mech. Engr., Erie Works, Gen. Elec. Co., E. Lake Rd., Erie; *for mail*, Godfrey Rd., North Girard, Pa.
Pasini, Albert Carl ('40) (BCFS), Tech. Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Passano, William Moore ('27; '35) (CG), Treas., Gen. Mgr., Waverly Press, Inc., Mt. Royal & Guilford Aves., Baltimore 2, Md.
Pastor, John Cope ('43) (KSEFZ), 818 Graham Bldg., Jacksonville 2, Fla.
Pastoriza, Hugh Girard ('22) (ACHT), Engr., Coffin & Burr, Inc., Rm. 5107, 70 Pine St., New York 5, N.Y.
Patch, Alfred Edward (J'38) (RASCEB), Lt., Civil Engr. Corps (S-1), U.S.N.R., 145 State St., Windsor, Vt.
Patch, Earl S. ('23) (BJM), Sales Mgr., Micro-Ferrocart Products Div., Maguire Industries, Inc., 375 Fairfield Ave., Stamford, Conn.; *for mail*, Ridgeway Rd., Dayton 9, Ohio.
Patchen, Mortimer S. (J'43) (CMD), V.P., Emco Porcelain Enamel Co. Inc., Box 53, Port Chester, N.Y.
Pate, Samuel Gordon, Jr. (J'46), 807 Vedado Way, N.E., Atlanta, Ga.
Patek, Sherwood Eddy (J'46), Apt. 47, 230 W. 76th St., New York 23, N.Y.
Paternoster, Joseph A. ('39) (D), 1820 S. 4th St., Alhambra, Calif.
Paterson, A. B. ('20) (CES), Pres., New Orleans Pub. Serv. Inc., 317 Baronne St., New Orleans 9, La.
Paterson, Albert Brown, Jr. (J'38) (CSE), Indus. Engr., La. Power & Light Co., 711—2nd St., Gretna; *for mail*, 1920 Bodenger Blvd., New Orleans 14, La.
Paterson, James V. ('12), 1025 Boylston Ave. N., Seattle, Wash.
Paterson, L. B. ('15; '21; '25) (JMS), Pur. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York; *for mail*, 79 Maple Ave., Tuckahoe, N.Y.
Pathak, Mukand L. ('25) (ACM), Bungalow 283/C, Khargpur, Bengal Napur Ry., Bengal, India.
Patitz, Gerhardt John ('00; '07) (CDKLOS), Cons. Engr., Stand. Brands, Inc., 595 Madison Ave., New York 22; *for mail*, 1601 Maple Ave., Peekskill, N.Y.
Patitz, Gerhardt Norman (J'36) (CLMDSO), Plant Mgr., Fleischmann Distilling Corp., Ewing Rd.; *for mail*, 2124 Locust St., Owensboro, Ky.
Paton, Alfred Hugh ('44) (AEH), Asst. Ch. Engr., Diesel Div., Dominion Engrg. Works Ltd., Box 220, Montreal, Que., Can.
Paton, William John Ross (J'43) (CMNJBE), Asst. to Gen. Mgr., Weatherhead Co. of Can., Ltd., Inkerman St., St. Thomas, Ont., Can.
Patrick, Merlyn Joseph ('44) (NMACYH), Ch. Engr., Supt., Allen Gauge & Tool Co., 421 N. Braddock Ave., Pittsburgh 21; *for mail*, 527 S. Graham St., Pittsburgh 6, Pa.
Patscheider, Walter ('26) (MJDCNO), Works Engr., Walworth Co., 798—1st St., South Boston 27, Mass.
Patt, I. Fred (J'35), Peerless Prod. Co., 16941 Hamilton Ave., Highland Park 3; *for mail*, 17409 Roselawn Ave., Detroit, Mich.
Patten, Charles Anthony (J'42) (CMDOLR), Mech. Engr., Dravo Corp., Neville Island, Pittsburgh 25; *for mail*, 915 Fleming St., Coraopolis, Pa.
Pattershall, Donald Smith (J'40) (NYSKB), Maint. Foreman, E. I. du Pont de Nemours & Co., Arlington; *for mail*, 44 Beech St., North Arlington, N.J.
Patterson, Anthony Russell (J'45) (LYMDCA), Project Engr., Armstrong Cork Co.; *for mail*, 150 E. Chestnut St., Lancaster, Pa.
Patterson, Charles Robley ('44; '44) (NKHMLA), Mech. Engr., Design, El Segundo Refinery, Stand. Oil Co. of Calif., El Segundo; *for mail*, 458—19th St., Santa Monica, Calif.
Patterson, Clare Bushey ('22; '35) (BCHKS), Power Engr., Hercules Powder Co.; *for mail*, 611 Corrine St., Hattiesburg, Miss.
Patterson, David Wills ('23) (FKEZ), Engr., Universal Oil Products Co., 310 S. Michigan Ave., Chicago 4, Ill.
Patterson, George A., Jr. (J'46) (EFH), Engr., Boeing Aircraft Co., Marginal Way, Seattle; *for mail*, 3865 E. 155th St., Seattle 55, Wash.
Patterson, Harry Rickert ('44) (DLN), Devel. Engr., Drackett Co., R.F.D. 1, Sharonville; *for mail*, Box 591, R.F.D. 6, Cincinnati 15, Ohio.
Patterson, James Leslie (J'39) (CDFKMS), Plant Supt., A. P. Green Fire Brick Co. Ltd., Commercial St., Toronto 12; *for mail*, 1747 Jane St., Weston, Toronto 15, Ont., Can.
Patterson, John Lee (J'41) (OMSWCB), Ensign, U.S.N.R.; 496 Montauk Ave., New London, Conn.
Patterson, Lawrence S. ('19; '35) (CGM), Prin. Statistician, N.Y. State Pub. Serv. Comm., 80 Centre St., New York; *for mail*, 83 S. Village Ave., Rockville Centre, L.I., N.Y.
Patterson, Lorne Aldon (J'40) (NCSYBK), Plant Engr., Fed. Wire & Cable Co. Ltd., 265 Suffolk St., Guelph, Ont., Can.
Patterson, Mark M. (J'41) (SJFEBK), 617 Union St., Schenectady, N.Y.
Patterson, Morehead ('45), Chmn., Am. Mch. & Fdy. Co., 511—5th Ave., New York 17; *for mail*, 15 E. 65th St., New York, N.Y.
Patterson, Peter C. ('97; '16), Retired; Boston, Pa.
Patterson, Robert C. (J'45) (SKFZR), Devel. & Research Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Patterson, Robert Olson (J'37) (KLS), Field Mech. Engr., Powers Regulator Co., 2720 Greenview Ave., Chicago 14; *for mail*, 1668 Farwell Ave., Chicago 26, Ill.
Patterson, Roy Kendall ('43) (SHKACM), Mar. Rep., Johns-Manville Sales Corp., 22 E. 40th St., New York 16; *for mail*, 38 Graham Rd., Scarsdale, N.Y.
Patterson, Stanley ('40), Supt., Bldgs. & Grounds, So. Methodist Univ., Dallas 5, Tex.
Patterson, Truett Herbert (J'42) (AJFNSE), Test Engr., Allison Div., Gen. Motors Corp., Indianapolis; *for mail*, 641 W. 30th St., Indianapolis 8, Ind.
Patterson, Ward S. ('26; '37) (KSFBEH), Asst. Mgr., Forced Circulation Boilers, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Pattison, Floyd ('19; '35), Home Study Serv., Kan. State College, Manhattan, Kan.
Pattison, Robert Cuttler ('20) (BJRS), Mech. Engr., Loco. & Car Design, Wheeling & Lake Erie Ry. Co.; *for mail*, 138—2nd St., S.W., Brewster, Ohio.

Pattison, Ronald (J'40) (ECONL), Asst. Engr., Messrs. Walker & Greig Ltd., Box 165, Colombo, Ceylon.
Pattiz, Gene Burton (J'44) (AMC), U.S. Army., 6030 McPherson Ave., St. Louis 12, Mo.
Patton, Edward Russell ('44), Tech. Serv. Dept., Devel. & Research Div., Internatl. Nickel Co., Inc., 67 Wall St., New York 5, N.Y.
Patton, Fred Talbot ('46) (BJP), Engr., Mfg. Dept., Stand. Oil Co. of Calif., 225 Bush St., San Francisco; *for mail,* 300 Yale Ave., Berkeley 8, Calif.
Patton, James Ross, Jr. (J'40) (ABEFHK), Lt. Comdr., Research & Devel. Engr., U.S.N., Pilotless Aircraft Unit, Mar. Corps Air Sta., Mojave, Calif.
Patton, John D. (J'41) (YBCLMD), 3710 Bell Ave., Houston, Tex.
Patton, John Warren (J'43), Piping Engr., Austin Co., Houston; *for mail,* 833-B Parker St., Amarillo, Tex.
Paugh, Charles Thomas ('18; '20; '25; F'45) (SFZKOB), Utilities Engr., U.S. Indus. Chemicals, Inc., Box 1797, Baltimore 3, Md.
Paugh, G. Ruel ('23) (SFEJKB), Sales Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* 325 Maolis Ave., Glen Ridge, N.J.
Paukner, Frank Joseph (J'29) (JBNML), Die Designer, E. W. Bliss Co., 53rd St. & 2nd Ave., Brooklyn; *for mail,* 86-61—104th St., Richmond Hill 18, N.Y.
Paul, Ellis E. ('38) (AYO), Maj., Army Air Forces, Wright Field, Dayton; *for mail,* Springfield Pike, Yellow Springs, Ohio.
Paul, Ralph C. ('25; '35) (ECSR), Asst. to Mgr., Diesel Eng. Div., Am. Loco. Co., 100 Orchard St.; *for mail,* 105 Mary St., Auburn, N.Y.
Paul, Robert Fulton ('41) (SBF), Mech. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5; *for mail,* 514 Berkley Rd., Narberth, Pa.
Paul, Stephen Bartlett (J'43) (EMR), Prod. Engr., Am. Loco. Co., Schenectady, N.Y.
Paull, William R. ('42) (BCH), Partner, Salinas Valley Pump Serv., Box 157, Salinas, Calif.
Paulsen, Alfred G. ('23) (CES), Assoc. Oper. Sponsor, Internatl. Div., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Paulsen, William Nahmen (J'44), Lt. (j.g.), U.S.N.; 1092 Aileen St., Oakland 8, Calif.
Paulson, Elmir Edward (J'41) (ABHKJC), Engr., Gen. Elec. Co., 920 Western Ave., West Lynn 3; *for mail,* 86 Pleasant St., Marblehead, Mass.
Pauly, Bruce Henry (J'46) (A), Design Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester; *for mail,* 218 W. Martin Lane, Norwood, Pa.
Pausin, Hugo Richard ('16; '17) (BJM), Partner, Gen. Mgr., Pausin Mfg. Co., 727-39 Frelinghuysen Ave., Newark 5, N.J.
Pautler, Anthony C. (J'41) (DJL), Ensign, U.S.N.R., 403 S. Oak St., Union, Mo.
Pava, Norman (J'44) (NMAWGC), Ch. Engr., Partner, Freemerman-Pava Mfg. Co., 380—3rd Ave., New York 16; *for mail,* 65-36 Wetherole St., Forest Hills, L.I., N.Y.
Pavelle, Leo ('44), Pres., Pavelle Labs. Inc., 16 E. 42nd St., New York, N.Y.
Pavlik, William Joseph (M'45) (KSOECH), Supt., Northport Elec. Generating Plant, Long Island Ltg. Co., Woodbine Ave.; *for mail,* 42 Oak St., Northport, N.Y.
Pawelczyk, Joseph Anthony (J'45) (CLO), Sales Dept., Gen. Elec. Supply Co., Buffalo; *for mail,* 28 N. Ogden St., Buffalo 6, N.Y.
Pawlowski, Bruno John (J'45) (CKO), Mech. Engr., RCA Victor Div., Radio Corp. of Am., Camden; *for mail,* 212 New Jersey Ave., Collingswood, N.J.
Paxson, Robert Devereaux (J'43) (RFCWA), 2934 Smith St., Ft. Wayne 5, Ind.
Paxton, Robert ('45) (CMNDJS), Works Mgr., Gen. Elec. Co., Pittsfield, Mass.
Payne, Alfred Day (J'42) (OCKFWY), Mech. Engr., Mgr., Insulation & Acoustics Dept., Otis Massey Co., Box 293, Houston 1, Tex.
Payne, Austin Lee (J'42) (AMS), Capt., 1684th Engr. Combat Bn., Camp Swift, Tex.
Payne, Don I. (J'38) (CMYR), Engr., Sales, Teletype Corp., 1400 Wrightwood Ave., Chicago 14; *for mail,* 815 Glenwood Lane, Glenview, Ill.
Payne, Earl C. ('28) (F), Cons. Engr., Consolidation Coal Co., 30 Rockefeller Plaza, New York 20, N.Y.; *for mail,* 109 Fairmount Ave., Chatham, N.J.
Payne, Frank E. ('22) (BHS), Pres., Crane Packing Co., 1800 Cuyler Ave., Chicago, Ill.
Payne, Harold Gould (J'45), Ensign, U.S.N.R.; 15 Chestnut St., Westerly, R.I.
Payne, Harold Richard (J'45) (KMBSEH), Contract Engr., Canadian Ice Mch. Co. Ltd., 65 Villiers St., Toronto, Ont., Can.
Payne, John Howard (J'39) (CAMJ), Lone Mt., Tenn.
Payne, Walter Edward (J'39) (SEKZAH), Engr., Plant Betterment, Tex. Elec. Serv. Co., Ft. Worth 1; *for mail,* 3133 Camp Bowie, Ft. Worth 7, Tex.
Payne, William Minor (J'44) (NJEF), Research Engr., Research Lab., Linde Air Products Co., Tonawanda, N.Y.
Peabody, Ernest Henry ('00) (EFSL), Pres., Peabody Engrg. Corp., 580—5th Ave., New York 19, N.Y.
Peacock, Frank Edward ('45) (HSON), Ch. Engr., Erik Floor & Associates, 139 N. Clark St., Chicago 2; *for mail,* 204 Lawndale Ave., Wilmette, Ill.
Peacock, Robert ('44), V.P., Charge of Prod., Broquinda Corp., 19th St., & 2nd Ave. S.; *for mail,* 157—12th Ave., N.E., St. Petersburg, Fla.
Peale, Walter Orville ('40), Cent. Sta. Supvr., Westinghouse Elec. Corp., 118 E. Lombard St., Baltimore 2, Md.
Pealer, Robert Ballard ('42) (NHM), Design Engr., Beaver Pipe Tools, Inc., 310 Dana Ave., Warren; *for mail,* Freedom St., Garrettsville, Ohio.
Pearce, Bert L. (J'35) (CDM), Ensign, U.S.N.R.; *for mail,* 1513 S. 4th St., Terre Haute, Ind.
Pearce, Clinton Ellicott ('40) (ANBG), Prof. Mch. Design, Kansas State College, Manhattan, Kan.
Pearce, Edwin Starling ('15; '25) (R), Pres., Ry. Serv. & Supply Corp., 510 S. Harding St., Indianapolis 7, Ind.
Pearce, Gordon Frederick (J'40) (KFCMWE), Indus. Gas Sales Engr., B. C. Elec. Ry. Co. Ltd.; *for mail,* 1526 Laurier Ave., Vancouver, B.C., Can.
Pearce, Lester Franklyn ('21; '35) (SDLNK), Designer, Colgate-Palmolive-Peet Co., 1 Newark Ave., Jersey City; *for mail,* 414 E. 5th Ave., Roselle, N.J.
Pearce, Sir (Standen) Leonard (H'44), Engr.-in-Ch., London Power Co., London, England.
Pearl, William A. ('39) (CJ), Box 46, Oregon City, Ore.
Pearlstein, Jack (J'43) (AMBC), Pvt., Co. D, 17 Bk., 5th Regt., Ft. McClellan, Ala.; *for mail,* 1170 Brighton Beach Ave., Brooklyn, N.Y.
Pearman, Edward ('29; '31; '35) (CMSTY), 57 Long Beach Blvd., Long Beach, L.I., N.Y.
Pearson, Charles-Robert (J'40) (A), Lead Engr., Boeing Aircraft Co., Wichita 1; *for mail,* 240 S. Hydraulic Ave., Wichita 7, Kan.
Pearson, Fred Leroy ('42) (OKSFH), Cons. Engr., 1310 Majestic Bldg., Detroit 26, Mich.
Pearson, Harry Randell ('34; '35) (CSFEO), Personnel Dir., Dallas Power & Light Co., 1506 Commerce St., Dallas 1, Tex.
Pearson, John Edwin (J'42) (GKSEJW), Instr., Gen. Engrg. Drawing Dept., Univ. of Ill., Urbana, Ill.
Pearson, John St. Clair ('45) (OSCZ), Control Engr., Utah Copper Co., Magna; *for mail,* 2063 Westminster Ave., Salt Lake City 5, Utah.
Pearson, Marion William (J'41) (NOGDMJ), Asst. Engr., Cent. Fdy. Co.; *for mail,* Holt, Ala.
Pearson, Nils A. ('39) (SEFZKL), Power Plant Ins. Engr., Ocean Accident & Guarantee Corp., 512 Trans. Bldg., Washington 6, D.C.
Pearson, Robert ('28) (AES), Cons. Engrg. Serv., 212 N. 3rd St., Harrisburg, Pa.
Pearson, Tage Balder ('44) (BCEFSZ), Pres., Vol-Amp Serv. Corp., 271 Madison Ave., New York 16, N.Y.
Pearson, Walter Herbert ('46) (ABJ), Power Transmission Engr., Kellett Aircraft Corp.; *for mail,* 1920 Robinson Ave., Manoa, Upper Darby, Pa.
Peart, Thomas Bertram (J'42), 166 Grenadier Rd., Toronto, Ont., Can.
Pease, Clarke Demorest ('44), Ch. Engr., Asst. to Pres., New England Indus. Devel. Corp., 80 Federal St., Boston, Mass.; *for mail,* Yale Club, Vanderbilt Ave. & 44th St., New York 17, N.Y.
Pease, Harold Arthur (J'43) (ACF), 37 Rustic Terrace, Bristol, Conn.
Pease, Ivan Ralph ('42) (RAEF), Supt. M.P., N.Y., Ont. & West. Ry., Middletown, N.Y.
Pease, Robert M. ('21; '35) (JM), V.P., Mgr., Axelson Mfg. Co., 3844 Walsh St., St. Louis 16; *for mail,* 530 Warren Ave., University City 5, Mo.
Peaslee, Dana Newton ('31) (CEKLOS), Charge of Estimates, E. B. Badger & Sons Co., 75 Pitts St., Boston 14, Mass.
Peaslee, W. ('16; '23) (JMY), Partner, Peaslee Metal Products Co., 470 Tolland St., East Hartford 8, Conn.
Peaslee, W. D. A. ('21; '30) (CMDAFB), Bendix Products Div., Bendix Aviation Corp., 401 N. Bendix Dr., South Bend 20; *for mail,* 3128 N. Pennsylvania St., Indianapolis, Ind.
Pechacek, Raymond Edward (J'40) (LBS), Prod. Engr., A. O. Smith Corp., Box 2531, Houston, Tex.
Peck, Charles V. ('41) (BCS), Spence Engrg. Co., Inc.; *for mail,* 156 Ulster Ave., Walden, N.Y.
Peck, Clair B. ('14; '35) (RECFK), *Vice-President,* '41-'43; Managing Editor, *Railway Mechanical Engineer,* Mech. Dept. Editor, *Railway Age,* Simmons-Boardman Publ. Corp., 30 Church St., New York 7, N.Y.
Peck, Clarence Erwin ('28; '35) (KLB), Engr., Charge of Indus. Htg. Sec., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 441 Spring St., Cambridge Springs, Pa.
Peck, Eugene C. ('09) (BDJ), Pres., Gen. Automatic Corp.; *for mail,* Macedonia, Ohio.
Peck, Sidney (J'40) (AFP), 217 Ocean Ave., Brooklyn 25, N.Y.
Pecker, Joseph S. ('19; '21; '35), Cons. Engr., 3969 Wilshire Blvd., Los Angeles, Calif.
Peebles, James Clinton ('39) (KES), Dean of Engrg., Ill. Inst. of Tech., 3300 S. Federal St., Chicago 16, Ill.
Peebles, Thos. A. ('12) (FJS), 6 Woodland Dr., South Hills P.O., Pittsburgh, Pa.
Peed, Harold E. (J'41), 810 N. Pennsylvania Ave., Roswell, New Mex.
Peery, David J. (J'41) (ABNO), Prof. & Head, Dept. of Aero. Engrg., Pa. State College, State College, Pa.
Peet, John L. ('41) (CDM), Div. Supt. of Maint., Gary Sheet & Tin Mills, Carnegie-Ill. Steel Corp.; *for mail,* 631 Pierce St., Gary, Ind.
Peets, Wilbur J. ('14; '25) Ch. Engr., Singer Mfg. Co., Trumbull St.; *for mail,* 973 Coolidge Rd., Elizabeth, N.J.
Peff, Ivan (J'36) (KCMZBD), Asst. Mgr., Superior Air Products Co., 132 Malvern St., Newark; *for mail,* 21 Grandview Ave., West Orange, N.J.
Pegler, James Leon ('46) (BKM), Ch. Draftsman, Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4; *for mail,* 663 Westfield Ave., Elizabeth, N.J.
Pegram, George Braxton ('28) (BAK), Prof. Physics, Columbia Univ., 116th & Broadway, New York 27, N.Y.
Pegram, William Braxton (J'34) (BCMNYZ), Research Engr., U.S. Time Corp., Waterbury; *for mail,* Orenaug Ave., Woodbury, Conn.
Pei, Ching Pong ('26; '28; '32) (BFR), Research Engr., Loco. Firebox Co., 310 S. Michigan Ave., Chicago 4, Ill.
Peickii, Vasalie Leo ('38; '45) (CMN), Mech. Engr., Equip. Engrg. Co., 3191 Casita, Glendale; *for mail,* 1140 N. Gower St., Los Angeles 38, Calif.
Peiffer, Harold Edward ('44) (BCMNHJ), Ch. Engr., Cleveland Duplex Mchy. Co. Inc., 626 Penton Bldg., Cleveland 13; *for mail,* 2700 Brookpark Rd., Cleveland 9, Ohio.
Peiler, Karl Ernest ('27) (BFKEJ), V. P., Research Dir., Hartford-Empire Co., Box 1620, Hartford 1, Conn.
Peirce, Andrew Winslow (J'43) (CMD), Indus. Engr., Clark Equip. Co.; *for mail,* 311 W. 4th St., Buchanan, Mich.
Peirce, Thomas Henry ('44) (EYRAO), Exec. Engr., Gen. Mgr., H. A. King Co., 1627 W. Fort St., Detroit 16, Mich.
Peirce, William Hollis (J'46) (EHS), U.S. Navy; 532 E. Mermaid Lane, Chestnut Hill, Philadelphia 18, Pa.
Peller, Leonard ('33; '42) (KSHLOZ), Indus. Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia, Pa.
Pelletier, Emile J. ('27; '35), Greiner Rd., Clarence, N.Y.
Pellett, William Henry ('35; '46) (CMNJEF), Plant Supt., D. R. Clark Mfg. Co. Ltd., 2462 Dufferin St., Toronto, Ont., Can.
Pellow, Richard Antrim (J'40) (CMAJF), Secy., Treas., Pellow Mch. Co., 13500 Foley Ave., Detroit 27, Mich.
Pelton, Benjamin Hiram (J'37) (C), Mountain Fuel Supply Co., Rock Springs, Wyo.
Peluso, Emidio (J'43) (SGENB), Radio Technician, 2nd Class, U.S.N., U.S.S. *Sarsfield,* DD-837, "C" Div., F.P.O., New York; *for mail,* 1747—79th St., Brooklyn, N.Y.
Pemberton, Carlysle ('12; '25; '35) (HCBEOA), Hyd. Engr., Head, Hyd. Sec., Ill. Div. of Waterways, 201 W. Monroe St.; *for mail,* 1513 W. Cook, Springfield, Ill.
Pence, Elbert Andrew, Jr. (J'42) (AJM), Ensign, U.S.N.R., Equip. Procurement Officer, Assembly & Repair, Naval Air Sta., Seattle, Wash.
Pender, Winnfield Russell ('20; '35) (EFKLSZ), Supt. Constr., A. M. Lockett & Co. Ltd., 308 Whitney Bldg., New Orleans 7, La.
Pendleton, Miles Stevens (J'32) (ZMSY), Mgr., Pur. & Sales, Pendleton Associates, 1724 Grand Cent. Terminal Bldg., New York 17, N.Y.
Pendleton, Pyam Loring ('44) (NOBCZT), Dir., Mech. Research, Ludlow Mfg. & Sales Co., 211 Congress St., Boston, Mass.; *for mail,* 13 Stratford Rd., Cranston 5, R.I.
Pendray, George Edward ('45) (AS), Pub. Relations Counselor, 55 W. 42nd St., New York 18, N.Y.
Penley, Benjamin Shaffer ('44), Plant Engr., Allied Chem. & Dye Corp., 2800 S. Sacramento Ave., Chicago 23; *for mail,* 1139 N. Le Claire, Chicago 51, Ill.
Penn, William Hill (J'44), Engr., Penn Mchy. Co., Deposit Guaranty Bank Bldg.; *for mail,* 400 N. State St., Jackson, Miss.
Pennebaker, Robt. H. ('31), Lub. Engr., Lion Oil Refining Co., El Dorado, Ark.
Penney, James H. (J'44) (CJM), 123 Spring Grove Rd., R.D. 1, Wilkinsburg, Pa.
Penney, Ronald Walter (J'41), Serv. Engr., Glenn L. Martin Co., Middle River; *for mail,* 6920 Old Harford Rd., Baltimore 14, Md.
Penniman, Abbott Lawrence, Jr. ('15; '21; '32) (CFS), Gen. Supt., Elec. Operas., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington & Liberty Sts., Baltimore 3; *for mail,* 1523 Bolton St., Baltimore 17, Md.
Penning, Cornelius J. H. ('21) (CDLORS), Adviser, Mysore Sugar Co. Ltd., Bangalore, South India.
Pennington, John Wesley (J'39) (EJK), Research Engr., Caterpillar Tractor Co., Peoria; *for mail,* 109 Indiana Ave., Morton, Ill.
Pennock, Donald W. (J'42) (AMSR), Engr., Prod. Engrg. Dept., East. Aircraft Div., Gen. Motors Corp., Linden; *for mail,* Y.M.C.A., Elizabeth, N.J.
Penny, William F. ('41), Broad Hollow Rd., Amityville R.F.D., East Farmingdale, N.Y.

Penrose, Charles ('18) (SCGR), V.P., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2, Pa.
Penrose, Edward Thornton (J'39) (OZE), 30 Parker St., Islington, Mass.
Penton, Paul (J'41) (AMB), Mech. Engr., Research & Devel., Ansco Div., Gen. Aniline & Film Corp., 12 Charles St., Binghamton; *for mail,* 51 Haynes Ave., Johnson City, N.Y.
Pepoon, Philip W. ('33; '42) (AB), Jr. Aero. Engr., Langley Memorial Aero. Lab., Natl. Advis. Com. for Aero., Langley Field; *for mail,* 155 Armstrong Dr., Hampton, Va.
Pepper, James D. (J'41) (EJK), Lt., U.S.A., Btry. D, 25th Coast Arty., A.P.O. 860, c/o Postmaster, New York, N.Y.
Pepper, Robert Hial (J'45) (HSNJBA), Engr., Test & Design, Newport News Shipbldg. & Dry Dock Co., Washington Ave., Newport News; *for mail,* 217 Chesterfield Rd., Hampton, Va.
Pepperman, Carl W. (J'42) (ACM), V.P., Ch. Engr., Lewis Engrg. Co., 339 Church St., Naugatuck, Conn.
Pepping, Raymond Austin (J'41) (ABECKM), 760 Harvard Ave., University City 5, Mo.
Peragallo, Joseph ('40) (SFBDHO), Sr. Engrg. Equip. Designer, Eastman Kodak Co., Kodak Park, Rochester 4; *for mail,* 5 Atkinson St., Rochester 8, N.Y.
Perchonok, Eugene (J'44) (AEKFBM), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Mun. Airport, Cleveland; *for mail,* 55 Buckingham Rd., Rocky River 16, Ohio.
Percival, Worth Harris (J'42) (NEAL), Box 302, Dormitory, Mass. Inst. of Tech., Cambridge 39, Mass.
Percy, James P. ('25; '30) (BCDLOZ), Gen. Supt., Ch. Engr., Cent. Aguirre Sugar Co., Aguirre, Salinas, P.R.
Percy, Willard Elmer (J'41) (B), Mar. Machinist, Mech. Div., Panama Canal; *for mail,* Box 209, Balboa, C.Z.
Perelman, Albert Abraham ('46) (JKM), Devel. Engr., Johnson-Cushing-Nevell, 101 Park Ave., New York; *for mail,* 4515—12th Ave., Brooklyn, N.Y.
Perelman, Robert, Jr. (J'44), Ensign, U.S.N.R., 3143 N. 59th St., Omaha 4, Neb.
Pergakis, Nick S. (J'44) (ANMCDH), Mech. Engr., Design, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 3808 W. 33rd St., Cleveland 9, Ohio.
Perine, Raymond Robert, Jr. (J'43) (CYMDH), 2227 Wirth Pl., New Orleans 15, La.
Perkins, Carl Keith ('41; '41) (HLSZ), N.Y. Rep., Builders Iron Fdy., 20 Vesey St., New York 7, N.Y.; *for mail,* 23 Bremond St., Belleville 9, N.J.
Perkins, Donald Lee ('18; '23; '28) (SMCY), Head, Mech. Engrg. Dept., Wayne Univ., 5140—2nd Ave., Detroit 1; *for mail,* 16511 Ward St., Detroit 27, Mich.
Perkins, Eugene V. (J'40) (ABR), Draftsman, Union Pacific R.R. Co., 15th & Dodge Sts.; *for mail,* 432 S. 39th St., Omaha, Neb.
Perkins, Harold Charles ('34) (BHRW), Assoc. Prof., Mechanics of Engrg., Cornell Univ., Ithaca, N.Y.
Perkins, N. Kenneth (J'39) (BDJ), Designer, Elgin Natl. Watch Co., National St.; *for mail,* 364 N. Worth Ave., Elgin, Ill.
Perkins, Porter Jewett, Jr. (J'44) (ABJSYZ), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 1640 Wooster Rd., Rocky River, Ohio.
Perkins, Robert Larson (J'42) (ANMCK), Design & Devel. Engr., Cent. Tech. Serv. Co., 520 Sexton Bldg., Minneapolis; *for mail,* Apt. A, 542 Lincoln, St. Paul 2, Minn.
Perkins, Sumner Ellis ('30; '41) (CDJ), Plant Supt., Charge of Operas., Sacramento Pipe Works, Inc., 16th & N. B Sts., Sacramento 3; *for mail,* 1233—40th St., Sacramento 16, Calif.
Perkins, Walter F. ('29), V.P., Mem., Bd. of Dirs., Bartlett Hayward Div., Koppers Co., Baltimore, Md.
Perkins, Wilder E. (J'25) (BDM), Dept. Mgr., Manhattan Rubber Mfg. Div., Raybestos-Manhattan, Inc., Willet St., Passaic; *for mail,* 14 Maple Pl., Nutley, N.J.
Perkinson, Thomas Francis ('41) (BERS), Mgr., R. R. Rolling Stock Div., Gen. Elec. Co., 2901 E. Lake Rd., Erie 2, Pa.
Perley, James Dwight (J'36) (CM), Dir. of Indus. Relations, Gen. Tire & Rubber Co., Akron, Ohio.
Perotto, Ribelle Vincent (J'45) (FEAMNS), Propr., Interborough Engrg. Co., 879 Bridge St., Brooklyn 1, N.Y.
Perret, Albert E. ('27; '35) (ZNMCBH), Engrg. Design, Pitometer Log Corp., 237 Lafayette St., New York 12; *for mail,* 8925 Parsons Blvd., Jamaica 2, N.Y.
Perrin, Arthur Mitchell ('39; '44) (CDS), Pres., Natl. Conveyors Co., Inc., 50 Church St., New York 7; *for mail,* 1245 E. 39th St., Brooklyn 10, N.Y.
Perrin, Felicien (J'46), Design Engr., Treen Engrg. Co., 540 N. Carrollton, New Orleans; *for mail,* 5213 Prytania St., New Orleans 15, La.
Perriton, Douglas Eric ('44) (CJMLN), Plant Mgr., Dominion Bridge Co. Ltd., 1139 Shaw St., Toronto, Ont., Can.
Perrone, Pio (H'20) (BEH), Via Pinciana Nuova 25, Rome, Italy.
Perrott, Wm. ('23), Head Mar. Engr., Office of the QM. Gen., War Dept., Washington, D.C.; *for mail,* 7505 Palisade Ave., Woodcliff, N.J.
Perrotta, Michael A. (J'39) (SF), N.Y. Steam Corp., 130 E. 15th St.; *for mail,* 200 E. 16th St., New York 3, N.Y.
Perry, Andrew Eugene, Jr. (J'37) (KEYM), Chem. Engr., Tech. Div., Edgewood Arsenal, Edgewood, Md.; *for mail,* Box 262, Denver 1, Colo.
Perry, David B. ('26) (CDMJ), Pres., Morse Chain Co., Turner Pl., Ithaca, N.Y.
Perry, George Edward (J'44) (DLS), Field Engr., Am. Blower Corp., 50 W. 40th St., New York, N.Y.
Perry, Gerald I. H. ('43) (GSC), Engr., R. R. Donnelley & Sons Co., 350 E. 22nd St., Chicago 16; *for mail,* 6502 S. Kenwood Ave., Chicago 37, Ill.
Perry, Harry M. ('20; '29) (BF), 737 N. Spring St., Los Angeles 12, Calif.
Perry, Leon Harold ('31) (NHSLGJ), Stands. Project Engr., J. W. Greer Co., 119-137 Windsor, Cambridge A-39; *for mail,* 43 Wilde Rd., Waban 68, Mass.
Perry, Millard F. (J'36) (AFM), 559 Madison Ave., Paterson 4, N.J.
Perry, Ralph Hamilton ('15; '21; '35) (C), Works Mgr., Progressive Mfg. Co., 52 Norwood St.; *for mail,* 46 Dawes Ave., Torrington, Conn.
Perry, Robert W., Jr. (J'43) (BS), 2nd Asst. Engr., U.S. Merchant Mar.; *for mail,* 2228 Pierce Ave., Niagara Falls, N.Y.
Perry, Rupert C. ('21; '35) (CDMH), Research Engr., U.S. Time Corp., Cherry Ave., Waterbury 6; *for mail,* Woodlawn Terrace, Waterbury 33, Conn.
Perry, Russell L. ('26; '40), Asst. Prof. Agric. Engrg., Asst. Engr., Exper. Sta., Univ. of Calif., Davis, Calif.
Perry, Russell Sanford (J'45) (NAH), Engr., Jackson Hyd. Mch. Co., Rockford Trust Bldg.; *for mail,* 1614 Winthrop Lane, Rockford, Ill.
Perry, Stewart Strout ('27; '41) (AEFHKR), Sales Engr., Worthington Pump & Mchy. Corp., 10 High St., Boston 10, Mass.
Perry, Thomas D. ('17; '25) (ADWY), Devel. Engr., Resinous Products & Chem. Co., 222 W. Washington Sq., Philadelphia 5, Pa.; *for mail,* 301 W. 2nd St., Moorestown, N.J.
Perry, Tom ('26) (SHF), Asst. Gen. Supt., Northwest. Elec. Co., Pub. Serv. Bldg., Portland 4, Ore.
Persak, George, Jr. (J'41) (MJNCBA), Ch. Engr., Oiljak Mfg. Co. Inc., 18 Depot Sq., Montclair; *for mail,* 25 Sutton Pl., Verona, N.J.
Person, Charles E., Jr. (J'40), Box 27, Bldg. 704, Naval Air Sta., Jacksonville, Fla.
Person, Earl R. ('28) (CFH), Dept. Supt., E. I. du Pont de Nemours & Co., Lancaster; *for mail,* 16 Devon Rd., Leominster, Mass.
Person, Howard A. ('19; '21; '35), Acting Ch. Engr., Fed. Light & Transmission Co., 70 Pine St., New York, N.Y.
Pert, David M. (J'38) (A), Capt., Army Air Forces; *for mail,* 4906 Guilford Ave., Indianapolis, Ind.
Perugi, Archie Harold (J'42) (BKHASC), Engr., Engrg. Gen. Dept., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1047 Phoenix Ave., Schenectady 8, N.Y.
Perutz, Frank (J'39) (CDMT), Prod. Engr., U.S. Finishing Co.; *for mail,* 173 Washington St., Norwich, Conn.
Peskin, Leonard C. (J'34) (ABJ), Prod. Engr., Kellett Autogiro Corp., Lansdowne Ave., Upper Darby; *for mail,* 209 E. Wharton Rd., Glenside, Pa.
Pesqueira, J. J. ('36; '41) (BJM), Engr., Morey Mchy. Co., Inc., Astoria, L.I.; *for mail,* Wentworth Hotel, 59 W. 46th St., New York 19, N.Y.
Peter, Bert H. ('30; '37) (CMJBNH), Plant Supt., Charvos, Inc., 185 Hackensack St., East Rutherford; *for mail,* 18 Maple Terrace, Verona, N.J.
Peter, Walter J. ('21; '35), Asst. Supt., Brooklyn Union Gas Co., 176 Remsen St., Brooklyn; *for mail,* 10 Sunset Rd., Valley Stream, L.I., N.Y.
Petermann, John Edward (J'39) (NJKBM), Mech. Engr., Allis-Chalmers Mfg. Co., West Allis 14; *for mail,* 2855 N. 55th St., Milwaukee 10, Wis.
Peters, A. Harold (J'36) (C), Maint. Engr., F. W. Woolworth Co., 33 W. 42nd St., New York; *for mail,* 247 Nassau Ave., Manhasset, N.Y.
Peters, Carl John (J'45) (BFJKMN), Ensign, U.S.N.R., YP-645, F.P.O., San Francisco, Calif.; *for mail,* 917 National Ave., Toledo 9, Ohio.
Peters, Charles William, Jr. (J'42) (BCW), Partner, Everett Transmission & Rubber Co., 2714 Broadway, Everett, Wash.
Peters, F. C. ('36) (CDL), Ch. Engr., Charge of Engrg. & Constr., N.J. Zinc Co.; *for mail,* Residence Park, Palmerton, Pa.
Peters, Fred Paul ('44) (JMNY), Editor-in-Ch., *Materials and Methods,* Reinhold Publ. Corp., 330 W. 42nd St., New York 18, N.Y.
Peters, Herbert Erskine ('25; '30) (EMS), Lt. Comdr., U.S.C.G., 13th Naval Dist., 920 Alaska Bldg., Seattle 4; *for mail,* 2006—35th Ave. W., Seattle 99, Wash.
Peters, Jacob Clarence ('38) (ZLC), Asst. to Dir. of Research, Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44, Pa.
Peters, Max Don (J'43) (AKJ), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 18139 Parkmount Ave., Cleveland 11, Ohio.
Petersen, Carl E. ('18; '29; '31) (BES), Comdr., U.S.N.R., c/o Supvr. of Shipbldg., U.S.N., Tampa, Fla.
Petersen, Charles Albert ('46) (BR), Designing Engr., Gen. Elec. Co.; *for mail,* Apt. 3, 730 Franklin Ave., Erie, Pa.
Petersen, Frank Fred ('45) (AEKN), Prof. Mech. Engrg., Stanford Univ., Palo Alto; *for mail,* 106 Glen Una, Los Gatos, Calif.
Petersen, John Henry (J'34) (BMSH), Lt. Comdr., U.S.N.R., N.Y. Naval Shipyard, Brooklyn; *for mail,* 2 Gloucester Court, Great Neck, L.I., N.Y.
Petersen, Knute Lund (J'42), Indus. Engr., Bell & Howell Co., 1801 W. Larchmont Ave.; *for mail,* 1530 N. Kedzie Ave., Chicago, Ill.
Petersen, Maurice Edward (J'41) (ACMN), Induction Motor Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 7 Circle Lane, Albany 3, N.Y.
Petersen, P. Jansen ('19; '27), Director, c/o Holbaek Maskinfabrik, Holbaek, Denmark.
Petersen, Paul E. ('37) (FKJDC), Fuel Engr., Chase Brass & Copper Co., Inc., Waterbury 91, Conn.
Peterson, Alfred Carl (J'40) (MCOJ), 1703 W. 6th St., Davenport, Iowa.
Peterson, Andrew Ira ('30; '35) (CMZJ), Mgr., Quality Control, Radio Corp. of Am., 5th & Sussex Sts., Harrison, N.J.
Peterson, Arthur Jonathan ('42), Maj., Supt. of Shops, 176th Ord. Bn., U.S.A., A.P.O. 828, New Orleans, La.
Peterson, Arthur W. (J'40) (BRS), 1st Lt., Corps of Engrs., Ord. Engr., Publ. Dept., Raritan Arsenal, Metuchen; *for mail,* 9 Livingston Ave., New Brunswick, N.J.
Peterson, Arvid ('21) (HSBE), Ch. Engr., Centrifugal Pump & Compressor Dept., De Laval Steam Turbine Co., Trenton 2; *for mail,* 1216 Riverside Ave., Trenton 8, N.J.
Peterson, Bertil Frederick (J'46), 1495 McKendrie St., San Jose, Calif.
Peterson, Burt A. ('31), Exper. Engr., Barber-Colman Co.; *for mail,* 1960 Harlem Blvd., Rockford, Ill.
Peterson, Conrad Leon (J'43) (CMOZ), Lt. (j.g.), Supply Corps, U.S.N.R., Naval Supply Depot, Oakland 4; *for mail,* 400 Perkins, Oakland 10, Calif.
Peterson, Edgar Wilfred ('45) (YNMCG), Ch. Planning Engr., Farmall Works, Internatl. Harvester Co., 505—41st St., Rock Island; *for mail,* 1197—25th St., Moline, Ill.
Peterson, Edward Carl (J'37) (CJM), 1st Lt., Ord. Dept., U.S.A., Philadelphia Ord. Dist., 150 S. Broad St., Philadelphia; *for mail,* Berkley Apt., Narberth, Pa.
Peterson, Edw. T. ('13), Ch. Engr., Birdsboro Steel Fdy. & Mch. Co., Birdsboro; *for mail,* 1701 Alsace Rd., Reading, Pa.
Peterson, F. P., Jr. ('33; '41) (E), 3030 Austin St., Corpus Christi, Tex.
Peterson, Fritz Waldemar (J'44) (BGESKJ), Gen. Engr., Joshua Hendy Iron Works, Sunnyvale; *for mail,* 628 Guinda St., Palo Alto, Calif.
Peterson, George Ellsworth (J'28) (SFBKEH), Asst. Prof. Mech. Engrg., Poly. Inst. of Brooklyn, 85 Livingston St., Brooklyn 2; *for mail,* 91 Harvard Ave., Rockville Centre, L.I., N.Y.
Peterson, George S. ('42) (BHEAK), Sec. Engr., Gulf Research & Devel. Co., Box 2038, Pittsburgh 30; *for mail,* 2631 Orlando Pl., Pittsburgh 21, Pa.
Peterson, Henning Loren (J'42) (CMW), 2900 E. 24th St., Minneapolis, Minn.
Peterson, Henry Ferdinand (J'41) (BZH), Research Engr., Borg-Warner Corp., 310 S. Michigan Blvd.; *for mail,* 4240 N. Mozart, Chicago, Ill.
Peterson, Howard Stanley (J'46), Sr. Product Engr., Turbine Engrg. Dept., Elliott Co., Jeannette; *for mail,* 307 Center Ave., Greensburg, Pa.
Peterson, Ivan L. ('36) (EF), Mgr., Lima Refinery, Stand. Oil Co., Drawer H; *for mail,* 1615 W. High St., Lima, Ohio.
Peterson, John D. (J'35) (CF), Mech. Engr., Atlantic Refining Co., 220 S. Broad St., Philadelphia; *for mail,* 517 S. Narberth Ave., Merion, Pa.
Peterson, Joseph Augustine (J'44) (BEL), 976 Troy Ave., Brooklyn, N.Y.
Peterson, Lloyd Edward ('41) (BHKN), Sales Consultant, Design Engr., Ilg Elec. Vent. Co., 2850 N. Crawford Ave., Chicago 41; *for mail,* 2152 Eastwood Ave., Chicago 25, Ill.
Peterson, Morris W. (J'40) (FKS), 106—5th Ave., Joliet, Ill.
Peterson, Oscar F. ('22; '35) (MS), Supt., Consltd. Edison Co. of N.Y., Inc., 7-18—37th Ave., Long Island City; *for mail,* 42-36—191st St., Flushing, N.Y.
Peterson, Ralph (J'41) (JLM), 313 Lupine Way, Short Hills, N.J.
Peterson, Richard Alwin (J'41) (ABJDK), 4923 West Medill Ave., Chicago, Ill.
Peterson, Richard George (J'38) (AMDS), Serv. Engr., Raymond Pulverizer Div., Combustion Engrg. Co., Inc., 1319 N. Branch St., Chicago; *for mail,* 8421 Oglesby Ave., Chicago 17, Ill.

Peterson, Robert Nord (J'42) (ZNLMCB), Electronic Technician's Mate, 2nd Class, U.S.N.R., U.S.S. *Adirondack*, AGC-15, F.P.O., New York, N.Y.; *for mail*, 2115 S. Patterson, Sioux City 20, Iowa.
Peterson, Roy Clifford (J'42) (CMS), 4247 Bertus St., St. Bernard, Ohio.
Peterson, Roy R. (J'44) (SKEBJN), Ensign, U.S.N.R., Engr., Turbine Devel., Bur. of Ships, Navy Dept., Code 641, Washington 25, D.C.; *for mail*, 4912 S. 28th St., Arlington, Va.
Peterson, Rudolph Earl ('26; '29; '35) (BA JN), *Student Award*, '26; Mgr., Mechanics Dept., Research Labs., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Peterson, Stephen Greacen ('42) (RBEJKH), Gen. Foreman, Portsmouth Shops, Seaboard Air Line Ry., Portsmouth; *for mail*, 6038 Westwood Terrace, Norfolk 8, Va.
Peterson, Vernon A. ('35; '42) (OSEF), Dist. Mgr., Ellicott Co., 714 W. Olympic Blvd., Los Angeles 15; *for mail*, 2753 Glendon Ave., Los Angeles 34, Calif.
Peterson, Vernon John Alfred (J'43), Lt., 428 S. Chestnut St., Spokane, Wash.
Peterson, Vernon Russell ('44) (CKS), 1214 Jewet Dr., Zanesville, Ohio.
Peterson, Victor Henry ('25; '32; '35) (RSC HEF), N.Y. Dist. Mgr., Baldwin Loco. Works, 120 Broadway, New York, N.Y.; *for mail*, Glenville Rd., Greenwich, Conn.
Peterson, Vincent C. (J'43) (NMCJW), Mech. Engr., Design & Devel., Harper Wyman Co., 8562 S. Vincennes; *for mail*, 8412 Peoria, Chicago 20, Ill.
Peterson, William Jerome (J'44), Research Dept., Propeller Div., Curtiss-Wright Corp., Caldwell, N.J.
Petit, Elwood Paul (J'42) (FKMS), 1756 S. 5th East St., Salt Lake City 4, Utah.
Petrescu, Ovid Stephen ('40) (BDHLNT), Sec. Head, Engrg. Dept., E. I. du Pont de Nemours & Co., Market St., Wilmington; *for mail*, 1824 Woodlawn Ave., Wilmington 73, Del.
Petrik, George L. (J'40), Ensign, c/o Joseph Petrik, Fairfax, Iowa.
Petrik, James Francis (J'42) (ES), Lt. (j.g.), U.S.N.R., 2805 Arlington Ave., Davenport, Iowa.
Petrinec, Julius Rudolf (J'45) (HAMSEJ), Demonstrator, Dept. of Engrg., Univ. of Toronto; *for mail*, 214 Beatrice St., Toronto, Ont., Can.
Petroe, Gregory Andrew (J'22) (NLTZKH), Devel. Engr., Mathieson Alkali Works, Inc., 2400 Buffalo Ave., Niagara Falls; *for mail*, 136 Kinsey Ave., Kenmore 17, N.Y.
Petroman, Onnie Mathias (J'32) (AJG), Draftsman, Sperry Gyroscope Co., Garden City; *for mail*, 304 Linden St., Bellmore, L.I., N.Y.
Petrovsky, Alex (J'43) (ABMJ), Aerodynamics Engr., Republic Aviation Corp., Farmingdale, L.I.; *for mail*, 294 Union Ave., Brooklyn 11, N.Y.
Petruccione, Anthony Francis (J'45) (AEK), Draftsman, Glenn L. Martin Co., Baltimore 3; *for mail*, 10 Harrison Ave., Baltimore 20, Md.
Petruzzi, Claude E. (J'38) (NCML), Asst. Ch. Engr., Nestle-Lemur Co., Evensville Ave., South Meriden, Conn.; *for mail*, 98 Pondfield Rd. W., Bronxville 8, N.Y.
Petruzzo, Albie Richard (J'45) (KAJF), Aviation Electronics Technician's Mate, 3rd Class, U.S.N.R., Bks. 676, Naval Air Sta., Pensacola, Fla.; *for mail*, 67 Kittredge St., Roslindale, Mass.
Petry, William M. (J'37) (SNJB), Engr., Babcock & Wilcox Co., Barberton; *for mail*, Clinton, Ohio.
Pettebone, Orlando Richard ('45), Calif. Packing Corp., 101 California St., San Francisco 19; *for mail*, 1856—102nd Ave., Oakland 3, Calif.
Pettersen, Hjalmar Henry Nicolay (A'43) (BH MNS), Mech. Engr., Gen. Plant Engrg., Canadian Gen. Elec. Co., Ltd.; *for mail*, 421 Wellington, Peterborough, Ont., Can.
Pettibone, Charles Eugene ('20) (CMG), V.P., Mgr., Engrg. Dept., Am. Mutual Liability Ins. Co., 142 Berkeley St., Boston 16, Mass.
Pettibone, Raymond Bruce (J'46) (BMEHKA), Sr. Test Engr., Vickers, Inc., 1400 Oakman Blvd., Detroit 32; *for mail*, 13571 Faust Ave., Detroit 23, Mich.
Pettingill, Fred. M. ('28) (WNLKHB), Design Engr., Masonite Corp.; *for mail*, Box 921, 861—7th Ave., Laurel, Miss.
Pettinos, George Francis ('13) (BCDHL), Pres., George F. Pettinos, Inc., 1206 Locust St., Philadelphia 7; *for mail*, 739 Beacom Lane, Merion Station, Pa.
Pettit, Albert R. ('34), Engr., Charge of Design, Philadelphia Elec. Co., 9th & Sansom Sts., Philadelphia, Pa.; *for mail*, 44 E. Main St., Rancocas, N.J.
Pettit, Jack Leland (J'45) (BKNJ), Engr., Metalwash Mchy. Co., 149 Shaw Ave., Irvington 11; *for mail*, 395 Franklin St., Bloomfield, N.J.
Petty, Paul Beal ('39; '44) (SOLEP), Asst. Gen. Mgr., Stacey-Dresser Engrg., 2063 E. 4th St., Cleveland 15; *for mail*, 3449 Westbury Rd., Shaker Heights, Cleveland 20, Ohio.
Petura, Frank Ensign ('36; '46) (CFS), Internatl. Gen. Elec. Co., Apartado 323, Cali, Colombia, S.A.
Petura, Richard C. (J'38) (SFE), Sales Engr., Babcock & Wilcox Co., 609 State Tower Bldg., Syracuse 2; *for mail*, 1314 Westmoreland Ave., Syracuse 10, N.Y.
Petzholt, Edmund John (J'41) (MNLZCY), Ch. Draftsman, Kollmorgen Optical Corp., 2 Franklin Ave., Brooklyn; *for mail*, 109 Park Ave., Williston Park, N.Y.
Fevear, John Clement (J'45), U.S. Navy; 138 Walton Park, Melrose, Mass.
Peverly, Arthur William (J'43) (RSYDZC), Methods Engr., Simplex Wire & Cable Co., 79 Sidney St., Cambridge 39; *for mail*, 60 Brunswick Pk., Melrose 76, Mass.
Pew, Joseph Newton, Jr. ('15; '35), V.P., Sun Oil Co., 1608 Walnut St., Philadelphia 3, Pa.
Peyinghaus, Robert ('19; '35) (CO), Elec. Engr., Army Air Forces, MacDill Field, Tampa; *for mail*, Box 8866, Tampa 4, Fla.
Peyrebrune, Henri E. ('37) (BGJM), Ch. Engr., Miehle Ptg. Press & Mfg. Co., 2011 W. Hastings St., Chicago 8, Ill.
Peyrot, Jean Bernard ('41) (OSRZJ), Pres., Min. Equip. Corp., 1270—6th Ave., New York 20; *for mail*, Apt. 3-D Dorset, Surry Strathmore Apts., 90 Bryant Ave., White Plains, N.Y.
Peyser, Leonard Francis (J'38) (JLMNYH), Plant Supt., Pioneer Watch Co. Inc., Hartford Ave. & Lyons Pl., Mt. Vernon, N.Y.
Pezzotti, Charles Peter (J'45) (CDM), Prod. Control Scheduling Asst., Paraffine Cos. Inc., 1550 Powell, Emeryville 9; *for mail*, 838 Hearst Ave., Berkeley 2, Calif.
Pfahler, Robert Duvall (J'38) (RFMC), Enginehouse Foreman, So. Ry. System, Davies St.; *for mail*, 245 Latta Ave., Ludlow, Ky.
Pfanstiehl, Paul D. (J'44) (AKM), 11 Benmor St., Medford 55, Mass.
Pfautsch, R. V. (J'30), Engr., Devel. Work, Ilg Elec. Vent. Co., 2850 N. Crawford Ave.; *for mail*, 2340 N. Kedvale Ave., Chicago, Ill.
Pfeffer, George Edward ('45) (HMO), Mech. Engr., So-Lo Works, Inc.; *for mail*, Box 118, Cornell Rd., R.R. 1, Sharonville, Ohio.
Pfeffer, James Fisher, Jr. (J'46), Engrg. Trainee, De Laval Steam Turbine Co., Trenton 2; *for mail*, 1800 Pennington Ave., Trenton, N.J.
Pfefferle, George H. ('39) (MJBEK), Tech. Dir., Dresser Industries, Inc., 1130 Terminal Tower, Cleveland 13, Ohio.
Pfeifer, Harry J. (J'42), 596 McBride Ave., Paterson, N.J.
Pfeiffer, Andrew Arthur (A'42) (CKOW), Pres., Treas., Dealers Supply Corp., 514 W. 57th St., New York 19, N.Y.
Pfeiffer, Charles George ('13) (DCM), V.P., Treas., Specialty Engrg. Co., Allegheny & Trenton Ave., Philadelphia; *for mail*, 220 Cypress Ave., Jenkintown, Pa.
Pfeiffer, David C. ('39) (BKS), Power Sales Engr., Dallas Power & Light Co., Dallas, Tex.
Pfeiffer, Frank Frederick ('37; '38) (LSKDGR), Indus. Engr., United Engrs. & Constructors Inc., 1401 Arch St., Philadelphia 5; *for mail*, 7421 Sommers Rd., Philadelphia 38, Pa.
Pfeiffer, Olin Simmers ('43) (BDHC), Checker, Alliance Mch. Co., 1049 S. Mahoning Ave.; *for mail*, 2205 Shunk Ave., Alliance, Ohio.
Pfirrmann, Henry Kenneth (J'45), 1169 S. Highland Ave., Los Angeles 35, Calif.
Pfister, Charles George (J'41) (CEJ), Warrant Officer (j.g.), 857—3rd St., Albany 5, N.Y.
Pflager, Harry Miller ('33; F'44) (RBJKLS), Retired; 19 Kingsbury Pl., St. Louis 12, Mo.
Pflug, Harry R. (J'46), 86-10—109th St., Richmond Hill, N.Y.
Pfluger, Walter Russell, Jr. (J'44), 10 Myrtle Ave., Summit, N.J.
Pfromm, John Martin (J'42) (CMDLY), 65 Brookside Ave., Hawthorne, N.J.
Phalen, James John ('40) (CDGMO), Indus. Engr., Mergenthaler Linotype Co., 29 Ryerson St., Brooklyn 5, N.Y.
Pharo, Charles Winchester ('43; '46) (FKLSZ), Engr., Hartford Steam Boiler Inspc. & Ins. Co., 319 N. 4th St., St. Louis 2; *for mail*, 5111 St. Louis Ave., St. Louis 15, Mo.
Phelan, John Joseph, Jr. (J'37) (AC), Spares Engr., Glenn L. Martin Co., Baltimore 3; *for mail*, 1002-D Wilson Point Rd., Baltimore 20, Md.
Phelan, P. A. ('19) (ORS), Design Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail*, 242 Congress Ave., Lansdowne, Pa.
Phelan, Richard Magruder (J'43) (BMJ), Lt. (j.g.), U.S.S. *Cumberland Sound*, AV-17, F.P.O., San Francisco, Calif.
Phelps, Alva Weaver ('44), Pres., Oliver Corp., 400 W. Madison St., Chicago 6; *for mail*, 241 Cumberland Ave., Kenilworth, Ill.
Phelps, Arthur Stewart ('88) (CMDJ), Safety Engr., Pratt & Letchworth Co., Inc., 189 Tonawanda St., Buffalo 7; *for mail*, 45 Courier Blvd., Kenmore 17, N.Y.
Phelps, Benjamin Louis (J'37) (CM), Cons. Engr., Olympic Engrg. Co., 3381 E. Olympic Blvd., Los Angeles 23; *for mail*, Box 1576, Beverly Hills, Calif.
Phelps, Charles C. ('09; '16; '35) (FLS), Propr., Charles C. Phelps Co., 39 Cortlandt St., New York, N.Y.
Phelps, Charles Worthington ('28; '34; '35) (SEBAFK), Instr., Mech. Engrg., Yale Univ., 400 Temple St., New Haven 11, Conn.
Phelps, Frederick Augustus ('92) (OS), 21 Fulton St., Newark 2, N.J.
Phelps, Harold Roy ('45) (CMSJNO), Asst. Ch. Engr., Am. Blower Corp., Detroit 32, Mich.
Phelps, Stuart Mason (J'38) (NRB), Mech. Engr., Devel. Gp., Gen. Ry. Signal Co., 801 West Ave., Rochester 2; *for mail*, 73 Cathaway Pk., Rochester 10, N.Y.
Philbrick, George Arthur (J'41) (ZLNB), Research Consultant, Foxboro Co., Neponset Ave., Foxboro; *for mail*, 14 Scott St., Cambridge 38, Mass.
Philbrick, Herbert Shaw ('07; '13) (EFS), Prof. Emeritus, Northwest. Univ.; *for mail*, 2130 Sherman Ave., Evanston, Ill.
Philbrick, Warren W. (J'39), Orcas, Wash.
Philipp, Gerald (J'42) (MCYLG), Sales Engr., F. W. Schiefer Mchy. Co., 39 State St., Rochester 4; *for mail*, 85 Lehigh Ave., Rochester 11, N.Y.
Philips, Jesse Orlando (J'38) (CJAM), Ingot Plant Supt., Aluminum Co. of Am., 5151 Alcoa Ave., Los Angeles 11, Calif.
Phillips, Albert A. (J'38) (CES), Lt., Co. A, 555th Engr. Heavy Pontoon Bn., Camp Polk, La.
Phillips, Alexander John ('44), Asst. Ch. Engr., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford; *for mail*, 26 Prospect St., Windsor, Conn.
Phillips, Carrington Brush ('41) (FEDK), Combustion Engr., United East. Coal Sales Corp., 1617 Pennsylvania Blvd., Philadelphia 3, Pa.
Phillips, Edmund Merrill ('17; '35) (BASNEF), Sec. Leader, Devel. Engrg. Sec., Aircraft Gas Turbine Engrg. Div., Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.
Phillips, Ellis L. ('01; '26), Pres., Long Island Ltg. Co., 50 Church St., New York, N.Y.
Phillips, George William MacPherson ('27) (LO KYZC), Chief, Chem. Engrg. Sec., East. Regional Research Lab., U.S. Dept. of Agric., Wyndmoor; *for mail*, 521 E. Willow Grove Ave., Philadelphia 18, Pa.
Phillips, Henry Weldon (J'40) (OCLNDT), Research, Devel. & Design Engr., Am. Mch. & Fdy. Co., 511—5th Ave., New York, N.Y.; *for mail*, Box 196, Murray Hill, N.J.
Phillips, Herbert Samuel (J'34) (CJKG), Asst. Sales Mgr., Gen. Plate Div., Metals & Controls Corp., Forest St.; *for mail*, 54 Upland Rd., Attleboro, Mass.
Phillips, Horace Pease, Jr. (J'39), Ensign, O-V(S), U.S.N.R.; 630—23rd St., Santa Monica, Calif.
Phillips, James Louis, Jr. (J'43) (SHFE), Field Engr., Combustion Div., Bethlehem Steel Co.; *for mail*, 503 N. Center St., Bethlehem, Pa.
Phillips, John C. (J'32) (BET), Engr., Bur. of Ord., U.S.N., 17th St. & Constitution Ave., Washington; *for mail*, 8426 Highwood Dr., Washington 20, D.C.
Phillips, John Earl (J'41) (FJOSYK), Lt. Cmdr., U.S.N.R., Bolivar, N.Y.
Phillips, Malcolm Everett, Jr. (J'40) (AJM), 2535 Windsor Rd., Parkville, Baltimore 14, Md.
Phillips, Philip Herliman ('43; '46) (CMO), 513 S. Main St., New Castle, Ind.
Phillips, Ralph Warren (J'43) (SZKCEF), Results Engr., Westvaco Chlorine Products Corp., South Charleston 3; *for mail*, 105 Maryland Ave., Charleston 3, W.Va.
Phillips, Ran Lorenzo ('42) (CFGSZ), Sta. Supt., S-E Sta., La. Power & Light Co., Sterlington, La.
Phillips, Thomas Islay ('43), V.P., Westinghouse Elec. Corp., East Pittsburgh; *for mail*, 6922 Thomas Blvd., Pittsburgh, Pa.
Phillips, Walter Benson ('37) (BKNJ), Engr., Kansas City Testing Lab., 700 Baltimore Ave., Kansas City 6, Mo.
Philo, Frank Goldsmith ('22) (SFK), Supt., Steam Power Stas., So. Calif. Edison Co., Ltd., 601 W. 5th St., Los Angeles 13; *for mail*, 4010 Pine Ave., Long Beach 7, Calif.
Phipps, Albert J. (J'36) (OKSF), Dist. Rep., Mex. Refractories Co., 1312—5th St., S.W., Canton 2, Ohio.
Phipps, Melville Arthur (J'43), Draftsman, Research Enterprises, Ltd., Leaside; *for mail*, 148 Keewatin Ave., Toronto, Ont., Can.
Phyl, Joseph ('27; '35) (OS), Project Engr., Research Corp., Lincoln Ave., Bound Brook; *for mail*, 191—2nd St, Fanwood, N.J.
Piacitelli, Joseph Albert ('23; '25; '35) (CDM), Dir. of Mgmt., Thompson & Lichtner Co., Inc., 806 Park Sq. Bldg., Boston 16, Mass.
Piasecki, Richard Francis (J'45) (SEFHNZ), Jr. Product Engr., Turbine Dept., Elliott Co., Jeannette, Pa.
Piaskowski, Fred Millard (J'45) (CJY), *Charles T. Main Award*, '44; Tool Designer, C. J. Martin & Sons, 5945 Martin, Detroit; *for mail*, 11688 Mendota, Detroit 4, Mich.
Piatt, Alvin Ray (J'42) (BMHZA), 1633 Florida St., Long Beach 4, Calif.
Piatt, Charles Robert ('38) (BKS), Asst. Supt. of Plants, Luzerne County Gas & Elec. Corp., 247 Wyoming Ave., Kingston; *for mail*, 95 Academy St., Plymouth, Pa.
Piazzoli, Louis P. ('25; '37) (CODFSM), 509 E. Crawford Ave., Connellsville, Pa.
Picado, Ramon M. ('18; '35), Cons. Engr., San Jose, Costa Rica, C.A.
Piccardo, Jack Eugene ('45) (CHM), Pres., Gen. Mgr., Shasta Pump Co., Inc., 22nd & West St., Oakland 12; *for mail*, 4381 Lincoln Ave., Oakland 2, Calif.

Picco, Peter Joseph (J'31) (CEFMN), Mgr., Agi-Gas Corp. of Mo., 7310 Ethel Ave., St. Louis 17, Mo.
Pichel, Pichel Wilson, (J'42) (AEJ), Box 536, Route 1, La Canada, Calif.
Pick, William J. ('44) (MNCJ), Stands. Engr., Sperry Gyroscope Co., Great Neck; *for mail,* 84 Horton Rd., Valley Stream, N.Y.
Pickel, William Feyreisen (J'46) (BJY), Field Engr., Rm. 249, L. B. Harrison Club, Victory Pkwy. & McMillan Sts., Cincinnati, Ohio.
Pickener, Elmore ('42) (NYLKJB), Supvg. Engr., Hood Rubber Co., 98 Nichols Ave., Watertown 72; *for mail,* 33 Indiana Terrace, Newton Upper Falls 64, Mass.
Pickett, Gerald ('37) (BHA), Prof., Applied Mechanics, Applied Mechanics Dept., Kan. State College, Manhattan, Kan.
Pickett, Louis (J'28) (JCNMLY), Mech. Engr., Reliable Tool Co., Inc., 60 Coit St.; *for mail,* 10—38th St., Irvington 11, N.J.
Pickle, Quinton L. (J'40) (EHP), Lt., U.S.N., Ship Repair Unit, Navy 140, F.P.O., San Francisco, Calif.
Pickles, Arthur Montgomery (J'43) (AHK), Devel. Engr., Aerojet Engrg. Corp., 285 W. Colorado St., Pasadena 1; *for mail,* 1104 Brent Ave., South Pasadena, Calif.
Picozzi, Dexter Anthony (J'42) (CMZ), 63 Lawn St., Providence 8, R.I.
Piecuch, Alfred Joseph (J'43) (OSLKZM), Mech. Engr., Fosdick & Hilmer, Cons. Engrs., Union Trust Bldg., Cincinnati; *for mail,* 2712 Alice St., Cincinnati 19, Ohio.
Piekarski, Joseph Bohdan (J'39) (KFGL), Steam, Water & Power Engr., Procter & Gamble Mfg. Co., 780 Washington St., Quincy 69; *for mail,* 89 Thaxter St., Hingham, Mass.
Piencinski, George (J'44) (KHBZYN), M.M.R., 2nd Class, U.S.S. *Bon Homme Richard,* CV-31, "A" Div., F.P.O., San Francisco, Calif.; *for mail,* 28-19—41st St., Astoria 3, L.I., N.Y.
Piepenbrink, Robert Earl (J'43) (SKFEDZ), Mech. Engr., Cochrane Engrg. Co., 53 W. Jackson Blvd., Chicago 4, Ill.
Pierce, Almon J. ('41) (AFL), Designing Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 206 Vernon St., Norwood, Mass.
Pierce, Brister Blake ('40) (CLSOFD), Ch. Engr., Mill Div., Gaylord Container Corp., 4th St. & Ave. S, Bogalusa, La.
Pierce, Clarence J. ('28) (FKMS), Head Mar. Surveyor, U.S. Maritime Comm., 45 Broadway, New York, N.Y.; *for mail,* 241 E. 40th St., Norfolk 8, Va.
Pierce, Conway ('22; '27; '35) (SOLEFD), Cons. Mech. Engr., Trans. Bldg., Detroit 26; *for mail,* 1130 Parker Ave., Detroit 14, Mich.
Pierce, Edgar Maxson ('39; '44) (SOELFD), Mech. Engr., Chas. T. Main, Inc., 201 Devonshire St., Boston 10, Mass.
Pierce, Frederic E. ('09), 108 E. 81st St., New York, N.Y.
Pierce, H. R. ('43), Box 741, R.R. 6, 3311 W. Easton, Tulsa, Okla.
Pierce, James Edward ('34; '38) (LKY), Engr., Dow Chem. Co., *for mail,* 2516 Rodd St., Midland, Mich.
Pierce, Joseph Douglas (J'36) (KBFHYO), Asst. Research Engr., Crane Co., 836 S. Michigan Ave., Chicago 5; *for mail,* 1634 E. 53rd St., Chicago 15, Ill.
Pierce, Lowell Jackson (J'43) (ABH), Supercharger Engrg. Dept., Gen. Elec. Co., 920 Western Ave., Lynn, Mass.
Pierce, Marion Charles ('37; '44) (YAC), Tire Constr. Engr., B. F. Goodrich Co., 500 S. Main St., Akron 8; *for mail,* 2341—19th St., Cuyahoga Falls, Ohio.
Pierce, Raymond C. ('20) (ONRHA), Cons. Engr., 1100 N. Dearborn St., Chicago 10, Ill.
Pierce, Theodore Braley ('43), Asst. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.; *for mail,* 108 Cranford Pl., Teaneck, N.J.
Pierpoline, Mario Frank (J'43) (ABCFRK), Design Engr., Westinghouse Elec. Corp., South Philadelphia; *for mail,* 222 Lafayette Ave., Swarthmore, Pa.
Pierse, Harold Edward (J'38) (JFKOSY), Material Takeoff (Piping) Dept., M. W. Kellog Co., 225 Broadway, New York 7; *for mail,* 56-24—211th, Bayside, L.I., N.Y.
Pierson, Edward D. (J'39) (BJS), Maj., Army Air Forces Hdq., I Staging Comd., Army Air Field, Merced, Calif.
Pieters, Ivon S. ('22; '25; '35) (DFS), V.P., Philbrico Jointless Firebrick Co., 1800 Kingsbury St., Chicago, Ill.
Pietsch, Eugene Harold (J'45) (KMNBC), Air Conditioning Engr., Midwest Air Control, 707 Locust, Des Moines, Iowa.
Pietsch, Herman August (J'40) (KFEMC), Ch. Engr., Heater Sec., Dravo Corp., 300 Penn Ave., Pittsburgh 22; *for mail,* 894 Heckler Dr., Pittsburgh 20, Pa.
Pigage, Leo Charles (J'39) (CMG), Asst. Prof. Gen. Engrg., Purdue Univ., Lafayette, Ind.
Pigott, Reginald James Seymour ('12; '13; '18; F'38) (EOABKZ), *Vice-President,* '36-'38; Ch. Engr., Gulf Research & Devel. Co., Box 2030, Pittsburgh 30, Pa.
Pigott, Sir Stephen J. (Non-Member), *A.S.M.E. Medallist,* '38; Managing Dir., John Brown & Co., Ltd., Clydebank, Scotland.
Pihlman, Arthur A. ('41), 130 E. 15th St., New York, N.Y.
Pike, Clinton B., Jr. (J'38) (DLN), Mech. Engr., Calif. Prune & Apricot Growers Assn., Market & San Antonio, San Jose; *for mail,* 1650 Shasta Court, San Jose 11, Calif.
Pike, Kenneth Whitacre ('24; '26; '35) (CDLMGO), Ch. Engr., Quaker Oats Co., 141 W. Jackson Blvd., Chicago 4; *for mail,* 472 S. Kenilworth Ave., Elmhurst, Ill.
Pike, Otto S. ('38) (SZKCO), Pres., Cochrane Steam Specialty Co., 80 Federal St., Boston 10, Mass.
Pike, Raymond James (J'46), Apprentice Engr., Frye Packing Co., Seattle; *for mail,* 1402 Atlantic St., Seattle 44, Wash.
Pilcher, John Alsop ('95; '04) (RSJD); Retired; 436 Walnut Ave., S.W., Roanoke 16, Va.
Pilgram, Walter, Jr. (J'44) (AFBHNZS), Test Engr., Exper. Div., Wright Aero. Corp., Paterson; *for mail,* 30 S. Maple Ave., Ridgewood, N.J.
Pinches, Conrad Hume, III ('44) (BCEFKS), V.P., Prat-Daniel Corp., 84 S. Water St., East Port Chester, Conn.
Pinder, Harold Clay ('45), Plant Engr., Gen. Engrg. Co., Ltd., Scarboro; *for mail,* 515 Old Orchard Grove, Toronto, Ont., Can.
Pindras, Raymond (J'36) (HCNM), Devel. Engr., Chicago Pump Co., 2336 W. Wolfram St.; *for mail,* 3937 N. Kedzie Ave., Chicago 18, Ill.
Pineles, Jacob (J'40) (EALJFB), 632 Washington St., Hoboken, N.J.
Pinger, George Christian ('44; '46) (CFJLOS), George S. May Co., New York; *for mail,* 5 Vine St., Bronxville, N.Y.
Pinkerton, Andrew ('92) (SOFEC), Retired; 1045 Orilla Delmar Dr., Santa Barbara, Calif.
Pinkus, Jerome Roy (J'44) (KFZH), Project Engr., Payne Furnace Co., 336 N. Foothill Rd., Beverly Hills; *for mail,* 1466 S. Bedford St., Los Angeles, Calif.
Pinn, Samuel, Jr. (J'41) (CSOHDN), Engr., Bur. of Ships, Navy Dept., 17th & Constitution Ave., Washington 25; *for mail,* 4021 Beecher St., N.W., Washington 7, D.C.
Pinnes, Robert William (J'38) (EBASKN), Research Engr., De Laval Steam Turbine Co., Trenton 2, N.J.; *for mail,* 1061 Wheeler Ave., New York 59, N.Y.
Pinney, Ernest F. ('23; '25; '35), M.M., Pratt Works, Socony-Vacuum Oil Co., Inc., Review Ave., Long Island City, N.Y.
Pinnolis, Samuel (J'42) (MAC), Seaman, 2nd Class, U.S.N.; *for mail,* 92-21—215th Pl., Queens Village, L.I., N.Y.
Pintar, Joe (J'40) (NMJHKS), Mech. & Mar. Engr., c/o Supvr. of Shipbldg., U.S.N., Todd Shipyards Corp., San Pedro; *for mail,* Apt. 8, Byram Apts., 1037 Byram St., Los Angeles 15, Calif.
Piper, Fred Farwell, Jr. (J'44) (BJM), Mech. Engr., Aluminum Research Lab., Aluminum Co. of Am., Box 772; *for mail,* 165 Aluminum City Terrace, New Kensington, Pa.
Piper, Harlow Harding (J'44) (MACJKD), Ensign, U.S.N.R., Stores & Communication Officer, LST-521, Green Cove Springs, Fla.; *for mail,* R.F.D. 3, Princeton, Ill.
Pippin, Clarence A. (J'40) (CKF), Supt., Magnesium Corp., Marysville; *for mail,* Route 4, 4126 Bay City Rd., Midland, Mich.
Pisarchik, Harry George (J'41) (MCBJYZ), Design Dept., Fed. Tel. & Radio Corp., 360 Thomas St., Newark, N.J.; *for mail,* 239 E. 5th St., New York 3, N.Y.
Piske, Richard Adolph, Jr. (J'45) (KLCTO), Engr., Carrier Corp., 300 Ivy St.; *for mail,* 236 Bolling Rd., N.E., Atlanta, Ga.
Pita, Edward Gerald (J'45) (KNEBCZ), Jr. Engr., Internatl. Div., Carrier Corp., 122 E. 42nd St., New York; *for mail,* 37-26—65th St., Woodside, L.I., N.Y.
Pitman, Richard Wallace ('44) (ABC), Franklin Inst., Philadelphia 3, Pa.
Pitman, W. Andrew (J'40) (ABNJY), Box 341, 43 Highland Terrace, Stratford, Conn.
Pitre, Melford John (A'44) (ACM), Supt., Engrg. Dept., Fidelity & Casualty Co. of N.Y., 80 Maiden Lane, New York 7, N.Y.
Pittelkow, Arthur G. ('39), Project Pur. Agt., Austin Co., Curtis Bldg., Detroit; *for mail,* Apt. C-18, 17140—3rd Ave., Detroit 3, Mich.
Pittelkow, Lawrence Albert (J'40) (TOMC), Partner, Mfrs. Rep., Albert L. Pittelkow & Son, 15 Wilder St., Minneapolis 1; *for mail,* 1366 Van Buren Ave., St. Paul 4, Minn.
Pittendreigh, William Wallace (J'39) (NAYJKM), 25 Pinette St., New Bedford, Mass.
Pittman, Paul Ray, Jr. (J'40) (AHZ), Engr., Jack & Heintz, Bedford; *for mail,* R.F.D. 1, Hudson, Ohio.
Pitts, Carl Elton (J'45) (AJP), Apt. 9, 2109 Austin, Houston 3, Tex.
Pitts, Derrell Dick (J'40) (ACEMSF), Capt., Supply & Munitions Officer, 527th Antiaircraft Arty. Automatic Weapons Bn., San Diego, Calif.; *for mail,* Grandview, Tex.
Piuck, Daniel (J'40) (JSYNZB), Devel. Engr., Kiemac Mfg. Co., 37-48—61st St., Long Island City; *for mail,* 1072—54th St., Brooklyn 19, N.Y.
Place, Louis Victor, Jr. (J'20) (CDK), Dir., Exec. V.P., Manati Sugar Co., 106 Wall St., New York 5; *for mail,* 2 E. 67th St., New York 21, N.Y.
Place, Oliver ('29; '38) (CDMSJ), Ch. Draftsman, Barber-Greene Co., 633 W. Park Ave.; *for mail,* Box 594, R.F.D. 2, Aurora, Ill.
Place, Palmer Bruce ('38) (EFKS), Research & Devel. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 211 N. Germantown Rd., Chattanooga, Tenn.
Placek, Earl Wayne (J'42) (ABNK), Serv. Engr., C & H Supply Co., 2723—1st Ave. S., Seattle; *for mail,* 3320 E. Republican, Seattle 2, Wash.
Placey, William Robert (J'42) (CDA), Prod. Mgr., Melvindale Forge Div., Timken Detroit Axle Co., 13500 Visger Rd., Melvindale; *for mail,* 1315 Dragoon Ave., Detroit 9, Mich.
Plagwit, Eric ('18) (KCAEFJ), Mgr., Chimney Div., Rust Engrg. Co., Clark Bldg., Pittsburgh, Pa.
Plamann, John August, Jr. (J'40) (SKF), Oper. Engr., Riley Stoker Corp., 9 Neponset St., Worcester, Mass.
Planck, Carl G., Jr. (J'40), 1st Lt., Pilot, 9th Fighter Sqd., 49th Fighter Gp. 5th Army Air Force; *for mail,* 8 Sutherland Ave., Charleston 19, S.C.
Planck, Ivan Albert (J'43) (EFK), Prof. Mech. Engrg., Ind. Tech. College, 221 E. Washington St., Ft. Wayne 2; *for mail,* 1913 Drexel Blvd., Ft. Wayne 5, Ind.
Planiol, Andre Paul Eugene ('42), V.P., Stratos Corp., Ranger Aircraft Engs. Div., Fairchild Eng. & Airplane Corp., Farmingdale; *for mail,* Chichester Rd., West Hills, Huntington, N.Y.
Plant, Albert Morton ('45) (SOKMFJ), Results Engr., United Illum. Co., 510 Grand Ave., New Haven 13, Conn.
Plant, William Albert ('39) (LWCDFH), Res. Mgr., Abitibi Power & Paper Co. Ltd.; *for mail,* Box 199, Smooth Rock Falls, Ont., Can.
Plantinga, John Everett (J'44), 1 Sutton Pl. S., New York, N.Y.
Plapp, Elmer Brauer ('19; '25; '35) (SFEHZB), Power & Combustion Engr., Am. Smelting & Refining Co., Box 1111, El Paso, Tex.
Plass, Raymond B. ('25; '32; '35) (SBKFLZ), Ch. Engr., Ray Oil Burner Co., 401 Bernal Ave., San Francisco 12; *for mail,* 1803 Oxford St., Berkeley 4, Calif.
Plass, Robert H. ('42) (BLT), 936 Broad St., Bloomfield, N.J.
Platou, Leiv Stoud ('36) (ABM), Ch. Engr., E. W. Bliss Co., Hastings Ave.; *for mail,* 1911 Joffre Ave., Toledo 7, Ohio.
Platt, Barney Spencer ('44) (SKFDEH), Supt. Power & Maint., Burroughs Adding Mch. Co., 6071—2nd Ave., Detroit 32; *for mail,* 15110 Faust Ave., Detroit 23, Mich.
Platt, Robert William (J'44) (ABE), Engrg. Div., Chrysler Corp., Highland Park; *for mail,* Apt. D-14, 17140—3rd Ave., Detroit 3, Mich.
Platt, Miss Virginia Provine (J'45) (BHJ), 3231—43rd W., Seattle 99, Wash.
Platt, Walter Kennedy (J'45), Ensign, U.S.N.R., N.Y. Naval Shipyard, Brooklyn, N.Y.; *for mail,* Marlkress Rd., Haddonfield, N.J.
Pletta, William Herman ('36; '45) (SFKH), Design Engr., Commonwealth Edison Co., 72 W. Adams St., Chicago 90, Ill.
Pleyte, Thomas William (J'42) (EZB), Exper. Engr., Le Roi Co., 1706 S. 68th St., West Allis; *for mail,* 3928 W. Galena St., Milwaukee 8, Wis.
Pliner, Norman S. (J'41) (TYACM), Mech. Engr., Celanese Corp. of Am.; *for mail,* Algonquin Hotel, Cumberland, Md.
Plonsker, Maurice J. ('17; '22; '35), Lt. Comdr., U.S.N.R., Naval Air Base, Annex 2, Little Creek, Va.
Plotner, Norman E. (J'40) (ECAN), Mar. Engr., Bur. of Ships, Navy Dept., Washington 1, D.C.; *for mail,* 6322—20th Ave., Green Meadows, Hyattsville, Md.
Plum, Charles Robert (J'46) (AEJ), Installation Engr., Gen. Elec. Co., Box 1858, Seattle 11, Wash.
Plumley, Richard Gardiner ('40) (CMG), 155 Holmes Ave., Glenbrook, Conn.
Plummer, Clayton Robert (J'42) (NG), Asst. Prof. Engrg. Drawing, Univ. of Tenn., Knoxville, Tenn.
Plummer, Fred Leroy ('42) (EJBYL), Dir., Engr., Hammond Iron Works, Box 629, Warren, Pa.
Plummer, Gilbert Ashton ('45), Dir., Ettingshall Engrg. Works, John Thompson Water Tube Boilers Ltd.; *for mail,* "Ryndleside," Spring Hill Lane, Penn, Wolverhampton, Staffs, England.
Plummer, Raymond Benton ('39) (BHKDOM), Francis Engrg. Co., Eddy Bldg., Saginaw; *for mail,* 632 Jefferson Ave., Saginaw 16, Mich.
Plummer, Richard Wentworth ('44), 2625 "O" St., N.W., Washington 7, D.C.
Plummer, Wade S. (J'36), 506 Jewell St., Delmar, Del.
Plunkett, Brian ('37) (CDMTNZ), Supt., Knitted Fabrics Div., Celanese Corp. of Am., Cumberland, Md.
Podell, Odif (J'40) (CLT), Cedar Grove Farm, Cedar Grove Rd., Toms River 58, N.J.
Podmore, Fred Herboth ('37) (C), Mgr., Constr. & Maint. Dept., Richfield Oil Corp., 155 W. Washington Blvd., Los Angeles 15, Calif.
Podnossoff, Jules (J'31) (CMKSZO), *Charles T. Main Award,* '30, *Student Award,* '31; Engr., Air Conditioning & Refrig. Dept., Internatl. Gen. Elec. Co., Inc., 570 Lexington Ave., New York 22; *for mail,* 1078 Madison Ave., New York 28, N.Y.

Podolsky, John Paul (J'43) (HBNAJM), Tool Designer, Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail*, 5600 Harley Dr., Philadelphia 43, Pa.
Poel, Harlan James (J'41) (DOCFLJ), Tech. Serv. Rep., Explosives Dept., E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington, Del.; *for mail*, 1142 Underwood Ave., S.E., Grand Rapids 6, Mich.
Pofahl, Thomas Herman (J'38), Mech. Engr., Pipe Line Design, El Paso Natural Gas Co., Bassett Tower; *for mail*, 1106 E. Yandell Blvd., El Paso, Tex.
Poggi, Martin Joseph (J'37) (AHKFC), Installation Engr., Airsupply Co., 5959 W. 3rd St., Los Angeles 36, Calif.; *for mail*, 3327 E. Republican St., Seattle 2, Wash.
Pogue, Joseph E. ('21) (AFGY), V.P., Charge of Petroleum Dept., Chase Natl. Bank, 18 Pine St., New York 5, N.Y.
Pogue, Roger Boston (J'41) (FKS), Ensign, U.S.N.R., U.S.S. *St. Louis*, F.P.O., San Francisco, Calif.; *for mail*, 547 Hawthorn Lane, Winnetka, Ill.
Pohl, Charles Albert (J'43) (NMWDOS), Route 1, Blaine, Wash.
Pohl, Richard Gunter (J'44) (BKS), Engr., Leach Mfg. Div., Condenser Serv. & Engrg. Co., Hoboken, N.J.; *for mail*, 200 W. 70th St., New York 23, N.Y.
Pohle, Hans Albert (J'43) (COY), Territorial Mgr., Sonotone Corp., 255 Main St., Pawtucket; *for mail*, 11 Elmhurst St., Hoxsie, R.I.
Pohlke, Philip A. (J'29) (LYTKC), Research Engr., Hercules Powder Co., *for mail*, Box 225, Hopewell, Va.
Pohlman, Donald Frederick (J'44) (SKBEFH), Technician, 3rd Grade, U.S.A., Manhattan Dist., Bks. Area, Oak Ridge, Tenn.; *for mail*, 2217 Court St., Saginaw, Mich.
Polak, Irving P. (J'43) (AKEB), Aero. Engr., Naval Air Exper. Sta., Naval Shipyard, Philadelphia 12; *for mail*, 7091 Forrest Ave., Philadelphia 38, Pa.
Poland, Robert Lenwood (J'46), Mch. Design, Lufkin Fdy. & Mch. Co., *for mail*, 401 S. 2nd, Lufkin, Tex.
Polaner, Jerome Lester (J'44) (AKS), Seaman, 1st Class, U.S.N.R., 417 Clinton Pl., Newark 8, N.J.
Pole, Lanier A. (J'43), The Old Mill Place, Poolesville, Md.
Polhamus, Edward Charles (J'44), 5304—41st St., N.W., Washington, D.C.
Poliakoff, Rouvime B. ('42) (JMC), Engrg. Consultant, Associated Fdys. & Mfrs., Inc., 555 W. 54th St., New York 19, N.Y.
Poliakoff, Theodor (J'40) (KL), Designer, F. L. Smidth & Co., 60 E. 42nd St.; *for mail*, 86 W. 183rd St., New York, N.Y.
Polick, John William (J'41) (MCN), Maj., Commanding Officer, Hdq., 1303rd Engr. Regt., A.P.O. 181, c/o Postmaster, San Francisco, Calif.; *for mail*, Box 126, Elrama, Pa.
Poling, Warren D. (J'43) (HLNSZ), Mech. Engr., Pittsburgh Plate Glass Co., Barberton; *for mail*, 722 Westerly Rd., Akron 7, Ohio.
Polivka, Jaro Joseph ('44) (ABH), Cons. Engr., 609 Hobart Bldg., San Francisco; *for mail*, 1150 Arch St., Berkeley 8, Calif.
Polivka, Norbert Leonard (J'46) (COS), 1903 S. 50th Ave., Cicero 50, Ill.
Polk, Albert F. (J'33) (OMNJD), V.P., Charge of Engrg., Sheffield Corp., 721 Springfield St., Dayton 1, Ohio.
Polk, Fred E. (J'39) (ADJ), Lt. (j.g.), U.S.N.R., U.S.S. *Oneota*, AN-85, F.P.O., San Francisco, Calif.
Polk, Gilbert Cameron ('33) (KSHBCL), V.P., Secy., Am. Blower Corp., Box 58, Roosevelt Park Annex, Detroit, Mich.
Polk, Irving (J'43), 214 Morrison Ave., Somerville, Mass.
Polk, Louis F. ('43) (CMZNJA), Pres., Sheffield Corp., 721 Springfield, Dayton 1, Ohio.
Pollak, John Paul (J'42) (MJDYCA), Project Engr., Atlas Aircraft Products Corp., 405 E. 42nd St., New York 17; *for mail*, 520 E. 90th St., New York 28, N.Y.
Pollak, Rudolf ('37) (KCOZBS), Ch. Engr., Rockefeller Center, Inc., 50 Rockefeller Plaza, New York 19, N.Y.
Pollard, Edw. V. (J'21), Asst. Engr., Gen. Elec. Co., West Lynn; *for mail*, 64 Atlantic St., Lynn, Mass.
Pollard, Edwin E. (J'43) (J), Pollogan Co., 1906 —6th Ave. S., Birmingham 3, Ala.
Pollard, Frank Waterman ('45) (BNZJHS), Serv. Engr., Westinghouse Elec. Corp., 420 S. San Pedro St., Los Angeles 43, Calif.
Pollard, Howard B., Jr. (J'38) (SFEDCK), Engr., Paul Wright & Co., Brown-Marx Bldg., Birmingham; *for mail*, 6 Myrtle St., Mobile 17, Ala.
Polleys, Herbert Rhodes ('28) (BYMN), Mech. Devel. Engr., U.S. Rubber Co., Maple St., Naugatuck; *for mail*, 285 Central Ave., New Haven 15, Conn.
Pollitz, Harold C. ('37) (DEC), Ch. Engr., Iowa Mfg. Co.; *for mail*, 331—27th St. Dr., S.E., Cedar Rapids, Iowa.
Pollock, Robt. T. ('39), Cons. Engr., 570 Lexington Ave., New York, N.Y.
Polomik, Edw. E. (J'35), 450 Mathilda Ave., Sunnyvale, Calif.
Polonsky, Abraham (J'43) (NHB), Mch. Tool Designer, Lapointe Mch. Tool Co., 34 Tower St., Hudson; *for mail*, 21 Exchange St., Brockton 12, Mass.
Polson, Joseph Albert ('06; '12) (AEFORS), Prof. Steam Engrg., Univ. of Ill., Urbana, Ill.
Polston, John Ray ('45) (HBEK), Mech. Engr., Stanolind Pipe Line Co., Box 591, Tulsa 2, Okla.
Pomeranz, Robert Edgar (J'43) (MNCAH), Devel. Engr., Head, Devel. Dept., Edwards Co., Div. of Rogers Diesel & Aircraft Corp.; *for mail*, Box 796, Sanford, N.C.
Pomeroy, Charles Ross ('21; '35) (NBCJM), Mch. Designer, E. W. Bliss Co., 1420 Hastings St., Toledo 7; *for mail*, 4213 Westway St., Toledo 12, Ohio.
Pomeroy, G. M. (J'19), Sales Mgr., Matthews Mfg. Co., 104 Gold St., Worcester, Mass.
Pomeroy, Morris Hayden (J'42) (AKBHEF), 1118 Divisadero, Fresno, Calif.
Pomeroy, Theodore M. (J'35) (NLDMG), Engr., Prod. Div., Am. Sugar Refining Co., 120 Wall St., New York 5; *for mail*, 116 Corlies Ave., Pelham 65, N.Y.
Pommer, Fredrick Joseph (J'45) (NHMYJC), Mech. Engr., Farrel-Birmingham Co., Inc., Main St., Ansonia; *for mail*, 71 High St., East Haven 12, Conn.
Pond, Clarence Edwards ('42) (RJFDCB), Asst. to Supt. M.P., Norfolk & West. Ry. Co.; *for mail*, 812 Maiden Lane, Roanoke, Va.
Pond, Henry Otis ('04; '07) (CDMSOF), Indus. Engr., Office of Chief of Engrs., U.S.A., Washington 25, D.C.; *for mail*, 4242 N. 2nd Rd., Arlington, Va.
Pond, Leander Norton ('44) (NCBDH), Mgr., Mch. Design, Corning Glass Works, Corning, N.Y.
Pond, Richard K. (J'39) (CMJD), Supvr., Indus. Engr., Westinghouse Elec. Corp., 150 Pacific Ave., Jersey City, N.J.
Ponomareff, Alex I. ('26; '32; '35) (HBSAE), Sec. Engr., Pumps & Blowers Div., Westinghouse Elec. Corp., Lester; *for mail*, 148 Ellis Rd., Havertown, Pa.
Pontius, Paul Edward (J'44) (ESFN), Lt. Comdr., (DE), U.S.N.R., Engr. Officer, U.S.S. *Proteus*, AS-19, F.P.O., New York, N.Y.
Pool, Ralph Y. ('21), Exec., Fla. Power & Light Co., Box 3100, Miami, Fla.
Poole, Edward Milton (J'36) (SEFKRO), Research Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail*, 528 Otterbein Ave., Barberton, Ohio.
Poole, Joe A. (J'46) (EKS), Ch. Engr., Dallas Air Conditioning Co., 2809 Canton St., Dallas 1; *for mail*, Apt. 4, 4819 Swiss Ave., Dallas 4, Tex.
Poor, Albert F., Jr. (J'40) (ABS), 1505 Hill Ave., Wilkinsburg, Pa.
Poor, Hustace Hubbard (J'39) (SFKB), Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 112 Park Ave., Yonkers 3, N.Y.
Poore, Everett Anderson (J'43) (ACS), 39 Inman St., Lawrence, Mass.
Poorman, George Ellmaker (J'36) (ABWYR), Draft Design Engr., Bell Aircraft Corp., Niagara Falls Blvd., Niagara Falls; *for mail*, 45 Kenwood Ave., Kenmore 17, N.Y.
Pooser, Arthur Keith ('42) (BHS), Plant Engr., Ecusta Paper Corp., Pisgah Forest, N.C.
Pope, Charles Linder ('43) (NLBESH), Lub. Engr., Eastman Kodak Co., Kodak Park, Rochester 13; *for mail*, 361 Sagamore Dr., Rochester 12, N.Y.
Pope, Clarence James (J'14) (SEFK), Sr. Engr., Pub. Utility Comm., 1060 Broad St., Newark; *for mail*, 399 Tremont Pl., Orange, N.J.
Pope, Harold Linder ('05; '12) (CDM), 112 New Haven Ave., Woodmont, Conn.
Pope, Howard Louis ('45), Mgr., Engrg. Serv., Cincinnati Milling Mch. Co., Cincinnati; *for mail*, Box 369, Route 10, Kugler Mill Rd., Cincinnati 27, Ohio.
Pope, Joseph ('15; '23) (SEOFLH), V.P., Stone & Webster Engrg. Corp., 90 Broad St., New York 4, N.Y.
Pope, Lyman B. ('88) (CMW), Pres., Treas., Pope Mchy. Corp., 261 River St., Haverhill; *for mail*, 112 S. Park St., Bradford, Mass.
Pope, Samuel Austin (J'15) (SOE), Pres., William A. Pope Co., 26 N. Jefferson St., Chicago 6, Ill.
Pope, Samuel Hatch ('45) (JPY), Staff Engr., Gulf Oil Corp., Box 661; *for mail*, 1722 S. Columbia Pl., Tulsa, Okla.
Popiel, Roman (J'46), 30 Grant Ave., Carteret, N.J.
Poritsky, Hillel (M'45), Cons. Engr., Gen. Elec. Co., Schenectady, N.Y.
Poritz, Julius Louis (J'43) (SKFENH), c/o Holleman, 39 W. Francis Ave., Pittsburgh, Pa.
Porro, Robert Michael (J'43) (C), Engr., Frankford Arsenal; *for mail*, 1421 Arch St., Philadelphia 37, Pa.
Porsche, Charles F. (J'40), 4388 White Plains Ave., New York 66, N.Y.
Porskievies, Anthony Joseph ('42) (ACH), Engr., Vickers, Inc., 1400 Oakman Blvd., Detroit, Mich.
Porter, David Burr ('16; '25) (CDM), Prof. Indus. Engrg., N.Y. Univ., University Heights, New York 53, N.Y.
Porter, George Arthur ('44) (SCFKJN), Asst. Plant Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Porter, George Jones (J'37) (SOFC), Test Engr., Power Sta., Va. Elec. & Power Co., Reeves Ave., Norfolk; *for mail*, 931 Graydon Ave., Norfolk 7, Va.
Porter, H. Hobart ('02; F'41), Chmn., Bd. of Dirs., Am. Water Works & Elec. Co., Inc., 50 Broad St., New York 4, N.Y.
Porter, Harry Thomas ('41) (HSZ), Owner, Harry T. Porter Co., Union Cent. Bldg., Cincinnati 2; *for mail*, 6 Sylvan Lane, Wyoming 15, Ohio.
Porter, James Clifton (J'40) (AEF), Mech. Engr., No. Regional Research Lab., University & Nebraska Sts.; *for mail*, 814 N. Glen Oak, Peoria, Ill.
Porter, James Harry ('36; '44) (SKERHO), Asst. Dean of Engrg., Johns Hopkins Univ., 34th & Charles St., Baltimore 18, Md.
Porter, Lawrence Ashton ('42) (REF), Ch. Mech., Officer, Seaboard Air Line R.R., Granby, Norfolk 10; *for mail*, Algonquin Park, Norfolk 8, Va.
Porter, Lloyd Judson ('29; '35; '35) (MDNCLS), Res. Engr., Travelers Indemnity Co., 26 Crocker Bldg.; *for mail*, 24 White St., Taunton, Mass.
Porter, R. Clay ('36) (FKS), Instr. in Mech. Engrg., Univ. of Mich., Ann Arbor, Mich.
Porter, Roy Hiram ('18) (CDES), Cons. Engr., N.J. Zinc Co., 160 Front St., New York, N.Y.; *for mail*, 462 Columbia Ave., Palmerton, Pa.
Porter, William Easton (J'46) (KOS), U.S. Navy 3286 Ardmore Rd., Shaker Heights 20, Ohio.
Portz, Kenneth Lamont (J'42) (OCJM), Draftsman, Wertz Engrg. Co. Inc., 441 N. 2nd St., Reading; *for mail*, 216 Philadelphia Ave., Shillington, Pa.
Posey, James ('07; '19; '34) (BEF), Cons. Engr., 10 E. Pleasant St., Baltimore, Md.
Pospisil, Louis John ('20) (SNHKEB), Retired; W. 103—17th Ave., Spokane 9, Wash.
Posse, Ernest W. (J'39) (AFT), Capt., Armd. Force, U.S.A., Gp. 2, Officers Quarters' 12, Armd. Force Replacement Trng. Center, Ft. Knox, Ky.
Posselt, Ejnar (CFS) ('07; '13), V.P., Charge Engrg. & Pur., Lone Star Cement Corp., 342 Madison Ave., New York, N.Y.
Possner, Louis (J'46) (CEJ), Jr. Engr., Rm. 203, Engrg. Design, Univ. of Calif.; *for mail*, 2317 Le Conte Ave., Berkeley 4, Calif.
Post, Arthur Edward (J'39) (BEK), Gas Utilization Supvr., Nassau & Suffolk Ltg. Co., 250 Old Country Rd., Mineola; *for mail*, 104 Kings Pkwy., Baldwin, L.I., N.Y.
Post, Madison (J'39), Capt., U.S.A., Aberdeen Proving Ground, Md.
Post, Nicholas (J'40) (AKEBC), Planning Asst., Control Equip. Branch, Equip. Lab., Air Tech. Serv. Comd., Wright Field, Dayton, Ohio.
Postel, Carl Elwood (J'42) (BJCH), Tool Designer, Farmall Works, Internatl. Harvester Co., 505—41st St., Rock Island, Ill.; *for mail*, 2724 Scott St., Davenport, Iowa.
Postin, George Daniel (J'44) (JSKEHL), Res. Engr., British Am. Oil Co., Ltd., Box 1120, Moose Jaw, Sask., Can.
Postlewaite, William Rogers ('43), Mech. Engr., Stand. Oil Co. of Calif., 225 Bush St., San Francisco; *for mail*, 360 Menlo Oaks Dr., Menlo Park, Calif.
Potter, Andrey A. ('12; F'26; H'45) (SKRFEC), *President*, '33; Dean of Engrg., Purdue Univ., West Lafayette, Ind.
Potter, Bruce G. (J'40) (BKM), Prod. Engrg., Leland-Gifford Co., 1001 Southbridge St., Worcester 1; *for mail*, 17 School St., Northboro, Mass.
Potter, Charles Gaunt (J'41) (REFS), 803 Pine St., Pine Bluff, Ark.
Potter, Erford M. ('17; '35) (SOG), V.P., Ivan T. Johnson Co., Inc., 95 Madison Ave., New York 16, N.Y.
Potter, James Harry ('36; '44) (SKERHO), Assoc. in Mech. Engrg., Johns Hopkins Univ., 34th & Charles Sts., Baltimore 18, Md.
Potter, James T. ('40) (FLSTWZ), Partner, Eshelman & Potter, 314 Wilder Bldg., Charlotte 2, N.C.
Potter, John D. ('23; '25; '35) (ACF), Box 122, 11 W. Henry St., Linden, N.J.
Potter, John Robert (J'37) (BCM), Gen. Mgr., East. Electronics Corp., 41 Chestnut St., New Haven, Conn.
Potter, Linn Edward (J'38) (BDLMNY), Pres., Gen. Mgr., Syracuse Spec., Mch. Co., Chimes Bldg., Syracuse, N.Y.
Potter, Philip Arthur ('37) (HOYLJ), Water Engr., N.J. Bd. of Pub. Utility Commrs., 1060 Broad St., Newark; *for mail*, 156 Sheridan Ave., Hohokus, N.J.
Potter, Philip J. (J'42) (SOEFK), Assoc. Prof. Mech. Engrg., Univ. of Kan., Lawrence, Kan.
Potter, Robert (J'37) (JC), E. W. Bliss Co., Salem, Ohio.
Potter, William Henry ('42) (CMOJTW), Supvr., McClure, Hadden & Ortman, 111 Washington, Chicago 2; *for mail*, 1535 E. 60th St., Chicago 37, Ill.
Pottharst, John Emile, Jr. (J'41) (BCEHKS), Partner, Gen. Mgr., Ch. Engr., Mech. Equip. Co., 861 Carondelet St., New Orleans 13, La.
Potts, Lawrence Donald (J'38) (BNLK), Research Engr., Linde Air Products Co., East Park Dr., Tonawanda; *for mail*, 711 Parker Blvd., Kenmore 17, N.Y.
Potts, Matthew William ('42) (CD), Matls. Handling Consultant, 134 Huguenot St., New Rochelle, N.Y.

Potts, William Kenneth ('20; '21; '35) (WHC), Partner, Business Mgr., Sills Bros., Box 741; *for mail,* Box 645, Auburn, Calif.
Poultney, J. Livingston ('08), Retired; Bryn Mawr, Pa.
Powel, Charles Alfred ('43) (EHSC), Mgr., Hdq. Engrg. Depts., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Powell, Clarence Earl ('29) (ACHRY), Mgr., Mchy. Div., R. D. Wood Co., 400 Chestnut St., Philadelphia 5, Pa.
Powell, Clarence Henry ('45) (ABE), Box 19, Amityville, N.Y.
Powell, Eben Morris (A'44), Mgr., York Office, Westinghouse Elec. Corp., 137 S. George St., York, Pa.
Powell, Edwin Burnley ('04; '12) (SFKJEB), Cons. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Powell, Elno M. (J'44), Asst. Dir., Engrg. Calculating Div., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Powell, James Alexander ('27) (DFKOS), Ch. Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Powell, John Aldrich (J'39) (CMJA), Gen. Foreman, Gilbert & Barker Mfg. Co., Union St.; *for mail,* 19 Sheridan Ave., West Springfield, Mass.
Powell, John Marvin (J'41) (CSYLA), Tech. Engr., Steam Design, Gen. Elec. Co., 920 Western Ave., Lynn 3, Mass.
Powell, Knox Archibald ('23; '25; '33) (MCNJDE), Shop Research Engr., Minneapolis-Moline Power Implement Co., Box 1050, Minneapolis; *for mail,* 1635 E. River Terrace, Minneapolis 14, Minn.
Powell, Morgan A. (J'41), Republic Plant, Vulcan Detinning Co., P.O. Branch 25, Pittsburgh, Pa.
Powell, Oliver I. (J'34) (CKL), Dir. of Engrg., Brewing Corp. of Am., Cleveland 6, Ohio.
Powell, Paul Rulison ('15; '23) (NMAC), Mech. Engr., West. Elec. Co., Inc., Broening Highway, Baltimore; *for mail,* 205 Stony Run Lane, Baltimore 10, Md.
Powell, Robert James (J'42), 1834 Edgewater, Muskegon, Mich.
Powell, Russell Vernon (J'38) (MDCJTR), Design Engr., Truitt Mfg. Co., Box T-1; *for mail,* 411 Woodlawn Ave., Greensboro, N.C.
Powell, Shannon Curtis (J'41) (SBKJH), Research Engr., Research & Devel. Branch, Cent. Tech. Dept., Shipbldg. Div., Bethlehem Steel Co.; *for mail,* 29 Edison Park, Quincy 69, Mass.
Powell, Sheppard Tappen ('39; F'44) (SZTKLO), Cons. Chem. Engr., 330 N. Charles St., Baltimore 1, Md.
Powell, William, Jr. (J'27) (SZF), Sales Serv. Engr., Bailey Meter Co., 403 City Centre Bldg., Philadelphia 7; *for mail,* 429 Greenview Lane, Llanerch, Pa.
Powell, William Thomas ('31; '35) (CMJNEL), V.P., Charge of Operas., Emsco Derrick & Equip. Co., 6811 S. Alameda St., Los Angeles 1, Calif.
Power, John Ambrose ('33) (RSEN), Retired; 3719 Audubon Pl., Houston 6, Tex.
Powers, James Henry ('37) (BCMNJH), Designing Engr., Gen. Elec. Co., 1285 Boston Ave., Bridgeport 2, *for mail,* R.D. 3, N. Church St., Westport, Conn.
Powers, Louis John (J'42) (NEAB), Prof. Mech. Engrg., Tex. Tech. College, Lubbock, Tex.
Powers, Michael Matthew (J'44) (ABLNMC), V.P., Gen. Mgr., P & M Mch. Co., 130-36 Prospect; *for mail,* Box 126, 413 Burg St., Granville, Ohio.
Powers, Ralph D. (J'43) (LAB), Part Owner, Mgr., Poway Dairy, Box 625, Route 1, Escondidio, Calif.
Powers, William Joseph (J'44) (F), Lt. (j.g.), U.S.N.R., Engr. Officer, LST-475, F.P.O., San Francisco, Calif.; *for mail,* 66 Price Ave., Lansdowne, Pa.
Poyser, John Reed, Jr. (J'39) (CDMJ), Methods Engr., Chain Belt Co., 1600 W. Bruce St., Milwaukee; *for mail,* 6903 W. Wells St., Milwaukee 13, Wis.
Poyser, Stanley John ('45) (NKBGWJ), Turbine Serv. Engr., Westinghouse Elec. Corp., 3451 E. Marginal Way, Seattle: *for mail,* 12348 Riviera Pl., N.E., Seattle 55, Wash.
Pozarycki, Henry Joseph (J'41) (SMCKAN), Lab. Technician, Westinghouse Elec. Corp., Pacific Ave.; *for mail,* 372 Bramhall Ave., Jersey City 4, N.J.
Pozniak, Victor (J'40) (AJM), Supercharger Engr., Gen. Elec. Co., 920 Western Ave., Lynn, Mass.
Pradl, George ('30; '37) (CJKLFD), Asst. Mgr., Canadian Copper Refiners Ltd., Box 489, Place-D'Armes Sta., Montreal; *for mail,* 4396 Coolbrook Ave., Montreal 28, Que., Can.
Prageman, Irving Henry ('26; '35) (ABCHMN), Prof. Mech. Engrg., Univ. of Me.; *for mail,* Box 173, Orono, Me.
Prager, William ('43) (BZA), Prof. Applied Mechanics, Brown Univ., Providence 12, R.I.
Pragst, Ernest ('28) (SHEK), Elec. Engr., Internatl. Gen. Elec. Co., Inc., Schenectady 5, N.Y.
Prandoni, Joseph Francis ('42) (BNSRC), Mech. Engr., Gen. Spring Corp., 500—5th Ave., New York 18, N.Y.; *for mail,* 121 Oakland St., Englewood, N.J.
Praner, Joseph Alexander ('32; '43) (CMO), 327 Tremont Ave., East Orange, N.J.
Prange, Charles Henry ('19; '23; '35) (CJMYA), V.P., Austenal Labs., Inc., 224 E. 39th St., New York 16, N.Y.
Prass, Herman ('26; '39) (KLNSZD), J. O. Ross Engrg. Corp., 350 Madison Ave., New York 17; *for mail,* 75 Prospect Park, S.W., Brooklyn 15, N.Y.
Prather, James Carter ('44) (CJMOBD), Operas. Supt., Rich's Inc., Box 4539, Atlanta 2; *for mail,* 663 Yorkshire Rd., N.E., Atlanta, Ga.
Pratt, Auguste G. ('15) (CFS), Pres., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* 345 Walnut St., Englewood, N.J.
Pratt, Bill Robinson (J'41) (JKM), 803—10th St., Perry, Okla.
Pratt, Charles A. ('27) (CFH), Hotel Sherry, 1725 E. 53rd St., Chicago 15, Ill.
Pratt, J. H. ('18; '40) (CMJF), Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Pratt, Orville R. ('42) (KFHS), Ch. Mech. Engr., U.S. Supply Co., 901 Farham St., Omaha 8; *for mail,* 5519 Mason St., Omaha, Neb.
Prable, Norman H. ('29) (CDJ), Exec. Asst., Detroit Harvester Co., 5450 W. Jefferson St., Detroit; *for mail,* 471 Rivard Blvd., Grosse Pointe, Mich.
Prechtel, William Richard (J'45) (BJY), Engr., Gates Rubber Co., 919 S Broadway; *for mail,* 720 Columbine St., Denver, Colo.
Preising, William Joseph ('45) (BJN), Product Designer, B. F. Avery & Sons Co., 1721 S. 7th St., Louisville 8, Ky.
Prendergast, Wm. A. (J'35) (MR), Mech. Engr., Research Labs., Tex. Co., Beacon; *for mail,* P.O. Box 342, Fishkill, N.Y.
Prentice, Donald Bishop ('13; '21; '24) (CMS), Pres., Rose Poly. Inst., Terre Haute, Ind.
Prentiss, John H. (J'43) (OCNMAJ), Gen. Partner, Engr., John H. Prentiss & Associates, 26 W. 58th St., New York 19, N.Y.
Presbrey, Otis ('43), 1156 E. 38th St., Brooklyn, N.Y.
Presby, Leroy Quinby ('20) (BGSAY), Supt., V.P., Am. Stay Co., 299 Marginal St., East Boston 28; *for mail,* 50 Trenton St., Melrose 76, Mass.
Prescott, Harold Edward ('42) (KLM), Ch. Engr., Golden State Co. Ltd., 425 Battery St., San Francisco, Calif.
Prescott, Perley Ralph ('18) (CDFKL), Mgr., Solvent Mfg. Co., 490 High St.; *for mail,* 605 Beech St., Holyoke, Mass.
Prescott, Robert William (J'42), Box 36, Kittery Point, Me.
Presdee, John James ('26; '35) (BCFSKY), Squad Leader, Engrg., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 83 Maple Ave., Tuckahoe 7, N.Y.
Presedo, Joseph (J'42) (BSMN), Pvt., QM. Corps, 9138th Tech. Serv. Unit, QM. Bd., Camp Lee, Va.; *for mail,* 212 Clinton St., Brooklyn 2, N.Y.
Pressey, Ralph Warwick ('40), Ch. Engr., Pa. Hospital for Mental & Nervous Diseases, 11 N. 49th St., Philadelphia 39; *for mail,* Ashley Farm, Newton Square, Pa.
Preston, Charles Henry (J'41) (CMJN), Papec Mch. Co.; *for mail,* Box 242, Shortsville, N.Y.
Preston, Ely (J'40) (NMJSAE), Mech. Designer, Submarine Sig. Co., 178 Atlantic Ave., Boston; *for mail,* 234 N. Beacon St., Watertown 72, Mass.
Preston, Everett August ('44) (BKMNR), Ch. Engr., Wilson Engrg. Corp., Rm. 1109, 122 S. Michigan Ave., Chicago 3, Ill.
Preston, Frank W. (J'25) (SDF), Plant Engr., New Haven Pulp & Bd. Co., 259 East St., New Haven 8, Conn.
Preston, Frederick William (J'44) (ARSE), Testing Engr., Gen. Elec. Co., 1 River Rd., Schenectady 1; *for mail,* 1030 Keyes Ave., Schenectady 8, N.Y.
Preston, Herbert E. ('23) (FHS), Retired; 1053 Allengrove St., Philadelphia 24, Pa.
Preston, Walter Bonham ('28; '34; '35) (LDSCEH), Mgr., Mech. & Mun. Depts., Tex. Gulf Sulphur Co., Newgulf, Tex.
Pretot, Armand V. ('25; '32; '35) (CKJ), Mar. Dept., Johns-Manville Corp., 22 E. 40th St., New York 16, N.Y.
Preusser, Henry Miner (J'40) (ZLYB), Assoc. Indus. Technologist, West. Regional Lab., U.S. Dept. of Agric., 800 Buchanan St., Albany; *for mail,* 1137 Grizzly Park Blvd., Berkeley 8, Calif.
Prian, Vasily D. ('34; '41) (LCMN), Acting Head, Mech. Engrg. Dept., Fenn College, E. 24th St., Cleveland 15, Ohio.
Pribuss, Rudolph Carl (J'41) (CKM), Htg. Engr., C. R. Reichel Engrg. Co., 718 Natoma St., San Francisco 3; *for mail,* 25 Hilltop Rd., San Mateo, Calif.
Price, Albert M. ('99; '04), Retired; 159 Seaspray Ave., Palm Beach, Fla.
Price, Beverley Charles (J'45), Indus. Div., Timken Roller Bearing Co., Canton 6; *for mail,* 1508—27th St., N.W., Canton, Ohio.
Price, Charles Grattan, Jr. (J'40) (RCFD), 279 Newman Ave., Harrisonburg, Va.
Price, Donald Jay (J'41) (RCANH), Westinghouse Air Brake Co.; *for mail,* Box 631, Wilmerding, Pa.
Price, F. Carr ('32; '35) (LCDKSZ), Plant Engr., Victor Chem. Works, 11th & Arnold Sts.; *for mail,* 1125 Campbell Ave., Chicago Heights, Ill.
Price, Floyd Ure ('21; '35) (FHS), Sales & Serv. Engr., Rep., Peerless Pump Div., Inc., Dayton-Dowd Co. & Stock Engrg. Co., 2970 W. Grand Blvd., Detroit 2, Mich.
Price, Gerald L. ('45) (OCDKLB), Asst. Civil Engr., Natl. Aniline Div., Allied Chem. & Dye Corp., 1051 S. Park Ave., Buffalo; *for mail,* 619 Elmwood Ave., Buffalo 9, N.Y.
Price, Harry (J'31), Maj., 390th Bomb Gp. (H), A.P.O. 559, c/o Postmaster, New York, N.Y.
Price, James William (J'33) (KCHFB), Ch. Engr., Pa. Indus. Engrs., Fulton Bldg., Pittsburgh 22; *for mail,* 537 Hamilton Rd., Pittsburgh 5, Pa.
Price, Joseph ('28) (BK), V.P., Charge of Engrg., Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.
Price, Leonard Cassell ('27; '31) (EAR), Prof. Mech. Engrg., Mich. State College, East Lansing, Mich.
Price, Lewis ('24; '35) (DJM), Engr., Charge Design, Shepard Niles Crane & Hoist Corp., Schuyler St., Montour Falls, N.Y.
Price, M. Lawrence (J'44) (NBHJM), Prof. Mch. Design, Worcester Poly. Inst., Worcester 2, Mass.
Price, Morton M. ('21) (BEFKS), Dist. Sales Mgr., Babcock & Wilcox Co., 1120 Packard Bldg., Philadelphia 2, Pa.
Price, Norman I. ('02; '21), Price, Martyn & Co.; *for mail,* Box 255, G.P.O., Sydney, New South Wales, Australia.
Price, Oleg A. (J'43) (NEBS), Mar. Engr., U.S. Maritime Comm., War Shipping Admin., 1081 Dexter Horton Bldg., Seattle 4; *for mail,* 3519 E. 90th, Seattle 5, Wash.
Price, R. B. (J'42) (DCM), Box 1585, Oakland 4; *for mail,* 681 Sycamore St., Oakland 12, Calif.
Price, Samuel Robert, Jr. (J'44), Engr., Ingersoll-Rand Co., 11 Broadway, New York 4, N.Y.
Price, Towson ('25)(JRDYKC), Pat. Atty., Westinghouse Elec. Corp., Bloomfield; *for mail,* 44 Stanford Pl., Glen Ridge, N.J.
Price, Wayne Keith (J'45) (SBKOEA), Trng. Engr., Allis-Chalmers Mfg. Co., Milwaukee; *for mail,* 1306 S. 75th, West Allis 14, Wis.
Price, William David (J'31) (ACG), Lt. Col., Army Air Forces; *for mail,* Box 243, Blacksburg, Va.
Price, William L. (J'42), Asst. to Field Engr., Boat Yard, Dravo Corp., Neville Island, Pittsburgh; *for mail,* 411 Maryland Ave., Oakmont, Pa.
Price, William Smith (A'45), Master Tech. Sgt., U.S.M.C., 277 Butler Dr., S., Midway Park, Camp Le Jeune, N.C.
Priedeman, George Walter ('21) (CJM), Pres., Treas., Minneapolis Ornamental Iron Co., 143—27th Ave., S.E., Minneapolis 14, Minn.
Priestman, Leslie Ernest (J'42) (CJMN), Indus. Engr., Towle Mfg. Co., 260 Merrimac St.; *for mail,* 433 Merrimac St., Newburyport, Mass.
Priff, George N. (J'42) (ABEFSO), Field Serv. Engr., East. Aircraft Div., Gen. Motors Corp., Edgar Rd., Linden; *for mail,* 351 W. Clay Ave., Roselle Park, N.J.
Prigozen, Harold Samuel (J'41), Inspr., Grumman Aircraft Engrg. Corp., Bethpage; *for mail,* 261 Central Ave., Lawrence, L.I., N.Y.
Primrose, John ('07) (EFKSZ), Vice-Chmn. of Bd., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Prince, David Chandler ('41) (ACRS), V.P., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Prince, David Chandler, Jr. (J'45) (AEK), Design Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 2 Union St., Marblehead, Mass.
Prince, Duffield ('16) (KPS), Designing Engr., Frank Heater Inc., Paterson; *for mail,* 188 Bellevue Ave., Upper Montclair, N.J.
Prince, Jerome Seymour (J'33) (DFO), Lt. Col., Ord. Dept.; 3155 Grand Concourse, New York 58, N.Y.
Princi, Mark Anthony ('46) (ALZ), Design Engr., Gen. Elec. Co., 40 Federal St., West Lynn, Mass.
Pringle, Edward Squire ('39; '41) (NCWMB), Ch. Engr., J. G. Wilson Corp., Box 599, Norfolk 1, Va.
Prior, John Alexander ('24; '35) (BA), Assoc. Prof. Mech. Engrg., Mech. Engrg. Dept., Univ. of Pa., 33rd & Locust St., Philadelphia 4; *for mail,* 5047 Hazel Ave., Philadelphia 43, Pa.
Pritchard, Eugene Roussell (J'42) (ACSJ), 1733 W. 105th St., Chicago 43, Ill.
Pritchard, Frederick Arthur ('38; '43) (LNZ), Specialty Engr., Bristol Co., Bristol St., Waterbury 91, Conn.
Pritchard, John Franklin ('23) (EKLS), Pres., J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Mo.
Pritchard, Willis Brown (J'40) (MJCZ), Time Study Engr., Am. Optical Co., Eggert Rd., Buffalo; *for mail,* 367 Millicent St., Buffalo 15, N.Y.
Pro, George M. ('38; '45) (NMCJ), Vendo Co., 1907 Grand Ave., Kansas City; *for mail,* 8425 Lee Blvd., Kansas City 5, Mo.
Probst, Joseph Rother ('35; '46) (TCNLMS), Project Engr., E. I. du Pont de Nemours & Co., Du Pont Blvd.; *for mail,* 720 Highland Ave., Waynesboro, Va.
Prochazka, Frank John ('45) (CEH), Design Engr., Racine Tool & Mch. Co., 1760 State St.; *for mail,* Box 354-A, Route 1, Racine, Wis.
Proctor, George Nelson ('26; '30; '35) (CMNJH), Mgr. of Mfg., Permutit Co., 330 W. 42nd St., New York 18, N.Y.; *for mail,* 44 Chestnut Rd., Verona, N.J.
Proctor, Redfield ('05; A'06), Pres., Vt. Marble Co., Proctor, Vt.

Proctor, Wm. E. (A'22), Pres., Proctor Engrg. Co., 106 Key Highway, Baltimore, Md.

Proescholdt, Walter H. ('37) (DEFLS), Asst. Ch. Engr., Quaker Oats Co., 141 W. Jackson Blvd., Chicago 4; *for mail*, 4146 Ellington Ave., Western Springs, Ill.

Proffitt, Russell P. ('30) (ACMS), Chicago Div. Mgr., Timken Roller Bearing Co., 2534 S. Michigan Ave., Chicago 16, Ill.

Profita, Joseph Carl (J'45) (CMN), Research Staff Mem., Turbo Lab., Bldg. 31-A, Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge; *for mail*, 20 Malvern Ave., West Somerville 44, Mass.

Prohammer, Frederick George (J'44), Graduate Asst., Univ. of Pa., Philadelphia, Pa.; *for mail*, 646 St. Ann's Ave., New York 55, N.Y.

Prohaska, James John (J'38) (LMD), Devel. Engr., Swift & Co., Union Stock Yards, Chicago; *for mail*, 12233 S. 75th St., Palos Heights, Ill.

Prohaska, Raymond Joseph (J'43) (SBAHEF), 1st Lt., Corps of Engrs., U.S. Engr. Office; *for mail*, 815 W. Lead, Albuquerque, New Mex.

Prohl, Melvin Albert (J'43) (BNSJ), Engr., Gen. Elec. Co., 920 Western Ave., West Lynn 8; *for mail*, 28 Sunset Rd., Stoneham 80, Mass.

Proper, Anthony F. (J'37), Lt., 2nd Sig. Co., (Depot), Ft. Lewis, Wash.

Propper, Clayton Harwood ('46), Pres., Propper Products Co.; *for mail*, 453 Rockwell St., Kent, Ohio.

Prosser, Jos. G. ('91; '98), 865 S. Grand Ave., Pasadena, Calif.

Prosser, Robert Gordon (J'44) (CMKOW), Sales Engr., Trane Co., 850 Cromwell, St. Paul 4; *for mail*, 4028 Upton Ave. S., Minneapolis 10, Minn.

Prosser, Roger D. ('30; '35) (WJMNZ), Propr., Thomas Prosser & Son, 120 Wall St., New York 5, N.Y.

Prostrednik, Eugene Joseph ('44) (COMNBD), Ch. Engr., Mch. & Tool Designing Co., 1011 Chestnut St., Philadelphia 7; *for mail*, 920 Fanshawe St., Philadelphia 11, Pa.

Protiva, Albert William (J'43), Irving, Kan.

Prout, Ed. R. (J'39) (SMC), Asst. Effic. Engr., San Diego Gas & Elec. Co., 6th & E Sts., San Diego 12, Calif.

Prouty, Frank Harrison ('25; '42; 'F'45) (CFORSZ), *Manager*, '37-'40, *Vice President*, '40-'42; Partner, Prouty Bros. Engrg. Co., 10th Floor, Exch. Bldg., Denver 2, Colo.

Prowell, Roy Walters ('45) (HBSEFK), Asst. Prof. Mech. Engrg., Carnegie Inst. of Tech., Schenley Park, Pittsburgh 13, Pa.

Prudden, Norman P. (J'39) (BOS), 115 Radford St., Yonkers 5, N.Y.

Prudden, Orrin Dwight ('36; '45) (SFEZLD), Supvr. Power Plants, Gen. Cable Corp., 26 Washington St., Perth Amboy; *for mail*, 58 Martin Terrace, Woodbridge, N.J.

Prudden, Peter Danforth (J'42) (CJM), Lt., U.S.N.R.; 1 Sutherland Rd., Montclair, N.J.

Pruden, Worrell Franzoni ('46) (CJM), Ch. Engr., Consltd. Steel Corp., Box 6880, East Los Angeles Branch, Los Angeles, Calif.

Prud'Homme, John Worsley (J'37) (LKBZCJ), Engr., Shell Devel. Co., 4560 Horton St., Emeryville; *for mail*, 510 Colusa Ave., El Cerrito, Calif.

Pruitt, Ralph S. ('26; '32; '35) (SFEZHD), Mgr., Mech. Dept., C. M. Guest & Sons, 129 Sharpe St., Anderson, S.C.

Prupis, Robert Irving (J'46) (BMTC), Project Engr., Conmar Products Corp., 140 Thomas St., Newark 1; *for mail*, 202 Orange Ave., Irvington 11, N.J.

Prussing, Rudolph Ernst ('15) (DRS), 20 E. Cedar St., Chicago 11, Ill.

Pryde, David Russell (J'45) (SKFOCA), Mech. Engr., Maint. Sec., Pittsfield Works, Gen. Elec. Co., 100 Woodlawn Ave.; *for mail*, 45 Ridge Ave., Pittsfield, Mass.

Pryor, Fred'k L. ('00; '06), Cons. Engr., 5 Colt St., Paterson, N.J.

Ptacek, Frank (J'42) (GMS), Mech. Engr., Yarnall-Waring Co., 102 E. Mermaid Lane; *for mail*, 224 W. Durand St., Philadelphia, Pa.

Puck, Richard Fosmark (J'40) (ASK), Project Engr., Autoflight Instruments Div., G. M. Giannini & Co., 161 E. California St., Pasadena, Calif.

Puckett, Harvey R. (J'39), 1701 University Ave., Madison, Wis.

Puder, William C. ('37) (LKEFDM), Research & Devel. Labs., Socony-Vacuum Oil Co., Inc., Paulsboro, N.J.

Puffer, Samuel Ray ('45) (ABE), Aircraft Gas Turbine Engrg. Div., Gen. Elec. Co., River Works, 920 Western Ave., West Lynn, Mass.

Pugh, Achilles Henry (A'16) (ABCG), Pres., A. H. Pugh Ptg. Co., 400 Pike St., Cincinnati 2, Ohio.

Pugh, Geo. A. (J'19) (BJM), Cons. Engr., 18 N. Phelps St.; *for mail*, 221 Overhill Rd., Youngstown, Ohio.

Pugsley, Charles Silvey, Jr. (J'37) (CKZ), Gp. Head, Engr. Dept., La. Div., Stand. Oil Co. of N.J., Box 551, Baton Rouge 1, La.

Pugsley, William Henry ('40) (CFLZ), Mgr. of Engrg., Hays Corp., Box 299, E. 8th St.; *for mail*, 324 Deweys St., Michigan City, Ind.

Puidokas, Stanley Victor (J'46) (ABC), Lt. (j.g.), U.S.N.R.; *for mail*, 4806—20th Ave., Kenosha, Wis.

Puishes, Alfons (J'30) (LCO), Procter & Gamble Mfg Co., Staten Island 3, N.Y.

Pulito, Dominick Mark (J'46) (BMS), Ensign, U.S.N., 142nd Naval Constr. Bn., F.P.O., San Francisco, Calif.; *for mail*, 1154 Glenmore Ave., Brooklyn 8, N.Y.

Pullar, William Rorke ('45) (SKFZC), Tech. Mgr., J. W. Dodds Pty. Ltd., 612 Collins St., Melbourne, C. 1, Victoria, Australia.

Puller, George ('19; '35), 82 Park Ave., Port Washington, L.I., N.Y.

Pulley, Frank L., Jr. (J'44) (KNHBZ), Engr., Devel. Dept., Carrier Corp., 300 S. Geddes St., Syracuse 1; *for mail*, 412 Greenwood Pl., Syracuse 10, N.Y.

Pumphrey, Keith Franklin (J'45), Welding Engr., Big Three Welding Equip. Co.; *for mail*, 1300 Thomas Pl., Ft. Worth, Tex.

Punnett, Frazer Doughty (J'34) (NB), Mch. Designer, Gleason Works, 1000 University Ave., Rochester; *for mail*, 200 Elmcroft Rd., Rochester 9, N.Y.

Punta, Ralph Charles, Jr. (J'45) (MOCARD), Plant Layout Engr., Electro-Motive Div., Gen. Motors Corp., La Grange; *for mail*, 3636 N. St. Louis Ave., Chicago 18, Ill.

Punton, Charles W. ('43) (CAMY), Works Mgr., Mine Safety Appliances Co., 201 N. Braddock Ave., Pittsburgh 8, Pa.

Purcell, Thomas Edward ('33) (SFCE), *Vice-President*, '42-'43; Gen. Supt., Power Stas., Duquesne Light Co., 435—6th Ave., Pittsburgh 19, Pa.

Purdie, David James ('21; '35) (HLS), Dist. Sales Mgr., Builders-Providence Inc., 20 Vesey St., New York 7, N.Y.; *for mail*, 75 Personette Ave., Verona, N.J.

Purdum, Ward Cumberland (J'45) (SNCHBA), Engrg. & Exec. Officer, U.S.N.R.; *for mail*, 520 S. Scott St., New Orleans 19, La.

Purdy, Albert Arthur (J'41) (NMLZJB), Mech. Engr., Jackson & Moreland, Park Square Bldg., Boston 16; *for mail*, 78 Safford St., Wollaston 70, Mass.

Purdy, Donald Frederick ('22; '25; '33) (ABCEFH), Box 180, Route 1, Orange Cove, Calif.

Purdy, Frank Albert (J'44) (BFSNLA), Engr., Asst. to Mar. Engr., Ore. Shipbldg. Corp.; *for mail*, 7034 N. Wilbur Ave., Portland 3, Ore.

Purdy, Henry T. ('09), Mgr., Power of Atty., H. T. Purdy, Inc., Box 750, San Jose, Costa Rica, C.A.

Purdy, Jas. B. (J'39), Test Engr., Weirton Steel Co., Weirton; *for mail*, 300 Mildren Ave., Hollidays Cove, W.Va.

Purdy, John L. (J'35) (BJS), Welding Foreman, Buckeye Steel Castings Co., S. Parsons Ave., Columbus; *for mail*, 1333 W. 1st Ave., Columbus 8, Ohio.

Purdy, Randell B. ('25; '32) (ENS), Engrg. Editor, Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4, N.Y.; *for mail*, 176 Beechwood Dr., Packanack Lake, N.J.

Purdy, William F., Jr. (J'38) (BCD), Capt., Ord. Dept., U.S.A., Detroit Ord. Dist., Detroit, Mich.; *for mail*, 122 Alnwick Rd., Malverne, L.I., N.Y.

Purinton, Dexter Jennison ('43) (ACM), 124 E. 24th St., New York 10, N.Y.

Purinton, Forrest Granville ('24) (NMCB), Research Engr., Pat. Button Co., Brown St., Waterbury 88, Conn.

Purkayastha, Promode Behari ('45), Ch. Inspr. of Boilers, Govt. of Assam; *for mail*, Laitumkhra, Shillong, Assam, India.

Pursel, Harold R. ('21; '26; '35) (CNKHAB), Univ. Club, 16th & Locust Sts., Philadelphia 2, Pa.

Pursell, William Updyke ('45) (RCEFOH), Equip. Engr., Lane Constr. Corp., 37 Colony St., Meriden 1, Conn.

Purves-Smith, Claude Dale ('43) (BHCM). 102 Highland Blvd., Berkeley 3, Calif.

Purvin, B. R. ('44) (BHM), 2221 Atkinson, Detroit 6, Mich.

Purvis, Edward D. (J'42), York & Jay Sts., Brooklyn, N.Y.

Pusnikas, Kazys Anthony ('28; '35) (CDJMN), Engrg. & Design, Natl. Can Corp., 5208 Grand Ave., Maspeth; *for mail*, 85-09—88th St., Woodhaven 21, L.I., N.Y.

Putman, Raymond S. (J'38) (B), Jr. Engr., Drafting, Naval Gun Factory, 11th & M Sts., S.E.; *for mail*, 2955 Carlton Ave., N.E., Washington, D.C.

Putnam, Abbott Allen (J'42) (EKBAHN), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail*, 219 Regent St., Hampton, Va.

Putnam, James Russell ('13) (MNZJS), Dir. of Engrg., U.S. Time Corp., 31 Cherry Ave., Waterbury 91, Conn.

Putnam, John A. ('45) (HBKZAE), Asst. Prof. Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.

Putryae, Edward Joseph ('42) (REACG), 1st Lt., 716th Ry. Opera. Bn., Ft. Sam Houston, Tex.; *for mail*, 651 Orizaba Ave., San Francisco 12, Calif.

Putt, J. William (J'39) (CBMJDN), Ch. Engr., Asst. Gen. Mgr., Hahn Motors, Inc., 400-60 S. 3rd; *for mail*, R.D. 3, 638 N. 5th, Hamburg, Pa.

Putz, Thomas John (J'46) (RSCB), Steam Div., Westinghouse Elec. Corp., Lester P.O., Philadelphia, Pa.

Putzar, Ralph David ('39) (BCKNZ), Asst. Mchy. Supt., United Engrg. Co., 500 Beale St., San Francisco; *for mail*, 382 Perkins St., Oakland 10, Calif.

Pyeatt, Frederick Ewing, Jr. ('45) (HEOM), Ch. Engr., Mid-Continent Pipe Line Co., Box 381, Tulsa 2, Okla.

Pyle, Ross S. (J'41) (BMZA), Mech. Design Engr., Collins Radio Co., 855—35th St., N.E.; *for mail*, 3011 O Ave., N.E., Cedar Rapids, Iowa.

Pyne, Frank Saxton (J'45) (HKS), Exper. Test Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail*, 28 Union St., Montclair, N.J.

Pyne, Frederick S. (J'39) (NMCBJA), 25 Chatham Rd., Longmeadow, Mass.

Pyster, J. N. (A'19), 4701 Lincoln Ave., Oakland 2, Calif.

Q

Quackenbush, Claude Fulton ('21; '25; '35) (BEA), Mech. Engr., Boeing Aircraft Co., 7755 E. Marginal Way; *for mail*, 3131—37th Pl., Seattle 44, Wash.

Quackenbush, E. Schuyler ('17; '25), Sales Engr., Indus. Washing Mch. Corp., 289 Burnet St., New Brunswick, N.J.; *for mail*, 635 Riverside Dr., New York 31, N.Y.

Quackenbush, Howard Menlo (J'44) (AMSCBE), Jr. Engr., Prod., Prest-O-Lite Inc.; *for mail*, 4965 W. 13th St., Speedway 6, Ind.

Quackenbush, Victor Charles (J'45) (CAEM), Test. Engr., Static Test Lab., N. Am. Aviation Inc., Imperial Ave., Inglewood; *for mail*, 1784½ Malcolm Ave., West Los Angeles 24, Calif.

Quackenbush, Willis Roland (J'38) (MJCDBL), West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail*, 950 Jackson Ave., Elizabeth 4, N.J.

Quaid, James Alexius ('45) (JMLCND), Dist. Mgr., Pressed Steel Co., 1033-A Broad St. Sta. Bldg., Philadelphia 3; *for mail*, 323 Hamilton Rd., Merion, Pa.

Quarles, Frank Williams ('21; 35) (SFKHZB), Asst. to Supt., Power Prod. Stas. Dept., Consltd. Gas, Elec. Light & Power Co., Lexington Bldg., Baltimore 3; *for mail*, 4040 W. Hayward Ave., Baltimore 15, Md.

Quarnstrom, Arthur A. (J'30) (LBKH), Asst. Engr., Natural Gas Dept., Richfield Oil Corp., Box 470, Long Beach 1; *for mail*, 3910 Falcon Ave., Long Beach 7, Calif.

Quast, Walter Flamm ('16; '26) (CFSKDH), Supvr. Prod. Costs, Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5; *for mail*, 6448 Woodcrest Ave., Philadelphia 31, Pa.

Quay, L. Lee (J'43) (YZALK), Mech. Engr., Carbide & Carbon Chemicals Corp., Box N, South Charleston 3, W.Va.; *for mail*, 304 N. Wayne Ave., Wayne, Pa.

Quayle, Alexander (J'36) (CM), Asst. Ch. Engr., Oil Well Supply Co.; *for mail*, 206 W. 1st St., Oil City, Pa.

Quayle, LeRoy Abell ('15; '17) (DHMNSY), Cons. Engr., Cleveland Quarries Co., Guild Hall Bldg., Cleveland; *for mail*, 164 Longfellow St., Elyria, Ohio.

Quealy, Lawrence Stuart (J'41) (ABC), Supt., Quealy Land & Livestock Co., Elk Mountain, Wyo.

Quentin, Charles Reath (J'45) (ABE), Designer, Gen. Elec. Co., Schenectady; *for mail*, 16 Washington Ave., Schenectady 5, N.Y.

Query, Leo Henry ('46), Prof., Univ. of Rochester, River Campus, Rochester, N.Y.

Quick, Guy C., Jr. (J'43) (OCS), Jr. Engr., United Engrs. & Constructors Inc., 1401 Arch St., Philadelphia; *for mail*, 828 Concord Ave., Drexel Hill, Pa.

Quick, Howard Prescott ('94), 125 Eaglecroft Rd., Westfield, N.J.

Quick, Ray S. ('22; '35) (HSEFC), Mgr. of Engrg., Baldwin Loco. Works, Philadelphia 42, Pa.

Quick, William Kay (J'44) (CEJ), Capt., Ord. Sec., Hdq., XXth Corps, A.P.O. 340, c/o Postmaster, New York, N.Y.

Quier, Kenneth E. ('33; '35) (BKLM), Assoc. Prof. Mech. Engrg., Pratt Inst., 215 Ryerson St., Brooklyn 5, N.Y.

Quigley, Edward Patrick (J'43), Lt., U.S.A., Hdq., 269th Replacement Co., A.P.O. 711, c/o Postmaster, San Francisco, Calif.; *for mail*, 14214 Savannah Ave., East Cleveland 12, Ohio.

Quinn, Arthur M. ('25; '35) (FS), Sales Engr., Haven Corp., 225 S. 15th St., Philadelphia; *for mail*, 630 Boyer, Cheltenham, Pa.

Quinn, Bayard Elmer (J'37) (NBEHAR), Asst. Prof., Cornell Univ.; *for mail*, 527 E. Buffalo St., Ithaca, N.Y.

Quinn, George Francis (J'44) (KBLFSJ), Instr., Dept. of Chem. Engrg., Columbia Univ., New York 27, N.Y.

Quinn, Joseph J., Jr. ('39) (CHL), U.S. Navy, 1106 E. Haines St., Germantown, Philadelphia, Pa.

Quinn, S. M. ('40) (EFS), Engr., Halifax Ins. Co., 88 Hollis St., Halifax, N.S., Can.

Quinnelly, James L. (J'41) (FK), 3801—8th St., Meridian, Miss.

Quintero, Calixto E. ('40), c/o Puerto Rico Water Resources Authority, Ponce, P.R.

Quintilian, Bartholomew Frank (J'42), May Oil Burner Corp., Maryland Ave. & Oliver St., Baltimore, Md.

Quirk, Clinton Henry ('17; '35) Mech. Engr., Pub. Works Div., 3rd Naval Dist., U.S.N., 90 Church St., New York; *for mail*, 465 Front St., Hempstead, N.Y.

Quirk, John Henry (J'42) (ABJ), Abrasive Engr., Carborundum Co., Box 337; *for mail,* Apt. 4, 359—1st St., Niagara Falls, N.Y.
Quirke, Edward D. ('11) (FKLMNS), Adv. Mgr., Kewanee Boiler Corp., Franklin St. & Q Tracks; *for mail,* 730 S. Chestnut St., Kewanee, Ill.
Quist, Stettler Howard (J'45) (MCHNLS), Process Engr., Remington Arms Co., Ilion; *for mail,* 120 W. Albany St., Herkimer, N.Y.

R

Rabbitt, James A. ('19) (EFJS), Cons. Engr., Internatl. Nickel Co., Inc., 67 Wall St., New York, N.Y.; *for mail,* Upland Dr., Greenwich, Conn.
Rabe, Frederick William ('42) (FKOS), Cons. Engr., Box 2727, Dallas; *for mail,* Cornell 3105, Dallas 5, Tex.
Rabe, Joseph S. ('36) (NMRSBA), Elec. Engr., Wm. Sellers & Co. Inc., 1600 Hamilton St., Philadelphia 30, Pa.
Raber, Benedict Frederick ('14) (SFEKM), Prof. Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.
Rabert, Arthur Paul (J'33) (ABL), 100—13th Ave., Brooklyn Park, Baltimore 25, Md.
Rabin, Arthur (J'45) (NJYMBZ), Design Engr., Univ. Labs. Co., 225 Varick St., New York 14; *for mail,* 2386 Morris Ave., Bronx 53, N.Y.
Rabinow, David ('44) (CMNDH), Ch. Engr., Conmar Products Corp., 140 Thomas St., Newark 1; *for mail,* 41 Tuscan Rd., Maplewood, N.J.
Raborg, Joseph H. (J'42), Application Engr., Worthington Pump & Mchy. Corp., Harrison, N.J.; *for mail,* 601 Eagle Ave., New York 55, N.Y.
Rach, J. Louis W. ('19; '35), Mech. Engr., Design, Bartlett Hayward Div., Koppers Co., Scott St.; *for mail,* 1947 E. 31st St., Baltimore, Md.
Rachals, Richard ('39; '45) (CN), Engr., Gibbs & Cox, Inc., 21 West St., New York 6, N.Y.
Radach, Paul Reinhardt (J'43) (NACM), Specialist, U.S.N.R., Mech. Engr. (Design), Div. N, Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field, Calif.
Radamaker, George Lewis (J'43) (SLCMOK), Mch. Designer, Spec. Equip., Spring Div., Eaton Mfg. Co., 9771 French Rd.; *for mail,* 8050 Dobel St., Detroit 5, Mich.
Radcliff, Leo Lewis (J'41) (SOLMDE), Mech. Engr., Devel., Corn Products Refining Co., Argo, Ill.
Radcliffe, John C., III (J'39), Trans. Sales, Westinghouse Elec. Corp., 411 N. 7th St., St. Louis 1; *for mail,* 221 E. Jefferson Ave., Kirkwood 22, Mo.
Radden, Charles O. (J'36) (ES), Indus. Serv. Engr., Gulf Oil Corp., Park Sq., Boston; *for mail,* 18 Miami Ave., West Roxbury 32, Mass.
Radeck, Justin E. (J'38) (BEHJN), Engr., Salesman, Cameron Iron Works, 5125 Santa Fe St., Los Angeles 14, Calif.
Radecki, Michael J. ('30; '35) (CMBJKL), Supt., Henry G. Thompson & Son Co., 277 Chapel St., New Haven 5, Conn.
Rader, Donald Arlie (J'42) (AMNB), Engr., Layout Design, Airplane Div., Curtiss-Wright Corp., 4300 E. 5th Ave., Columbus 16; *for mail,* 851 Kenwick Rd., Columbus 9, Ohio.
Radford, G. S. ('15; F'41) (MTW), Cons. Engr., Box 426, New Canaan, Conn.
Radinse, Carl H. (J'39) (BNMH), Engr., Mfrs. Brush Co., 12501 Elmwood Ave., Cleveland 11, Ohio.
Radziminski, Whitold Wladislaw ('44) (MNRSZ), Designer, Engrg. Dept., E. I. du Pont de Nemours & Co.; *for mail,* 703 W. 22nd St., Wilmington, Del.
Rae, Otis Ott ('44) (CKMORS), Mgr., Southeast. Cent. Sta., Westinghouse Elec. Corp., 1299 Northside Dr., N.W., Atlanta, Ga.
Raech, Harry, Jr. (J'44) (ZNL), Devel. Engr., Natl. Electronics Corp., 3141 W. 110th St., Cleveland; *for mail,* 367 South Island, Rocky River, Ohio.
Raess, Theodore F. (J'42) (SKEJ), Engr., Diesel Devel. Dept., Joshua Hendy Iron Works, Sunnyvale; *for mail,* Box 142, Palo Alto, Calif.
Raettig, Alvin Ernest, Jr. (J'40) (N), 213 Winona Dr., Decatur, Ga.
Raetz, Gibbs Stanley (J'42) (ACE), Flight Test Analyst, Lockheed Aircraft Corp., Burbank; *for mail,* 1553 N. Stanley Ave., Los Angeles 46, Calif.
Raetz, Stephen James ('21) (CR), Supvr., Cars & Shops Maint., Independent Div., N.Y.C. Transit System, 3961—10th Ave., New York 34; *for mail,* C-42, Hudson View Gardens, 183rd St. & Pinehurst Ave., New York 33, N.Y.
Ragsdale, Earl J. W. ('45) (RGAJ), Ch. Engr., Ry. Div., Edw. G. Budd Mfg. Co., 2450 Hunting Park Ave., Philadelphia 32, Pa.
Rahikka, Reino Erick (J'43) (CMA), Equip. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 148 Orient Ave., Jersey City 5, N.J.
Rahm, Fenton D. ('23) (CEHL), Mgr., Houston Office, A. M. Lockett & Co. Ltd., 808 Elec. Bldg., 1016 Walker Ave., Houston 2, Tex.
Rahm, Frederick W. ('38) (MLJCB), Designer, Geo. Scher Engrg. Co., 1st St., Newark, N.J.; *for mail,* Greene Pl., New York, N.Y.
Rahm, Louis Frank ('23; '34) (Y), Assoc. Prof. Mech. Engrg., Sch. of Engrg., Princeton Univ., Princeton, N.J.
Rahm, Robert Carl ('42) (BNY), Treas., Ch. Engr., Berks Engrg. Co., 6th & Chestnut Sts., Reading, Pa.
Raife, Archie (J'44) (BCGJMN), Mech. Engr., Stewart-Warner Corp., 1826 Diversey Pky., Chicago 14; *for mail,* 5749 N. Central Park, Chicago 45, Ill.
Raimondi, Albert Anthony (J'46), Ensign, U.S.N.R., U.S.S. *Renville,* A.P.O. 227, F.P.O., San Francisco, Calif.; *for mail,* 25 Crescent St., North Plymouth, Mass.
Rainesalo, Charles Ilmari ('38) (SEM), Ch. Engr., Ajax Iron Works, 26 W. Washington St.; *for mail,* 152 Park St., Corry, Pa.
Raisch, William ('24; '34) (LO), Cons. Engr., 227 Fulton St., New York 7, N.Y.
Raisig, Charles L. ('18; '35) (BJM), Ch. Engr. of Sales, Mesta Mch. Co., Pittsburgh; *for mail,* Park Hill, R.D. 1, Coraopolis, Pa.
Raisler, Herbert (J'37) (CK), Engr., Raisler Corp., 129 Amsterdam Ave.; *for mail,* Apt. 16-B, 49 E. 96th St., New York, N.Y.
Raisler, Robert K. ('26; '33; '35) (OKESCA), Treas., Raisler Corp., 129 Amsterdam Ave., New York 23, N.Y.
Raitt, George H. ('21; '28) (C), Pres., Gen., Mgr., Steel Tank & Pipe Co. of Calif., 1100—4th St., Berkeley 5, Calif.
Raju, C. S. N. ('27; '34; '35), Deputy Indus. Adviser, "N" Block, Raisina Rd., New Delhi, India.
Rakatansky, Harold (J'40) (BES), U.S. Army; 52 Gladstone St., Providence, R.I.
Raley, Ray E., Jr. (J'45) (P), Box 197-X, Route 1, Bakersfield, Calif.
Rall, Charles Otto (J'30) (CMS), Prod. Engr., Pittsburgh Piping & Equip. Co., 10—43rd St., Pittsburgh 1; *for mail,* 210 Amber St., Pittsburgh 6, Pa.
Rall, Edwin Beresford ('41) (CFOSR), Asst. Plant Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 1344 Yorkshire Rd., Grosse Pointe Park 30, Mich.
Ralph, John Roger (J'44) (SDAR), Mech. Engr., Draftsman, Austin Co., 16112 Euclid Ave., Cleveland; *for mail,* 14737 Lorain St., Cleveland 11, Ohio.
Ralston, Alexander Hoyle ('45) (HOS), W. M. Anderson Co., 600 Schuylkill Ave., Philadelphia; *for mail,* 2452 Eldon Ave., Drexel Hill, Pa.
Ralton, Francis Arthur ('22; '23) (SKFHCT), Designing Engr., Am. Woolen Co. Inc., 1 Mill St., Lawrence, Mass.
Ram, R. Anant, Sr. ('30; '34; '35) (MST), Ch. Mech. & Elec. Engr., Hira Mills, Ltd., Ujjain, Gwalior, Central India.
Ramage, Edwin C., Jr. ('22; '27; '32) (CFL), Pinehurst Farms, Route 3, Maryville, Tenn.
Ramage, Raymond William (18; '26; '35) (NJYMR), Instr. in Mch. Design, Tech. Dept., North High Sch., E. Frederick St.; *for mail,* 71 Homer St., Binghamton, N.Y.
Ramaswami, E. K. ('29), Prof., College of Engrg., Bangalore, S. India.
Ramberg, Walter ('36) (ABJ), Sr. Physicist, Natl. Bur. of Stands., Dept. of Commerce, Connecticut Ave. at Upton St., Washington 25, D.C.
Rambo, Harold Edwin (J'42), Engr., Cardox Corp., Monee; *for mail,* 3706 Union Ave., Steger, Ill.
Ramey, John Randolph, Jr. (J'45) (HOZADN), Asst. Project Engr., U.S. Waterways Exper. Sta., Vicksburg; *for mail,* Box 35, West, Miss.
Ramey, Roy H. ('39), Serv. Engr., R.D. Div., Worthington Pump & Mchy. Corp., 224 Townsend St., San Francisco, Calif.
Ramirez, S. John (J'43) (LSKYFD), Asst. Ch. Draftsman, S. D. Hicks & Son, 1671 Hyde Park Ave., Boston 36; *for mail,* 22 Suffolk Ave., Revere 51, Mass.
Ramm, Henry F. ('36; '38) (AHL), Prod. Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford; *for mail,* R.F.D. 1, Columbia, Conn.
Ramsay, Erskine ('25), Chmn. of Bd., Gen. Cons. Engr., Ala. By-Products Corp., 1915—1st Natl. Bldg., Birmingham 3, Ala.
Ramsay, W. A. ('17), Pres., W. A. Ramsay, Ltd., Box 1721, Honolulu, T.H.
Ramsdell, Roger Gibbs, Jr. (J'41) (SHK), Engrg. Asst., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 28 Walnut Ave., Rockville Centre, N.Y.
Ramsden, John T. ('06) (ABCHJK), Exec. Ch. Engr., Tabor Mfg. Co., 6225 Tacony St., Philadelphia; *for mail* 1911 N. 17th St., Philadelphia 21, Pa.
Ramsey, Clifford Houston ('20) (BCY), Pres., John Royle & Sons, 10 Essex St., Paterson 3, N.J.
Ramsey, George ('16) (J), Partner, Ramsey, Kent & Chisholm, 11 Park Pl., New York 7, N.Y.
Ramsey, Justin Houston ('44) (YNJRKL), V.P., Sales & Engrg., John Royle & Sons, 10 Essex St., Paterson 3, N.J.
Ramson, John Richard, Jr. (J'42), Design Engr., Iowa Mfg. Co.; *for mail,* 1501—6th Ave., S.E., Cedar Rapids, Iowa.
Ranck, Clayton Ernst (J'41) (DLM), Ensign, U.S.N.R.; *for mail,* 212 St. Mark's Sq., Philadelphia 4, Pa.
Rand, Irvin Jerome ('45) (SHLKEO), Sales Engr., Ingersoll-Rand Co., 224 E. 9th St., Cincinnati 2; *for mail,* 3245 Orion Ave., Cincinnati 13, Ohio.
Rand, William Arthur (J'42) (MAJCBD), Ensign, U.S.N.R., LSM-283, F.P.O., San Francisco, Calif.
Randall, Guy B. ('22) (FLS), Cons. Engr., Champion Paper & Fibre Co.; *for mail,* 851 Gray Ave., Hamilton, Ohio.
Randall, John Foster (J'36) (JKFS), Engr., Automotive Research Dept., Ford Motor Co., Village Rd. Bldg., Dearborn; *for mail,* 14516 Terry Ave., Detroit 27, Mich.
Randall, Merwyn C. ('40), Sr. Engr., Philadelphia Elec. Co., 17th Floor, Edison Bldg., 9th & Sansom St., Philadelphia; *for mail,* 6211 Jefferson St., Philadelphia 31, Pa.
Randall, Ross D. ('31; '45) (BKM), Dept. 239, Stewart-Warner Corp., Indianapolis, Ind.
Randich, Erasmus A. (J'43) (MJBYDL), Design Draftsman, Plant Engrg. Dept., Bethlehem Steel Co.; *for mail,* 302 E. Broad St., Bethlehem, Pa.
Randle, Winslow Allhands (J'39), 1st Lt., Corps of Engrs., U.S.A., Co. A, 47th Div.; *for mail,* Port Sulphur, La.
Randolph, Frank Harrison ('22; '32) (KFOSCG), Prof. Hotel Engrg., Cornell Univ., Box 55, Roberts Hall, Ithaca, N.Y.
Raney, Russell Raymond ('45) (BCH), Asst. Ch. Engr., McCormick Works, Internatl. Harvester Co., 26th & Western Ave., Chicago 8, Ill.
Rang, Eugene ('30; '35) (S), Boiler Inspr., Hartford Steam Boiler Inspc. & Ins. Co., 56 Prospect St., Hartford; *for mail,* 466 Main St., Norwich, Conn.
Raninen, Arnold Birchard (J'44) (BMNJ), Engr., Ord. Design, Aberdeen Proving Ground; *for mail,* 228 Seneca Ave., Havre de Grace, Md.
Rank, Homer Lee ('18; '22) (FSD), Sales Mgr., Pulverizer Dept., Strong Scott Mfg. Co., Northwest. Terminal, Minneapolis 13; *for mail,* 7040 Oak Grove Blvd., Minneapolis 9, Minn.
Rankel, Robert A. (J'40) (LEARZS), Research Engr., Anemostat Corp. of Am., 10 E. 39th St., New York; *for mail,* 55 Baisley Ave., East Rockaway, L.I., N.Y.
Ranken, Howard B. ('29; '31) (BLOJHY), Professional Engr., Land Surveyor, Box 202, Edgewater, Fla.
Rankin, Andrew William ('45) (BSJN), Turbine-Generator Engrg. Div., Gen. Elec. Co., Schenectady, N.Y.
Rankin, Charles John ('45) (MJKCOS), Exec., Robert A. Rankin & Co., 1420 Sherbrooke St. W., Montreal, Que., Can.
Rankin, Charles Rule (J'45) (CBHAM), Lab. Engr., Caterpillar Tractor Co., Peoria 8; *for mail,* 526 Linn St., Peoria 5, Ill.
Rankin, Dan Richart ('45) (BHJ), Ch. Engr., Peerless Pump Div., Food Mchy. Corp., 301 W. Ave. 26, Los Angeles 31; *for mail,* 5310 Aldama St., Los Angeles 42, Calif.
Rankin, Robert Arthur ('40) (LOCSFD), Pres., Robert A. Rankin & Co., 1420 Sherbrooke St. W., Montreal, Que., Can.
Rankin, Robert L. (J'40) (EF), Jr. Mech. Engr., Shell Oil Co., Inc., 965—1st Natl. Bldg., Oklahoma City, Okla.
Ranney, Leo ('40) (EFH), Pres., Gen. Mgr., Ranney Oil Mining Co., Box 67, Morro Bay, Calif.
Ransdell, Clifford Howell (J'45) (BM), Asst. Prof., Engrg. Drawing Dept., Texas Agric. & Mech. College, Box 202 F.E., College Station, Tex.
Ransohoff, Nathan ('12; '21) (LKMRSW), Pres., N. Ransohoff, Inc., Township St., Elmwood P.O., Cincinnati; *for mail,* 3509 Biddle St., Cincinnati 20, Ohio.
Ransom, J. Frederick ('35; '44) (SBEZK), Turbine Engr., Rocky Mountain Dist., Gen. Elec. Co., Box 2331, Denver 1; *for mail,* 2531 Clermont St., Denver 7, Colo.
Ransom, John Parke (J'42), 2nd Lt., Corps of Engrs., A.P.O. 702, c/o Postmaster, Seattle, Wash.
Ransom, R. Lanning ('42) (HBCNMZ), V.P., Gen. Mgr., High Pressure Pump Corp., 620 Brooks Bldg., Wilkes-Barre; *for mail,* 26 E. Walnut St., Kingston, Pa.
Ransom, W. G. ('17), Partner, Cook & Ransom, Ottawa; *for mail,* Homewood, Kan.
Ranstead, Norman H. ('27) (KL), Designing Engr., Universal Oil Products Co., 310 S. Michigan Ave., Chicago; *for mail,* 2154 Thornwood Ave., Wilmette, Ill.
Rao, Thagarthi Nagesha (J'45), Maint. Engr., Mysore Chemicals Fertilisers, Ltd., Vanivilas Rd.; *for mail,* Belagula, Mysore, India.
Rapp, John William (J'42) (BJM), Asst. M.M., Foote Bros. Gear & Mch. Corp., 4545 S. Western Blvd.; *for mail,* 6907 S. Harper Ave., Chicago, Ill.
Rapp, Philip Crowe (J'45) (BER), Project Engr., Houde Engrg. Div., Houdaille Hershey Corp., 537 E. Delavan Ave., Buffalo 11; *for mail,* 965 West Ave., Buffalo 13, N.Y.
Rardon, Norman Charles ('44) (REZASF), Engr., Loco. Equip. Div., Manning, Maxwell & Moore, Inc., 1257 Little Bldg., 80 Boylston St., Boston 16; *for mail,* 33 Hobomack Rd., Quincy 69, Mass.
Rashevsky, Michael ('44) (BSJN), Mech. Design Engr., Westinghouse Elec. Corp., 700 Braddock Ave., East Pittsburgh; *for mail,* 1506 Foliage St., Pittsburgh 21, Pa.

Raskin, Franz Joseph M. ('44) (FKEHZL), Pres., Mgr., F. J. Raskin Inc., 370 Rachel St. E., Montreal 18; *for mail,* 2620 Pie IX Blvd., Montreal 4, Que., Can.
Rasmussen, Carl Arthur (J'39) (MCNLSA), Process Engr., Prod. Engrg. Dept., Wright Aero. Corp.; *for mail,* 566—21st Ave., Paterson 3, N.J.
Rasmussen, Find ('13) (BJH), Retired; 7807—12th Ave., Brooklyn 28, N.Y.
Rasmussen, Frank ('30) (DBJRM), Dist. Engr., Link-Belt Co., 300 W. Pershing Rd., Chicago 9; *for mail,* 735 Judson Ave., Evanston, Ill.
Rasmussen, Harold V. ('23; '30) (BES), De Laval Steam Turbine Co.; *for mail,* Cadwaleder Apts., 640 W. State St., Trenton 8, N.J.
Rasmussen, Monrad Harvey (J'41) (JBAMDC), Ensign, U.S.N.R.; 3812 N. Troy St., Chicago 18, Ill.
Rasmussen, Murlin Albert (J'43) (ASMOEF), 3824 N. Calispel, Spokane 12, Wash.
Rasmussen, Roy A. (J'39) (CDFKZ), Asst. Supt., Lone Star Cement Corp., 1217—1st Natl. Bank Bldg., Dallas; *for mail,* 2759 Overcrest, Dallas 11, Tex.
Rasmusson, William Elmo (J'43) (SHBKZ), Sr. Design Engr., Oldsmobile Div., Gen. Motors Corp., Townsend St., Lansing; *for mail,* 302 Leslie St., Lansing 12, Mich.
Ratcliff, Vern H. (J'34) (SHFC), Ch. Engr., Englewood Y.M.C.A., 6545 S. Union Ave., Chicago 21, Ill.
Ratcliffe, Frank Damon ('45) (CEFHS), Dist. Mgr., Oil Industry Sales, Fairbanks, Morse & Co., 1335 Hunt Bldg., Tulsa 3, Okla.
Ratcliffe, Fred'k R. ('22) (CDM), Cons. Engr., 36 Woodside Ave., Strathfield, Sydney, New South Wales, Australia.
Rath, Robert Earle (J'46), 200 Park Ave., Swarthmore, Pa.
Rathaus, Joseph T. (J'46) (ABO), Dynamic Test Engr., Chance-Vought Aircraft Div., United Aircraft Corp.; *for mail,* 268 West Ave., Stratford, Conn.
Rathbone, Thomas Cosnett ('29) (BSEH), Ch. Engr., Mchy. Div., Fidelity & Casualty Co. of N.Y., 80 Maiden Lane, New York 7; *for mail,* 5-H, 2 Grace Court, Brooklyn 2, N.Y.
Rathburn, John Altmeier (J'42) (SKHJN), Sgt., Army Air Forces, Wright Field, Dayton; *for mail,* 4 Shopping Court, Dayton 3, Ohio.
Rathe, Alex W. ('43; '46) (CDM), Indus. Engr., Box 42, Ampere, N.J.
Rathgeb, Albert ('26; '35), 734 Park Ave., Plainfield, N.J.
Rathman, Gilbert ('25) (EHY), 149 Indian Run Park, Union, N.J.
Rattenbury, David James (J'41) (BJNAW), Ch. Draftsman, Canadian Sumner Iron Works, 3550 E. Broadway; *for mail,* 3756 W. 3rd Ave., Vancouver, B.C., Can.
Ratushinsky, Walter Gillus (J'41), 5218 Delmar Blvd., St. Louis 8, Mo.
Ratzel, Elmer Alfred (J'46), Assoc. Engr., Armour Research Foundation, 35 W. 33rd St., Chicago 16; *for mail,* 8532 Ashland Ave., Chicago 20, Ill.
Raub, Jonas Heartt ('34; '44) (NBJMH), Engr., Waterbury Farrel Fdy. & Mch. Co., 453 Bank St., Waterbury 86; *for mail,* Good Hill Rd., Woodbury, Conn.
Rauch, Carl ('44) (FKS), 596 Riverside Dr., New York 31, N.Y.
Rauch, Ralph T. (J'36) (EMS), Clem Lumber Co., 2500 Live Oak St., Dallas; *for mail,* 3102 Ivandell Ave., Dallas, 11, Tex.
Rausch, Victor H. ('42) (AMR), 14 Coolidge Rd., Maplewood, N.J.
Rautenstrauch, Robert Franklin (J'38) (A), Assoc. Prof. Aero. Engrg., N.C. State College, Raleigh, N.C.
Rautenstrauch, Walter ('04; '11) (CDM), Prof. Emeritus, Columbia Univ., 117th St. & Broadway, New York 27, N.Y.; *for mail,* 235 Dorin Court Rd., Palisade, N.J.
Ravese, Thomas (J'39) (FSKZ), Field Test Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 4 Fairview Pl., Port Chester, N.Y.
Ravnsbeck, Fred W. ('41) (FKN), Ch. Engr., Ace Engrg. Co., 1435 W. 15th St., Chicago, Ill.
Rawson, Arthur J. ('28; '31; '35) (NBCZ), Assoc. Prof. Biological Engrg., Johnson Foundation, Univ. of Pa., 36th & Spruce Sts., Philadelphia 4; *for mail,* 519 S. Orange St., Media, Pa.
Rawson, Ralph Harvey ('45) (WKDLF), Cons. Timber Engr., 1206 Yeon Bldg., Portland 4; *for mail,* 2750 S.W. Fern St., Portland 1, Ore.
Ray, David H. ('04; '11) (BCGO), Retired; 1235 S. 1st Ave., Arcadia, Calif.
Ray, Thomas ('23) (HKL), Pres., Manistee Iron Works Co.; *for mail,* 339—2nd St., Manistee, Mich.
Ray, William Robert (J'41) (SFNZB), Supvg. Inspr., Hartford Steam Boiler Inspc. & Ins. Co., 1040 Leader Bldg., Cleveland 14; *for mail,* 177 Carrol Ave., Painseville, Ohio.
Raye, Alexander H. (J'40) (ABEM), Elliott Co., Jeannette; *for mail,* 625 Welty St., Greensburg, Pa.
Rayle, Roy E., Jr. (J'38) (JMR), Maj., Hdq., XIIth Tactical Air Comd., A.P.O. 374, c/o Postmaster, New York, N.Y.
Raymond, Allen Arthur ('31) (RFKC), N.Y. Cent. R.R., 1301 Cent. Terminal, Buffalo, N.Y.
Raymond, Allen Arthur, Jr. (J'46) (C), 154 Nassau St., New York 7, N.Y.
Raymond, Ernest P. ('42) (BHMN), Engrg. Stand. Mchy. Div., Ex-Cell-O Corp., Oakman Blvd., Detroit 6; *for mail,* 18231 Snowden Ave., Detroit 21, Mich.
Raymond, Eugene Thomas (J'46) (BCK), Ensign, U.S.N.R., 4117 California Ave., Seattle 6, Wash.
Raymond, Fairfield Eager ('21; '26; '35) (CMS DJZ), 28 Meadow Way, Cambridge 38, Mass.
Raymond, Gwynne ('30) (CMNEJW), V.P., Mfg. & Engrg., Black Sivalls & Bryson Inc., Box 1714, Oklahoma City; *for mail,* 3110 N.W. 20th St., Oklahoma City 7, Okla.
Raymond, Jack Darrel (J'43) (AJKFRC), Box 240, Marshfield, Mass.
Raymond, Leonard ('39) (EBFAS), Asst. Supvr., Automotive Lab., Socony-Vacuum Oil Co., Inc., Paulsboro, N.J.
Raymond, Thomas Emmons (J'40) (HB), Pres., Simplex Engrg. Co., Newark Rd.; *for mail,* 964 Findley Ave., Zanesville, Ohio.
Raymond, Ward ('01) (CJM), Pres., Pa. Pump & Compressor Co.; *for mail,* 908 Paxinosa Ave., Easton, Pa.
Raynes, S. Herbert, Jr. (J'42) (BJM), Instr. in Mech. Engrg., Drexel Inst. of Tech., 32nd & Chestnut St., Philadelphia; *for mail,* 581 Pedley Rd., Philadelphia 28, Pa.
Raynor, Arthur Edward ('43) (SEKFC), Sec. Leader, Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Rea, Joseph T. ('28) (BCFJN), Constr. Engr., Bethlehem Pacific Coast Steel Co., Los Angeles, Calif.
Rea, Ralph Goodman ('44) (SFLO), Ch. Engr., Power House, Ohio Oil Co., E. Main St.; *for mail,* 503 E. Emmons St., Robinson, Ill.
Rea, Thatcher W. ('38) (CFS), Sales Engr., Detroit Stoker Co., 5th Floor, Gen. Motors Bldg., Detroit, Mich.
Rea, William Edward ('38) (EH), c/o Williams Bros. Corp., 30 Rockefeller Plaza, New York 20, N.Y.
Read, Carleton Allen ('93; '00) (FEKS), Cons. Engr., 15 Hackfeld Rd., Worcester 2, Mass.
Read, Miles H. ('26) (HOZ), Asst. Ch. Engr., Simplex Valve & Meter Co., 68th & Upland Sts., Philadelphia 42; *for mail,* 1331 S. Lindenwood St., Philadelphia 43, Pa.
Reagan, Leon Smith ('42) (FKE), V.P., Gen. Mgr., Webster Engrg. Co., 419 W. 2nd St., Tulsa, Okla.
Reagor, Arthur, Jr. ('40) (OFSLZK), Mech. Engr., Bldg. Design, Geo. L. Dahl, Archs., Engrs., 1920½ Main St., Dallas 1, Tex.
Reamy, William Clarence, Jr. ('44) (ZSKHF), Partner, Meleney Engrg. Co., 720 Mills Bldg., Washington 6, D.C.
Reaney, Ernest ('44) (CJMN), Ch. Engr., O K Tool Co., Div. of Aerodynamic Research Corp., 33 Hull St., Shelton; *for mail,* 4709 N. Main St., Stratford, Conn.
Rearick, Charles B. ('92; '01) (EJS), Propr., Firm of Charles B. Rearick, 30 Church St., New York 7, N.Y.
Rearick, Walter S. ('41) (C), 1021 Parkview Ave., New Kensington, Pa.
Reaser, Wilbur Wilson (J'36) (KACB), *Spirit of St. Louis Junior Award,* '41; Air Conditioning Gp. Engr., Douglas Aircraft Co., Inc., 3000 Ocean Pk. Blvd., Santa Monica; *for mail,* 16018 Junaluska Way, Pacific Palisades, Calif.
Reaser, William E. ('35; '41) (FSL), Assoc. Prof. Mech. Engrg., Lafayette College, Easton, Pa.
Rebaleati, Joseph Norman (J'45), U.S. Navy; Box H, Eureka, Nev.
Reber, Clarence George (J'41) (MABSRJ), Project Engr., Plax Inc., 133 Walnut St., Hartford; *for mail,* 350 Holcomb St., Hartford 5, Conn.
Reber, Louis E. ('91; F'43) (ACP), Retired; 242 Lakeland Dr., West Palm Beach, Fla.
Rech, Herbert F. ('27) (AFKRSY), Dir., Power Sec., Gen. Motors Corp., Gen. Motors Bldg.; *for mail,* 1246 Edison Ave., Detroit 2, Mich.
Rechtien, Henry Charles, Jr. (J'42) (AC), 641 Post Pl., East St. Louis, Ill.
Recknagel, Reed Truman (J'43) (AC), Ensign, U.S.N.R.; The Albotsford, 186 Commonwealth Ave., Boston, Mass.
Redden, Clarence Albert (J'40) (SWHALN), Ch. Engr., Hollingsworth & Whitney Co.; *for mail,* 5 Herd St., Waterville, Me.
Reddert, Earl J. ('22; '25; '31) (CLSZDO), Ch. Engr., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City; *for mail,* 94 Green Ave., Madison, N.J.
Reddick, Marshall E. ('26; '32; '35) (SEFLZO), Mech. Engr., Stearns-Roger Mfg. Co., 1720 California St., Denver 2, Colo.
Redding, Arnold Hurxthal (J'44) (ABSEMK), Sec. Engr., Westinghouse Elec. Corp., Lester; *for mail,* 227 Park Ave., Swarthmore, Pa.
Reddy, Dermot (J'38) (LMT), Celanese Corp. of Am., Cumberland; *for mail,* Long Post Rd., La Vale, Md.
Rede, G. Ross ('45) (ABC), Devel. Engr., Gen. Engrg. & Consulting Lab., Gen. Elec. Co., Schenectady; *for mail,* 353 State St., Albany, N.Y.
Redfield, Snowden B. ('10), 1918 Paul Ave., Bethlehem 1, Pa.
Redington, Rowland Wells (J'45) (ABESY), Jr. Engr., Cornell Aero. Lab., Cornell Research Foundation, Inc., 4455 Genesee St., Buffalo 5; *for mail,* 2 Rugby, Buffalo 16, N.Y.
Redlitz, Alfred Carl (J'46) (ACMLKZ), Jr. Research Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 8012 Lake Ave., Cleveland 2, Ohio.
Redmerski, Edmund S. ('27; '34; '35) (CDHJ MT), Supt., James Hunter Mch. Co., North Adams, Mass.
Redmon, John King (J'45) (CLSDMK), Sales Asst., Westinghouse Elec. Corp., 1180 Raymond Blvd., Newark; *for mail,* Apt. 3-E, 471 Madison Ave., Elizabeth 4, N.J.
Redpath, Hugh Smith ('41) (MOS), Plant Engr., Fairbanks Co., 2 Glenwood Ave., Binghamton; *for mail,* 203 E. Union St., Endicott, N.Y.
Redway, Albert Sessions ('38) (CMDN), V.P., Gen. Mgr., Geometric Tool Co., 1 Valley St., New Haven 15, Conn.
Reece, Risden Patterson ('36) (SFDCKL), Ch. Engr., R. J. Reynolds Tobacco Co., Winston-Salem 3, N.C.
Reed, Alonzo Barker ('21; '23) (HKOST), Owner, Firm of Alonzo B. Reed, Cons. Engr., 88 Broad St., Boston 10, Mass.
Reed, Charles Ezra (J'37) (EKBC), Mechanics Helper, Morenci Branch, Phelps-Dodge Co.; *for mail,* Box 915, Morenci, Ariz.
Reed, Chester Arthur ('37) (FKS), Dir., Engrg. Dept., Natl. Coal Assn., 804 So. Bldg., Washington 5, D.C.
Reed, Chester T. ('06; '16), Pres., Reed & Prince Mfg. Co., 1 Duncan Ave., Worcester 1; *for mail,* 354 Salisbury St., Worcester, Mass.
Reed, Daniel Raymond (J'44) (NBMDGJ), Sr. Draftsman, Link-Belt Co., 2045 Hunting Park Ave., Philadelphia 40; *for mail,* 923 Duncannon Ave., Philadelphia 41, Pa.
Reed, Edwin E. (J'37), Capt., Coast Arty. Corps, 58th Antiaircraft Arty. Brigade, Ft. Bliss, Tex.
Reed, F. Everett, Jr. (J'37) (BHESKO), Asst. Prof. Mech. Engrg., Rm. 3-382, Mass. Inst. of Tech., Cambridge, Mass.
Reed, Frederick Jerome ('27; '35) (NSKBEF), Assoc. Prof. Mech. Engrg., Duke Univ., 263 College Sta., Durham, N.C.
Reed, Homer Charles ('41) (LEFCZO), Ch. Engr., Union Oil Co. of Calif., 617 W. 7th St., Los Angeles 14; *for mail,* 1808 Cleveland Rd., Glendale 2, Calif.
Reed, Hudson William ('37) (CEF), Pres., Philadelphia Gas Works Co., 1401 Arch St., Philadelphia, Pa.
Reed, Hugh DeWitt (J'40) (CMS), Asst. Mgr., Prod. Dept., Shop Specialist, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Reed, John Clifford ('28; '45) (SFKHBE), Prof. & Head, Mech. Engrg. Dept., Bucknell Univ., Lewisburg, Pa.
Reed, Karl Fulmer (J'44) (BSCHOM), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 1630 Carrie St., Schenectady 8, N.Y.
Reed, Kenneth W. ('27) (CNMJBD), Owner, Firm of Kenneth W. Reed, Cons. Engr., 4614 Prospect Ave., Cleveland 3; *for mail,* 1839 Wymore Ave., East Cleveland 12, Ohio.
Reed, Lester John (J'43) (CYMDG), Supvr., A. C. Spark Plug Div., Gen. Motors Corp., Dort Highway, Flint 2; *for mail,* 2208 E. 2nd St., Flint 3, Mich.
Reed, Macdonald Seymour ('19; '25; '30) (NHB YMJ), Ch. Engr., Erie Fdy. Co., 1253 W. 12th St.; *for mail,* 402 W. 7th St., Erie, Pa.
Reed, Malcolm Volley (J'39) (BJDM), Head, Engrg. & Planning Depts., Wyatt Metal & Boiler Works, Box 3052, Houston 1, Tex.
Reed, Maurice J. ('29; '35) (EFSAR), Gasoline & Fuel Dept., Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4, N.Y.
Reed, Miles Allen, Jr. (J'43) (CSH), Asst. to Comptroller, Univ. of New Mex.; *for mail,* 1305 Las Lomas Rd., Albuquerque, New Mex.
Reed, Ramsey Marion ('32) (CHJ), Sales Engr., U.S. Pipe & Fdy. Co., 1711—1st Natl. Bldg., Birmingham 3; *for mail,* 509 Hampton Dr., Homewood, Birmingham 9, Ala.
Reed, Robert DeHart ('42) (EFKLM), Engr., John Zink Co., Box 310, Tulsa 1, Okla.
Reed, Robert W. (J'41) (ACSMFJ), 233 S. Kensington Ave., La Grange, Ill.
Reed, Theron W. (J'44) (CJMYND), Metallurgist, Owens-Corning Fiberglas Corp., Newark, Ohio.
Reed, Thomas E. ('29) (CTMSNJ), M.M., Ponemah Mills, Norwich Ave.; *for mail,* 30 N. 3rd Ave., Taftville, Conn.
Reed, Van Applegate, Jr. ('15) (BDEFKS), Mech. Engr., Fed. Engrg. Co., 239—4th Ave., Pittsburgh 22, Pa.
Reed, William Anthony ('28) (CLFS), Gen. Supt., Solvay Process Co., Box 417, Roosevelt Park Annex, Detroit 32, Mich.
Reed, William E. ('98), Secy., Treas., Morewood Realty Corp., 551—5th Ave.; *for mail,* 175 W. 72nd St., New York, N.Y.
Reed, William Hiram (J'36) (SEHKO), Mar. Engr., U.S. Maritime Comm., Commerce Bldg., Washington, D.C.; *for mail,* 2744 S. Troy St., Arlington, Va.
Reeder, William Edward (J'46), U.S. Navy; 22 Wheelwright Rd., West Medford, Mass.

Reedy, Frank (J'38) (CE), Lub. Engr., Stand. Oil Co. of La., 2134 St. Charles Ave., New Orleans; *for mail*, Box 246, Franklinton, La.
Reemelin, Wallace Mark (J'44) (NBAYWD), Engr., Will-Burr Equip. Co., 641 Gilman St., Berkeley; *for mail*, 1846 Spruce St., Berkeley 4, Calif.
Rees, N. John ('44) (BJ), Piping Engr., M.W. Kellogg Co., 225 Broadway, New York; *for mail*, 147-23—73rd Ave., Flushing, N.Y.
Rees, Thomas William (J'44) (ABZJSR), Structures Engr., Bell Aircraft Corp., 2050 Elmwood Ave., Buffalo 7; *for mail*, 1138 Pierce Ave., Niagara Falls, N.Y.
Reese, Eber Oram, III (J'39) (CDFK), Factory Mgr., Reese Padlock Co., 128 Sherman St.; *for mail*, 41 Cottage Ave., Lancaster, Pa.
Reese, Edwin William, Jr. (J'40) (LDKJCN), Mech. Engr., Plant Engrg. Office, Ivorydale Factory, Procter & Gamble Co.; *for mail*, 3992 Parker Pl., Cincinnati 17, Ohio.
Reese, Hermann ('38) (CL), Supt., Ch. Engr., Liebmann Breweries, Inc., 36 Forrest St., Brooklyn 6; *for mail*, 7223—61st St., Brooklyn 27, N.Y.
Reese, John S. (J'40) (BKS), Comdr., U.S.N., U.S.S. *Duluth*, CL-87, c/o Postmaster, San Francisco, Calif.
Reeve, David Douglas ('39; '45) (LBDNJH), Engr., Aluminum Co. of Can. Ltd., 1700 Sun Life Bldg., Montreal; *for mail*, Box 33, Beauharnois, Que., Can.
Reeve, Kenneth Aldrich ('33; '44) (LTOCMB), Engr., Calco Div., Am. Cyanamid Co., *for mail*, Woodland Terrace, Bound Brook, N.J.
Reeve, William Alexander ('46) (BDJ), Sr. Engr., Bridge Div., John A. Roebling's Sons Co., Trenton 2, N.J.
Reeves, Jerry Healand, Jr. ('45) (FST), Application Engr., Westinghouse Elec. Corp., Box 1591, Greenville, S.C.
Reevy John H. (J'41) (CJS), 1033—7th St., Huntington, W.Va.
Regan, Joseph C. ('11), Pres., Gen. Mgr., E. Horton & Son Co., Windsor Locks, Conn.
Regazzi, John James, Jr. (J'44), 246 Carlton Ave., Brooklyn 5, N.Y.
Reggel, Walter G. A. (J'44), Huntington Works, Internatl. Nickel Co., Inc., Huntington, W.Va.
Reh, Paul A. (J'37) (CDMSZ), Lt., U.S.N.R., Head, Navigation Dept., Naval Reserve Officers' Trng. Corp. Univ. of Mich., Ann Arbor, Mich.
Rehfuss, William Clifford ('24) (KSFNRB), Mech. & Sales Engr., Schutte & Koerting Co., 12th & Thompson Sts., Philadelphia; *for mail*, 4346 Woodland Ave., Drexel Hill, Pa.
Rehm, Gustav Adolf ('43) (SKFNJB), Asst. Ch. Mech. Engr., Riley Stoker Corp., 9 Neponset St., Worcester 3; *for mail*, 15 Alexander Rd., Worcester 5, Mass.
Rehnborg, Gordon Alvan (J'39) (CMJN), Ensign, U.S.N.R.; *for mail*, c/o C. H. Lehmann, 414 Franklin Ave., Wilkinsburg, Pa.
Reiber, Harry Paul ('46) (ACO), Asst. Products Engr., A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1; *for mail*, 925 E. Wells St., Milwaukee, Wis.
Reich, Henry Leo ('23; '32) (EHZSDJ), Comdr., U.S.N.R., Fleet Maint., Philippine Sea Frontier, c/o Postmaster, San Francisco, Calif.
Reich, Robert Leon (J'45) (MNJHBC), Ch. Engr., Arlon Co., 1150 Broadway, New York 1; *for mail*, 522 Hanson Court, Far Rockaway, L.I., N.Y.
Reichard, Harry George ('42) (SFK), Dist. Mgr., Combustion Engrg. Co., Inc., 1523 Guardian Bldg., Cleveland 14, Ohio.
Reichard, Herman Charles (J'43) (SMCEFK), Estimating Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Reichel, Curt Rudolph ('34; '35) (CK), Owner, C. R. Reichel Engrg. Co., 718 Natoma St., San Francisco 3; *for mail*, 929 Laurel Ave., San Mateo, Calif.
Reichelt, Clarence V. ('26; '33; '35) (CLKS), Maint. Engr., Chas. Pfizer & Co., 11 Bartlett St., Brooklyn 6; *for mail*, 18 Westcott Blvd., Staten Island 10, N.Y.
Reichert, George L., Jr. (J'40) (ASFB), 6522 Greenview Ave., Chicago 26, Ill.
Reichert, William George ('41) (CDJL), Pres., W. G. Reichert Engrg. Co., 1060 Broad St., Newark 2, N.J.
Reichl, Ray Peter (J'45) (SKOFE), Instr. in Mech. Engrg., Marquette Univ., 1515 W. Wisconsin Ave., Milwaukee 3, Wis.
Reid, David C. ('39), Gen. Supt., Boston & Me. R.R., N. Sta., Boston, Mass.
Reid, Donald G. ('41) (SFJO), Mech. Engr., Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Reid, Harry (A'22), Indianapolis Rys., 110 N. Illinois St., Indianapolis, Ind.
Reid, Henry P. ('35) (CDF), Oper. Engr., Universal Atlas Cement Co., 135 E. 42nd St., New York, N.Y.
Reid, James C., Jr. (J'40) (EKS), Asst. Mech. Engr., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail*, 6400 Knollbrook Dr., Hyattsville, Md.
Reid, James W., Jr. (J'35) (ABM), 709 F St., Copeland Park, Newport News, Va.
Reid, Joseph Gilman ('38) (DHK), Design Mech. Engr., Ala. By-Products Corp., 1st Natl. Bank Bldg., Birmingham 2; *for mail*, 1437—25th Pl. S., Birmingham 5, Ala.
Reid, Stephen Holland ('37) (SFKOL), Sales Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Reid, William Charles ('41) (CGNZ), V.P., Sales Engrg., Metallizing Engrg. Co. Inc., 38-14—30th St., Long Island City 1, N.Y.
Reid, William T. ('40) (FSKL), Supvg. Engr., Fuel Sec., Bur. of Mines, Dept. of the Interior, 4800 Forbes St., Pittsburgh 13, Pa.
Reidl, Albert L. ('28) (MKSZHL), Plant Engr., Gen. Elec. Co., 6901 Elmwood Ave., Philadelphia 42, Pa.
Reiley, Trevor Dale (J'42) (ZFSKE), 680 Vine St., West Lafayette, Ind.
Reilly, Bertram B. (J'36) (FKL), Engr., Mchy. Div., Dravo Corp., 300 Penn Ave., Pittsburgh, Pa.
Reilly, Cornelius Tyson ('42) (ESFLC), Indus. Lub. Engr., Stand. Oil Co. of Pa., Broad & Chestnut Sts., Philadelphia 1; *for mail*, 1962 Howard Ave., Pottsville, Pa.
Reilly, James Harry ('43) (BFKLOS), 14 Watson Ave., East Orange, N.J.
Reilly, James William (J'42) (YJMA), Mfrs. Rep., 327 Smith St., Newark 6, N.J.
Reilly, Thomas Arthur ('42) (JSNR), Designing Draftsman, River Works, Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, 19 Kensington Park, Lynn, Mass.
Reim, Ernest Philip ('44), Sr. Lecturer in Mech. Engrg., Natal Univ. College; *for mail*, 71 Manor Dr., Manor Gardens, Durban, S. Africa.
Reimann, Howard Charles ('45) (BDJNZ), Partner, Engr., Reimann & Georger, Ellicott Sq. Bldg., Buffalo 3; *for mail*, 25 Fernhill Ave., Buffalo 15, N.Y.
Reime, Bert N. (J'42) (AJ), Ensign, U.S.N., 26 W. Main St., Belleville, Ill.
Reimer, Clarence Carl ('20; '35) (BNJHM), Engr., Gen. Mchy. Corp., N. 3rd St., Hamilton; *for mail*, 1445 Maple Ave., Hamilton 1, Ohio.
Reimer, Gerhard Martin (J'41) (SNLARB), Test Engr., Plant Engrg., Swift & Co., Union Stock Yards, Chicago; *for mail*, 6608 S. Wolcott Ave., Chicago 36, Ill.
Reimer, Otto M. (J'42) (MJADGY), Technician, 4th Grade, D Btry., 387th Antiaircraft Arty. (AW) Bn., A.P.O. 209, c/o Postmaster, New York, N.Y.
Reimers, Bernhardt Frederick ('25; '31) (BFT), Indus. Fuel Rep., Pub. Serv. Elec. & Gas Co., Box 1459, 188 Ellison St., Paterson 1, N.J.
Rein, Francis Becht ('45) (CHKLSZ), Plant Engr., Commercial Solvents Corp., Foot of Darst St.; *for mail*, 1104 Fair Oak, Peoria, Ill.
Reinarz, Alvin R. (J'38), Capt., Hdq. Sqd., Abadan Air Base, A.P.O. 824, c/o Postmaster, New York, N.Y.; *for mail*, 505 East Evergreen, San Antonio 2, Tex.
Reinecke, Herman H. ('23; '35), 601 N. McCadden Pl. Los Angeles, Calif.
Reingold, Irving (J'42) (MJACY), Mech. Engr., Radar Lab. Div., Watson Labs., Red Bank; *for mail*, 289 Weequahic Ave., Newark 8, N.J.
Reinhardt, Albrecht Edward (J'42) (CEHKL), Ch. Engr., Derbyshire Mch. & Tool Co., Belfield Ave. & Collom St., Philadelphia 44, Pa.
Reinhardt, Richard G. (J'42), Engr., Charge of Maint., Clinton Engr. Works, Tenn. Eastman Corp.; *for mail*, Rm. 144, Casper Hall, Oak Ridge, Tenn.
Reinhart, Chauncey Osmond ('43; '45) (CM), Prod. Planner, Watson-Stillman Co., Box 26; *for mail*, 220 Park St., Roselle, N.J.
Reinicker, Norman G. ('13; '16; '35), V.P., Gen. Mgr., Pa. Power & Light Co., 901 Hamilton St., Allentown, Pa.
Reiniger, Eberhard Martin ('32) (MJBH), Field-Engr., Cincinnati Milling & Grinding Mchs. Inc., 1421 Chestnut St., Philadelphia 2; *for mail*, 26 S. Hillcrest Rd., Springfield, Delaware Co., Pa.
Reintjes, George Peter ('41) (KOS), Owner, Geo. P. Reintjes Co., 2517 Jefferson Ave., Kansas City 10, Mo.
Reiser, Fred, Jr. (J'42) (BNS), Cincinnati Div. Mgr., Timken Roller Bearing Co., 1601 Reading Rd., Cincinnati 2; *for mail*, 7023 Ohio Ave., Silverton, Cincinnati 13, Ohio.
Reisert, Thomas Donald (J'42) (ACM), Aero. Engr., Gust Research Sec., Natl. Advis. Com. for Aero., Langley Field; *for mail*, Apt. 2, 3785 Kecoughton Rd., Hampton, Va.
Reisman, Frederick William ('26; '28) (CS), Pres., Keystone Refractories Co., 120 Liberty St., New York 6, N.Y.
Reiss, Andrew Edward ('40; '45) (BHKO), Ch. Design Engr., Fedders Mfg. Co. Inc., 57 Tonawanda St., Buffalo 7; *for mail*, 335 Louvaine Dr., Kenmore 17, N.Y.
Reissig, August Richard (J'40) (BEMNS), Indus. Lub. Engr., Socony-Vacuum Oil Co., Inc., Church St., Albany 1, N.Y.; *for mail*, Hotch Kiss Grove Rd., Branford, Conn.
Reissner, Eric (J'44) (AB), Asst. Prof. Math., Mass. Inst. of Tech., Cambridge; *for mail*, 120 Elm St., Belmont, Mass.
Reissner, Hans Jacob ('42) (ABO), Prof. Aerodynamics & Aerostructures, Poly. Inst. of Brooklyn, Livingstone St., Brooklyn; *for mail*, 35 Clark St., Brooklyn 2, N.Y.
Reiter, Bela Zoltan ('29) (CGDZ), Mgr., Illustration Dept., McGraw-Hill Publ. Co. Inc., 330 W. 42nd St., New York 18, N.Y.; *for mail*, 445 Claremont Ave., Teaneck, N.J.
Reith, Miss Marie ('44) (C), Consltd. Edison Co., of N.Y., Inc., 4 Irving Pl., New York 3; *for mail*, 120-05 Long St., Jamaica 5, N.Y.
Reitz, Frederick M. (J'43) (BDHJMN), Mech. Engr., Design, Naval Research Lab., Anacostia Sta.; *for mail*, Apt. 4, 64 Forrester St., S.W., Washington 20, D.C.
Reitzel, Holger Blocker (J'36) (MLJCDS), Factory Planning Engr., West. Elec. Co., Inc., 22nd & Cicero Ave., Cicero; *for mail*, 5317 Berteau Ave., Chicago 41, Ill.
Reiziss, Daniel (J'45) (BCMNHE), Draftsman, Todd Combustion Equip., Inc., 601 W. 26th St., New York 1; *for mail*, 3858—3rd Ave., New York 57, N.Y.
Reker, Carl H. ('21; '27; '35) (COMES), Gen. Consultant, Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Relph, Edward Herbert (J'43) (FSKEBC), Fireman, 1st Class, 3rd Bn., Naval Trng. & Distribution Center, Shoemaker, Calif.; *for mail*, 656 Edgewater Ave., Ridgefield, N.J.
Relyea, John DeWitte ('42) (SMO), Maj., Royal Canadian Elec. & Mech. Engrs., Natl. Defence Hdq., Ottawa; *for mail*, 68 Jackman Ave., Toronto 6, Ont., Can.
Relyea, Louis Kirk (J'45) (SACJ), Pvt., U.S.M.C., Engrg. Draftsman, Hdq. Sqd., M.C.A. T.S., Brown Field, Quantico, Va.; *for mail*, 4029 Bowser St., Dallas 4, Tex.
Remeschatis, Ralph Ernest (J'43) (LDOZKN), Engr., Great Atlantic & Pacific Tea Co., 151 Northern Ave., Boston; *for mail*, 7 Beacon St., Stoneham, Mass.
Remington, Wm. F. (J'39), 3426 Whittier St., San Diego, Calif.
Remp, George Edward (J'39) (SOLZK), Research Asst., Fla. Engrg. & Indus. Exper. Sta.; *for mail*, Box 2657, Univ. Sta., Gainesville, Fla.
Rendos, John Joseph (J'40) (LENKZF), Research Mech. Engr., Air Reduction Sales Co., 41 Magee Ave., Stamford, Conn.
Renfrew, Clinton Alexander (J'39) (SKNB), Steam Turbine Design Engr., Gen. Elec. Co., 920 Western Ave., West Lynn 3; *for mail*, 188 S. Common St., Lynn, Mass.
Rengert, Joseph Stephen (J'45) (NCMB), Engrg. Draftsman, Eastman Kodak Co., Kodak Park, Lake Ave.; *for mail*, 100 Gibbs St., Rochester, N.Y.
Renn, Jos. Anton (J'45) (HKM), 502 Bank St., New Albany, Ind.
Renn, William John, Jr. ('45) (DWY), Brown-Renn Co., Highland & Summit Aves.; *for mail*, 5 Fair Acres Rd., Jenkintown, Pa.
Renner, Elmer John (J'43) (DKM), Ensign, U.S.N.R., 610 W. Nittany, State College, Pa.
Renner, Roland B. ('06; '15; '27) (DCL), Mech. Engr., Robinson Mfg. Co., 80 Church St., New York 7, N.Y.; *for mail*, 22 Ridgewood Terrace, Maplewood, N.J.
Renner, William E. ('37) (BCMW), Factory Contact Engr., Carrier Corp., S. Geddes St., Syracuse, N.Y.
Rennie, John Arthur ('21) (DJMN), Ch. Mech. Engr., Boorum & Pease Co., 192 Front St., Brooklyn 1, N.Y.
Rennie, Robert ('22) EKR), Retired; R.F.D. 1, Collinsville, Conn.
Reno, Harold P. (J'07) (CTSLM), Retired; Oxford St., R.F.D. 1, Seymour, Conn.
Renton, Vinal Stevenson (J'39) (SCFOZH), Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark; *for mail*, R.F.D. 2, Plainfield, N.J.
Rentscher, Roland Roy (J'40) (ACMN), Gen. Mgr., Rentscher Tool & Mch. Corp., 2752 Elston Ave., Chicago 47, Ill.
Renwick, Edward B. ('14) Retired; Western Dr., Short Hills, N.J.
Reoch, Albert G. ('24; '35) Del., Lackawanna & West. Coal Co., 271 Church St., New York; *for mail*, 80 Fairview Ave., Oceanside, N.Y.
Repass, Frank Marvin, Jr. (J'39) (CHSP), Asst. to V.P., Crosby Steam Gage & Valve Co., 165 Broadway, New York 6, N.Y.
Repath, Charles House ('91), Retired; 331 Mt. Holyoke Ave., Pacific Palisades, Calif.
Repetto, Arthur V. (J'40) (FS), Instr. in Engrg., College of the City of N.Y., 140th St. & Amsterdam Ave., New York; *for mail*, 3660 Waldo Ave., Riverdale, N.Y.
Repetto, Felix Edward ('43) (FSEA), Serv. Engr., Pocahontas Fuel Co. Inc., 820 Statler Bldg., Boston 16; *for mail*, 3 Adams St., Belmont 78, Mass.
Repino, Philip Andrew ('39; '41) (OSLFEC), Plant Engr., Lebanon Steel Fdy., Lebanon, Pa.
Repscha, Albert Henry ('31; '38) (SEFKZB), Assoc. Prof. Mech. Engrg., Drexel Inst. of Tech., 32nd & Chestnut Sts., Philadelphia 4, Pa.
Resek, J. Verne ('37) (KFESNC), Dir. of Research, Cleaver-Brooks Co., 5100 N. 33rd St., Milwaukee 9, Wis.
Resek, Marc ('23; '25) (NFKJZ), Ch. Engr., Perfection Stove Co., 7609 Platt Ave., Cleveland 4, Ohio.
Retrum, Rowland (J'39) (WMCY), Asst. Mgr., Du-Crafts, Inc., 307 Geiger St.; *for mail*, 112 Main St., Huntingburg, Ind.
Rettaliata, John Theodore (J'39) (SEBRAJ), *Junior Award*, '41, *Pi Tau Sigma Medallist*, '42; Dir., Dept. of Mech. Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.
Retz, Andrew M. (J'36) (EMKF), Engr., Combustion Equip. Div., Todd Shipyards Corp., 601 W. 26th St., New York 1, N.Y.

Reubush, Robert Floyd, Jr. (J'44) (RSFN), Draftsman & Designer, So. Ry. System, 15th & K St., Washington, D.C.; *for mail*, 819 S. Alfred St., Alexandria, Va.
Reul, Richard Philip (J'42) (A), Flight Test Engr., Airplane Div., Curtiss-Wright Corp., 4300 E. 5th St., Columbus 16; *for mail*, 3646 E. Main St., Columbus 9, Ohio.
Reunert, Theodore ('01), Chmn., Reunert & Lenz, Ltd., Box 92, Johannesburg, S. Africa.
Reuschlein, Clifford Joseph (J'42), U.S.N.R., 2018 Madison St., Madison 5, Wis.
Reuter, Otto Ernest (A'45) (CMDZFJ), Gen. Mgr., A. Henkel & Son, Inc., Keene, N.H.
Reuter, Peter Theodore ('45) (SZFLTR), Mgr., New England Dist., Bailey Meter Co., 89 Broad St., Boston 10, Mass.
Revere, Francis James ('17) (BEJKLM), Engr., Allis-Chalmers Mfg. Co., Milwaukee; *for mail*, 6719 Cedar St., Wauwatosa, Wis.
Rex, Clifford N. ('34; '35) (CLM), 239 E. Basline Rd., San Dimas, Calif.
Rexford, Robert Stanley (J'45) (CLYDNK), Asst. Dir. of Research, Am. Mch. & Metals, Inc., East Moline; *for mail*, 1145—15th St., Moline, Ill.
Rey, Robert Michael (J'40) (LCWMY), 1175 Grand Concourse, Bronx 52, N.Y.
Reyling, Geo. ('20), 214 Pomander Rd., Mineola, L.I., N.Y.
Reyling, Theodore R. (J'46), Lt. Comdr., U.S.N.R., 16 Thompson Park, Glen Cove, N.Y.
Reyna, Leon C. ('23; '34) (EFS), 480 Yates St., Albany 8, N.Y.
Reynell, Carleton ('20; '23) (CJOFDM), Gen. Mgr., Purchases & Traffic, Worthington Pump & Mchy. Corp., Harrison, N.J.
Reynick, Henry Fish ('26) (DMW), Owner, Reynick Co., 404 St. Charles Ave., New Orleans 12, La.
Reynolds, Alexander Thomas (J'41) (SKEF), Engr., Natl. Power Constr. Co., 348 S. Dearborn St., Chicago 4; *for mail*, 6426 Stewart Ave., Chicago 21, Ill.
Reynolds, Donald Dean (J'39) (CHJ), Engr., Boeing Aircraft Co.; *for mail*, 2210—10th Ave. N., Seattle, Wash.
Reynolds, Glenn E. ('39) (SWFO), Co-Partner, Natl. Power Constr. Co., 343 S. Dearborn St., Chicago 4, Ill.
Reynolds, Harry Weatherly (J'42) (NBH), Ch. Engr., Kilby Steel Co., W. 10th St., Anniston, Ala.
Reynolds, Herbert B. ('12; '19; '22; F'41) (SFE HJZ), Supt. Power Generation, N.Y. City Transit System, 600 W. 59th St., New York 19, N.Y.
Reynolds, Herman Foster ('45) (SNEFB), Mech. Designer, Philadelphia Elec. Co., 9th & Sansom Sts., Philadelphia 5, Pa.; *for mail*, 259 Harvard Ave., Collingswood, N.J.
Reynolds, James Irving ('45) (SERKZ), Asst. Prof. Mech. Engrg., College of Engrg., W.Va. Univ., Morgantown, W.Va.
Reynolds, Larry (J'43) (BJM), Prod. & Tooling Engr., Navy Tooling Dept., Naval Ord. Div., Consltd. Steel Corp. Ltd., 5700 Eastern Ave., Maywood; *for mail*, 6919 Paseo del Serra, Hollywood 28, Calif.
Reynolds, Lewis Minter ('43) (ONLS), Asst. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Reynolds, Robert Livingston ('44) (JKS), Sec. Engr., S. Philadelphia Works, Westinghouse Elec. Corp., Lester; *for mail*, 16 W. Hinckley Ave., Ridley Park, Pa.
Reynolds, Samuel Dornon ('40) (SKLH), Engr., Mech. Engrg. Div., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia; *for mail*, 226 Rutledge Ave., Rutledge, Pa.
Reynolds, Spencer Walter ('33; '45) (OESCBM), Sales Engr., Continental Engrg. Corp., 30 Church St., New York, N.Y.; *for mail*, 531 St. Mark's Ave., Westfield, N.J.
Reynolds, Thurlow Weed ('44) (FSKE), Cons. Engr., 100 Pinecrest Dr., Hastings-on-Hudson, N.Y.
Reynolds, Winfred Everett ('28) (CO), V.P., Dir., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Reznek, Ben (J'35) (JBSEKF), Design & Project Engr., Reed Research, 1048 Potomac St., N.W., Washington 7; *for mail*, 1925—1st St., N.W., Washington 1, D.C.
Rhame, Frank Phipps ('16; '19; '35) (SCJ), Pres., Gen. Mgr., Lunkenheimer Co., Box 360, Cincinnati 14, Ohio.
Rhea, Hal Haynes (J'44) (KHB), Jr. Engr., Devel. Dept., Carrier Corp., 800 S. Geddes St., Syracuse 1; *for mail*, 408 Greenwood Pl., Syracuse 10, N.Y.
Rheingans, William Jacob ('38) (ABH), Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1, Wis.
Rhine, Carl K. (J'32) (SMCEFK), Power Engr., Gen. Chem. Co., 1100 Line St., Camden; *for mail*, 205 Morgan Ave., Collingswood, N.J.
Rhinehart, John R. ('23; '30; '33), 3825 Campbell, Kansas City, Mo.
Rhoades, John Francis ('22) (FLS), Project Engr., H. S. Ferguson & Co., 200—5th Ave., New York 10, N.Y.
Rhoades, Robert Paul (J'37) (CEGHIS), Lt. (j.g.), U.S.N.R., 301st Naval Constr. Bn., F.P.O., San Francisco, Calif.; *for mail*, Box 121, Glasgow, Mont.
Rhoads, George Elwood ('06; '26) (RESFK), Retired; *for mail*, 1320—20th Ave., Altoona, Pa.
Rhoads, J. Edgar ('16) (CYTS), Partner, J. E. Rhoads & Sons, Box 71, Wilmington 99, Del.
Rhoads, Philip G. ('28; '35) (CMTDA), Partner, J. E. Rhoads & Sons, Box 71, Wilmington 99, Del.
Rhoda, Ralph Allen (J'45) (HNBC), Ch. Engr., Berkeley Pump Co., 829 Bancroft Way, Berkeley 2; *for mail*, 1546 Beverly Pl., Berkeley 6, Calif.
Rhodes, Almon Erle (J'44) (SEC), Hyd. Engr., U.S. Bur. of Reclamation, Golden Eagle Bldg., Denver 1; *for mail*, 14 Sunset Ridge, Littleton, Colo.
Rhodes, Augustus Loring (J'42) (ACMR), Gen. Mgr., Dri-Steam Products Inc., 29 Broadway, New York 6; *for mail*, 54 Mayflower Ave., New Rochelle, N.Y.
Rhodes, Edward Keith (J'29) (SG), Design Engr., Cent. N.Y. Power Corp., 800 Erie Blvd. W., Syracuse 2; *for mail*, 158 Fay Rd., Syracuse 9, N.Y.
Rhodes, George H. ('18; '25) (DNMB), Mgr., Engrg. Dept., Natl. Biscuit Co., Box 3, Sta. O, New York 11, N.Y.
Rhodes, Geo. I. ('19), V.P., Ford, Bacon & Davis, Inc., 39 Broadway, New York, N.Y.
Rhodes, Thomas John (J'41) (BLM), 39 Pine St., Ramsey, N.J.
Rhyne, Cecil, Jr. (J'37), Engr., Gen. Elec. Co., 1 River Rd.; *for mail*, 2036 Coolidge Rd., Schenectady, N.Y.
Rianhard, Thomas McMahon, Jr. ('42) (CMDJ), Indus. Engr., Incentive Stands., Chase Brass & Copper Co., Inc., 236 Grand St., Waterbury 91; *for mail*, Charcoal Ave., Middlebury, Conn.
Ribble, George William (J'42) (FJK), Staff Engr., Aluminum Co. of Am., Alcoa; *for mail*, 4 Park Court, Maryville, Tenn.
Ricard, Pierre Jean ('45), Ch. Engr., Union d'Electricité, 3 rue de Messine, Paris 8, France.
Ricardo, Harry R. (H'42), Tech. Dir., Ricardo & Co., Ltd., *for mail*, Tottington Manor, Small Dole, Sussex, England.
Rice, Arthur H. (J'36) (CGM), Mech. Engr., Miehle Ptg. Press & Mfg. Co., 2011 W. Hastings St., Chicago; *for mail*, 8113 Maryland Ave., Chicago 19, Ill.
Rice, Arthur Louis ('95; '02; F'36) (SF), *Vice-President*, '22; Editorial Dir., *Power Plant Engineering*, Tech. Publ. Co., 53 W. Jackson Blvd., Chicago 4; *for mail*, 518 Central Ave., Wilmette, Ill.
Rice, C. Milton (J'42) (AKCF), Sr. Test Engr., Wright Aero. Corp., Beckwith Ave., Paterson; *for mail*, 510 Valley Rd., Upper Montclair, N.J.
Rice, Cyrus William ('15) (ELSTW), Pres., Cyrus Wm. Rice & Co., Inc., 17 Noble Ave., Pittsburgh 5, Pa.
Rice, Dixon Burnett (J'41) (AJHCFM), Lt. Comdr., U.S.N., VP-130, MCAS-Miramar, San Diego 45, Calif.
Rice, Milton H. (J'39) (BMJ), Mold Designer, Scintilla Magneto Div., Bendix Aviation Corp.; *for mail*, Overlook Dr., Sidney, N.Y.
Rice, Norman Douglas (J'39) (CMNDJO), Prod. Engr., Inland Steel Container Co., 6532 S. Menard Ave., Chicago 38; *for mail*, 7122 Normal Blvd., Chicago 21, Ill.
Rice, Paul Eugene (J'40) (BEJ), Engrg. Analyst, Webster Brinkley Co., 651 Alaska St.; *for mail*, 12009—70th Ave. S., Seattle, Wash.
Rice, Robert Barton ('26; '31; '35) (EFOSR), Dir., Diesel Sch., N.C. State College, Raleigh, N.C.
Rice, Robert Gray (J'39) (JMON), Treas., Milwaukee Chaplet & Mfg. Co., 1023 S. 40th St., Milwaukee 4, Wis.
Rice, William E. ('37) (FCEKSZ), Ch. Admin. Serv., Bur. of Mines, Washington 25, D.C.
Rice, William Maxwell (J'44) (OAGC), Arch. Services, Wurster, Bernardi & Emmons, Archs., 402 Jackson St., San Francisco 11; *for mail*, 4806 Davenport Ave., Oakland 2, Calif.
Rice, William Mitchell (J'44) (MJCDBN), Capt., Army Air Forces, 955th Air Engrg. Sqd., 529th Air Serv. Gp., A.P.O. 534, New York, N.Y.; *for mail*, 812 Agnes Ave., Marlin, Tex.
Rich, Abraham Moses (J'46) (EGM), 102 Devon St., Roxbury 21, Mass.
Rich, Bruce Paul (J'45) (MSBEDK), 405 S. 3rd St. W., Mt. Vernon, Iowa.
Rich, George R. ('35) (HBS), Hyd. Engr., Charles T. Main, Inc., 201 Devonshire St., Boston, Mass.
Rich, Keith (J'37) (NBHCFK), Devel. Engr., Goss Ptg. Press Co., 1535 S. Paulina; *for mail*, 11521 S. Oakley Ave., Chicago 43, Ill.
Richard, James Ashburner (J'44) (DRBYNK), Mech. Engr., Heyl & Patterson Inc., 56 Water St., Pittsburgh 22, Pa.
Richard, LaSalle Anthony (J'41) (JHMF), Asst. Hyd. Engr., Steam Engrg. Dept., Tenn. Coal, Iron & R.R. Co., Ensley Sta., Birmingham 8, Ala.; *for mail*, 841 W. Bellevue St., Opelousas, La.
Richards, Allen H. ('45) (CMY), 5238 College View Ave., Los Angeles 41, Calif.
Richards, Donald G. (J'39) (BJA), Project Engr., Research & Devel., Hamilton Stand. Propellers Div., United Aircraft Corp., East Hartford 8; *for mail*, R.F.D. 2, Andover, Conn.
Richards, Edwin H. (J'42) (CKF), 1640 Irving St., N.W., Washington, D.C.
Richards, George B. (J'35) (FH), 9416 Karlov Ave., Skokie, Ill.
Richards, Gerald Reed (J'24) (CMDLY), Factory Supt., Am. Chicle Co., 30-30 Thompson Ave., New York 1; *for mail*, 182 Quaker Ridge Rd., Manhasset, N.Y.
Richards, H. F. (J'43), Mech. Design Engr., Calif. Inst. of Tech., 1201 E. California St., Pasadena; *for mail*, 771 E. 14th St., Los Angeles 21, Calif.
Richards, Harry Edward (J'36) (SJHKB), Draftsman, Yuba Mfg. Co., 351 California St., San Francisco; *for mail*, 1366 Delaware St., Berkeley 2, Calif.
Richards, James Verne ('44) (CSKFAB), Ch. Constr. Engr., Chicago Dist. Elec. Generating Corp., 122 S. Michigan Ave., Chicago 3, Ill.
Richards, John Young, Jr. ('44) (SOACEL), Field Engr., Sullivan Mchy. Co., 307 N. Michigan Ave., Chicago 1, Ill.
Richards, Keene ('27) (COS), Gen. Mgr., Cons. Engr., Vassar College, Poughkeepsie, N.Y.
Richards, Raymond Gardner (J'40) (ACMORS), Field Serv. Engr., Lockheed Aircraft Corp., Burbank; *for mail*, 11318 Erwin St., North Hollywood, Calif.
Richards, Roscoe C. (J'39), 321 N. Washington, Ottumwa, Iowa.
Richards, Thomas Edmund, Jr. (J'42) (CJM NYD), Anchor Hocking Glass Corp., Lancaster; *for mail*, 1622 Glenn Ave., Columbus 8, Ohio.
Richards, William Arthur (J'42) (ZSBCLK), Engr., E. I. du Pont de Nemours & Co.; *for mail*, 1347 Haupt St., Richland, Wash.
Richards, William Merril Shaw (J'38) (KAE BSF), Asst. Project Engr., Wright Aero. Corp., Paterson 3, N.J.
Richards, William Newman (J'43) (MDAKSN), Head, Mech. Engrg. Schs., Internatl. Correspondence Schs., 1001 Wyoming Ave., Scranton 9, Pa.
Richardson, Ammi Carleton ('13; '19) (TSK OF), Asst. Gen. Engr., Am. Thread Co., 260 W. Broadway, New York 13, N.Y.
Richardson, Bayard E. ('29), Pres., Treas., Products Engrg. Serv., 1615 Eastern Ave., S.E.; *for mail*, 921 Sherman St., S.E., Grand Rapids, Mich.
Richardson, Carl (J'45) (KA), Graduate Student, Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.; *for mail*, 159 Hemlock Rd., New Haven 15, Conn.
Richardson, Edward Adams (J'22) (EBAK), Spec. Engr., Publications & Cons., Bethlehem Steel Co., E. 3rd & Buchanan Sts.; *for mail*, 1133 N. New, Bethlehem, Pa.
Richardson, Francis Everett (J'35) (MD), Prod. Clerk, Mercury Mfg. Co., 4044 S. Halsted St., Chicago 9; *for mail*, 1119 Maple Ave., Evanston, Ill.
Richardson, Fred C. ('37) (SFBKO), Mech. Engr., Westcott & Mapes, Inc., 109 Church St., New Haven 5; *for mail*, 245 Ellsworth Ave., New Haven 11, Conn.
Richardson, Frederick Pierce (J'41) (AHZJR), Sales Engr., Bean-Cutler Div., Food Mchy. Corp., San Jose; *for mail*, 905 N. Stockton St., Stockton, Calif.
Richardson, George Partridge ('01; '17), Partner, Bellport Associates, 441 Baldwin Rd., Maplewood, N.J.
Richardson, Harold C. ('25; '37) (CM), Works Mgr., Morris Mch. Works, Baldwinsville; *for mail*, 2533 W. Genesee St., Syracuse 9, N.Y.
Richardson, Harry Duchesne (J'43) (SEFKYH), Lt., Co. B (Misc.), A.P.O. 11984, c/o Postmaster, New York, N.Y.
Richardson, Henry Martyn ('42) (YOCNBW), Partner, DeBell & Richardson, Box 240, P.O. Square Bldg., Springfield 1, Mass.
Richardson, James K. ('37), Asst. Chief, Mech. Engrg. Div., U.S. Bur. of Reclamation, Denver, Colo.
Richardson, Lawrence ('41) (ACDEJR), Mech. Asst. to V.P. & Gen. Mgr., Boston & Me. R.R., 150 Causeway, Boston; *for mail*, 19 Ware St., Cambridge 38, Mass.
Richardson, Marion B. ('24; '35) (ACR), Lt. Col., Trans. Corps, U.S.A., Indus. Relations Officer, Hdq., N.Y. Port of Embarkation, U.S. Army Base, Brooklyn, N.Y.; *for mail*, Plaza Hotel, 1553 N. Clark St., Chicago 10, Ill.
Richardson, Paul Holman ('46) (AEM), Mar. Div., Bendix Aviation Corp., Norwood; *for mail*, Seekouk St., Norfolk, Mass.
Richardson, Percy Laurie (A'45) (SEFCTL), Power Rate Consultant, E. I. du Pont de Nemours & Co., Market St., Wilmington 98, Del.
Richardson, Richard LeVoyle (J'41) (BHJK LY), Instr., Oregon State College; *for mail*, 626 N. 31st St., Corvallis, Ore.
Richardson, Robert William ('46), Engr., Sanderson & Porter, 52 William St., New York 5, N.Y.
Richardson, William H. (J'46), 6025 Upland St., Philadelphia 42, Pa.
Richardz, Folke ('44) (BCAMN), Mgr. of Engrg., Westinghouse Elec. Corp., 200 McCandless Ave., Pittsburgh 1, Pa.
Richeson, William Lyle (A'45), Asst. V.P., Sales, Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.
Richmond, Ernest Leon (J'42) (NOJHBA), Mech. Engr., Ransome Mchy. Co., Dunellen; *for mail*, 1189 Myrtle Ave., Plainfield, N.J.

Richmond, Harold Anthony ('98; '21) (C), Treas., Gen. Abrasive Co., College Ave.; *for mail*, 625 Buffalo Ave., Niagara Falls, N.Y.
Richmond, J. H. M. ('45) (CMYHBN), Gen. Mgr., Pedlow Mch. Co.; *for mail*, 2107 Madison St., Chester, Pa.
Richmond, Julian ('03; '08; '19), Chmn., Potdevin Mch. Co., Brooklyn 18; *for mail*, 71 Dunwoodie, Yonkers 4, N.Y.
Richmond, Julius David ('24; '25; '35) (HRB NSM), Asst. Mech. Engr., Bd. of Water Supply, 120 Wall St., New York; *for mail*, 201 Eastern Pkwy., Brooklyn 16, N.Y.
Richmond, Robt. L. (J'35), Asst. Supt., Potdevin Mch. Co., 1221—38th St., Brooklyn; *for mail*, 7 Pine Terrace, Bronxville, N.Y.
Richmond, William Osborn ('31; '40) (BNJM), Assoc. Prof., Univ. of B. C., Vancouver, B.C., Can.
Richter, Frederick Henry (J'37) (SECM), Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Richter, George A. ('24; '35) (EFS), Sales Engr., Denver Fire Clay Co., Denver, Colo.
Richter, William W. ('42), Ch. Engr., Moore & McCormack Lines, 5 Broadway, New York; *for mail*, 24-46 Butler St., East Elmhurst, N.Y.
Richtmann, William Muir ('43) (SKE), Prof. & Head, Mech. Engrg. Dept., Colo. Sch. of Mines, Golden, Colo.
Rick, Constantine ('29; 36) (NJABHR), Mech. Engr., Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Ricker, Richard (J'38) (MCD), Engr., Merchant Calculating Mch. Co., Powell St., Emeryville; *for mail*, Box 596, Route 1, Lafayette, Calif.
Rickerman, J. Herman ('44) (BFGJKL), Asst. Ch. Mech. Engr., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.; *for mail*, 276 Monmouth Ave., River Edge, N.J.
Ricketts, Edwin Burnley ('08; '16) (SFEL), *Vice-President*, '40-'42; Mech. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Ricketts, John Blaine, (J'44), Solane Apts., 42-637 Bennett Ave., Vallejo, Calif.
Ricketts, Raymond (J'36) (EOSCY), Mech. Engr., Union Sulphur Co. Inc., Sulphur, La.
Rickley, Samuel Spring (J'45) (NJAKMY), Mech. Engr., Superior Steel Corp., Carnegie, Pa.
Ricks, John Leslie (J'41) (OCESML), Mech. Engr., Ford, Bacon & Davis Constr. Corp., 805 S. Grand; *for mail*, 706 Glenmar, Monroe, La.
Riconda, Leo Joseph ('34; '45) (BKL), Mech. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail*, 415 Minneford Ave., City Island 64, N.Y.
Riddiford, Arthur B., Jr. ('31; '41) (CHM), Ch. Engr., John S. Barnes Corp., S. Water St.; *for mail*, 1318 Camp Ave., Rockford, Ill.
Riddle, George Walton (J'43) (SCK), Refrigeration Unit, Engrg. Div., Gen. Elec. Co., E. Lake Rd.; *for mail*, c/o S. A. Salamone, 636 E. 21st St., Erie, Pa.
Riddle, Kenneth W. (J'43) (BGM), 37 Woodycrest Lane, Rose Valley Acres, Moylan, Pa.
Ridenour, James Otto ('45) (NBAJY), 1447 Anderson, Manhattan, Kan.
Rider, Harry Nichol ('43) (HKMFCJ), Mgr., Tech. Div., Automatic Sprinkler Co. of Am., Box 360, Youngstown 1; *for mail*, 108 McKinley Ave., Youngstown 9, Ohio.
Ridge, Charles Kennedy (J'43), Pvt., U.S.A., 247 W. Glen Ave., Ridgewood, N.J.
Ridgely, Holland Farthing (J'45) (NBHJMA), Mech. Engr., Naval Ord. Plant, 6000 E. 21 St.; *for mail*, 1416 N. Olney St., Indianapolis, Ind.
Ridley, Earl Lewis ('29; '35) (CDFST), Plant Engr., Bigelow-Sanford Carpet Co., Inc., Main St., Thompsonville, Conn.
Ridley, Kenneth Joseph ('39) (CDFMST), Plant Engr., Bigelow-Sanford Carpet Co., Inc., 37 Prospect St.; *for mail*, 451 Guy Park Ave., Amsterdam, N.Y.
Ried, Robert C. (J'34) (LDKH), Res. Mgr., Process Equip. Div., Gen. Am. Trans. Corp., 2310 Koppers Bldg., Pittsburgh 19, Pa.
Riede, John Robert (J'43) (NWDER), Mech. Engr., Manhattan Project (Univ. of Calif.), Box 1663, Santa Fe, New Mex.
Riede, Peter M. (J'40) (NLC), Lab. Div. Head, Linde Air Products Co., E. Park Dr., Tonawanda, N.Y.
Riedel, Carl William, Jr. (J'41) (MCODRH), 1413 Rose Virginia Rd., Reading, Pa.
Riedel, Harvey John (J'41) (SOCEHN), Export Div., Worthington Pump & Mchy. Corp., 2 Park Ave., New York 16, N.Y.; *for mail*, 223—75th St., North Bergen, N.J.
Riedel, Richard Joseph (J'45) (MNY), Engrg. Draftsman, Goodyear Tire & Rubber Co. of Kan.; *for mail*, 1821 Huntoon, Topeka, Kan.
Rieder, Ernest Victor ('41) (SZBK), Tech. Engr., Trenton Channel Plant, Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail*, 2210 Ruskin Rd., Trenton, Mich.
Riegels, Olaf Leonhard ('27) (EBHAMK), 22 Webster Pl., Orange, N.J.
Riehl, Harmon B. ('24; '35), Ch. Engr., Textile Div., Proctor & Schwartz, Inc., 7th St. & Tabor Rd., Philadelphia 20; *for mail*, R.D. 3, Norristown, Montgomery Co., Pa.
Rieke, Vernon W. (J'40) (BMCLSA), Maint. Engr., Aluminum Cooking Utensil Co.; *for mail*, Aluminum Club, New Kensington, Pa.
Rienks, Geo. W. ('18) (DFK), Supvg. Engr., Great West. Sugar Co., Sugar Bldg., Denver, Colo.
Riera, Pelayo Vincent (J'40) (CHSEA), Exec. Engr., Cia. Riera Toro & Van Twistern, S.A., Box 916, Havana, Cuba.
Riesenberg, Frank Walter (J'45), Test Engr., Gen. Elec. Co., Philadelphia; *for mail*, 6548 Reedland St., Philadelphia 42, Pa.
Riesing Ellwood Franklin ('42) (ABY), Ch. Automotive Engr., Firestone Indus. Products Co., Inc., Akron, Ohio.
Riester, Robert Andrew (J'39) (MS), Devel. Engr., Elliott Co.; *for mail*, Spanish Villa, Jeannette, Pa.
Rietz, Carl Austin ('31; '34) (BDGLNW), Pres., Engr., Process Mchy. Co. & Jos. Wagner Mfg. Co., 441 Folsom St., San Francisco, Calif.
Rietz, Charles Freund (J'43) (SFE), Ensign, U.S.N.R.; 924 E. Wells, Milwaukee 2, Wis.
Rietz, Elmer Weber ('32) (ZLWTC), Mgr., Specialty Div., Powers Regulator Co., 2720 Greenview Ave., Chicago 14, Ill.
Riford, Charles Payne (J'39) (HMN), Prod. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Rigby, Edward Joseph ('18) (AYTLJ), Tech. Dir., Chmn., Robert Bryce & Co., Pty. Ltd., 526-32 Little Bourke St., Melbourne, Victoria, Australia.
Rigby, James E. ('37) (CM), Plant Engr., Blue Ridge Glass Corp.; *for mail*, 1440 Brightridge Dr., Kingsport, Tenn.
Rigdon, Carl ('07; '16), 171 Montclair Ave., Montclair, N.J.
Riggs, Harold E. ('27) (CES), Owner, Mgr., Riggs Farms, R.F.D. 3, Belmont, N.Y.
Riggs, John D. (J'92), Retired; Box 333, R.R. 10, Indianapolis, Ind.
Riggs, Kenneth (J'42) (BKS), c/o Miss Anne Yeganian, 95 Radford St., Yonkers 5, N.Y.
Rightmire, Brandon Garner ('35; '43) (BAH), Asst. Prof. Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Riis, Erling ('27), So. Kraft Corp., Mobile, Ala.
Riker, George Edwin ('37) (SHEBFL), Mech. Engr., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.; *for mail*, 1528 Bradford Terrace, Union, N.J.
Riley, Benjamin A. (J'40), Devel. Engr., Heald Mch. Co., Bond St., Worcester; *for mail*, 16 Eastford Rd., Auburn, Mass.
Riley, Joseph C. ('09) (BEH), Retired; 9 Pond View Ave., Jamaica Plain 30, Mass.
Rilliet, Jean Louis ('25) (BKSZ), Ch. Mech. Engr., City of St. Louis, Market & 12th Sts., St. Louis 3; *for mail*, 3952-A Sullivan Ave., St. Louis 7, Mo.
Rimbach, Richard ('45), Cons. Metal. Engr., 1117 Wolfendale St., Pittsburgh 12, Pa.
Rincliffe, Roy George ('41) (CFHS), V.P., Charge of Elec. Operas., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Rindner, Jack (J'44) (CJM), Electronics Technician's Mate, 3rd Class, U.S.S. *Fogg*, Destroyer Escort 57, F.P.O., New York, N.Y.
Rindsberg, Harry D. ('30) (BCDGMY), Prod. Mgr., *The Cincinnati Enquirer*, 617 Vine St., Cincinnati 2, Ohio.
Ring, Roy Alfred ('29) (LFS), Design Engr., Atlantic Sugar Refineries Ltd., Charlotte St., St. John, N.B., Can.
Ring, Vincent P. ('23; '33; '35) (CM), Treas., Knapp-Monarch Co., 3501 Bent Ave., St. Louis 16, Mo.
Ringlee, Norman Paul (J'42) (CMSO), Ch. Estimator, Puget Sound Bridge & Dredge Co., 2929—16th Ave., S.W.; *for mail*, 8464—42nd Ave., S.W., Seattle 6, Wash.
Ringo, William Robert (J'41) (MJNL), Tool Process, Delco Products Div., Gen. Motors Corp., 329 E. 1st St., Dayton 1; *for mail*, 2837 Kingston Ave., Dayton 10, Ohio.
Ringwald, Elmer August ('22; '32; '35) (ABD LNW), Designing Engr., Ashland Div., Am. Rolling Mill Co., Ashland, Ky.; *for mail*, 302 Cherry St., Chillicothe, Ohio.
Ringwalt, Victor George, Jr. (J'42) (BHMS), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady, N.Y.; *for mail*, 145 New St., Glenside, Pa.
Rink, George William ('07) (R), Retired; 603 Brobst St., Shillington, Pa.
Riola, Michael Ralph (J'42) (AHJKY), Engr., Mch. Devel. Dept., Remington Arms Plant, E. I. du Pont de Nemours & Co., Bridgeport; *for mail*, 3035 Main St., Bridgeport 6, Conn.
Riopelle, Constantine Paul ('28) (EFNS), 310 Court St., Alton, Ill.
Riordan, Hugh Ernest (J'44) (BEL), 517—8th Ave., Troy, N.Y.
Riordan, John M. ('21) (ACM), Pres., Gen. Mgr., Riordan Mchy. Co., 213 Curtis Bldg., Detroit 2; *for mail*, 2233 Tuxedo Ave., Detroit 6, Mich.
Riordan, William Joseph (J'37) (RANE), Gp. Engr., Sikorsky Aircraft Div., United Aircraft Corp., South Ave., Bridgeport; *for mail*, 60 Hawkins St., Derby, Conn.
Ripley, Charles Trescott ('29; F'45) (RJE), Ch. Engr., Wrought Steel Wheel Industry, 310 S. Michigan Blvd., Chicago 4, Ill.
Ripley, E. Bradford, Jr. ('29; '35) (SBWH), Plant Supt., Conn. Light & Power Co., Devon, Conn.
Ripley, Mills Norton ('41) (HTRC), Hyd. Press Mfg. Co., 500—5th Ave., New York 18; *for mail*, 114 Brite Ave., Scarsdale, N.Y.
Ripley, Reginald Lincoln ('31) (CFJLM), Prod. Engr., Sturgis & Ripley Associates, 60 State St., Boston; *for mail*, 14 Elm St., Wellesley Hills 82, Mass.
Rippe, Carlos Chacon (J'35) (BCHNSW), Mgr., Carlos Rippe Ch. & Cia.; *for mail*, Carrera 41, 63-09, Barranquilla, Colombia, S.A.
Rippey, George Stone ('43) (HNEB), Mech. Engr., 3685 Midiron Dr., Winter Park, Fla.
Rise, Kaare (J'45) (DLS), Plant Engr., U.S. Gypsum Co., 561 Richmond Terrace, Staten Island; *for mail*, 5408—6th Ave., Brooklyn 20, N.Y.
Rising, Simeon Marshall, Jr. (J'41) (ARCHDF), Installation Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St.; *for mail*, 20 Wind Rd., East Hartford 8, Conn.
Risley, Donald Lloyd (J'37) (MHJLZS), Engr., Mch. Designer, Cleveland Twist Drill Co., 1242 E. 49th St., Cleveland 14; *for mail*, 1343 E. 141st St., East Cleveland 12, Ohio.
Risley, William Linton, Jr. (J'45) (LMNCD), Mech. Engr., Food Mchy. Corp., Box 760, San Jose, Calif.
Risteen, Horace William ('25; '35) (AEK), Assoc. Prof. Mech. Engrg., Mich. College of Min. & Tech., Houghton, Mich.
Ritcheske, William Frederic, Jr. (J'44) (CF EA), Mech. Engr., Tech. & Research Div., Tex. Co., Beacon, N.Y.
Ritchey, Lloyd B. (J'37), Instr., Univ. of Ill., Urbana, Ill.; *for mail*, R.R. 4, Lafayette, Ind.
Ritchie, Albert P. ('24; '35), Managing Dir., Taylor Mchy. Co. Ltd., Grange Lane, Leicester, England.
Ritchie, Arthur Harman (J'38) (C), Mem. of Firm, Boso & Ritchie, Inc., Ravenswood, W.Va.
Ritchie, Frank Albert (J'42) (CDM), 854 Charlotte Rd., East Windsor, Ont., Can.
Ritchie, Paul ('21; '33) (YSCFMZ), Mech. Engr., Bradstone Rubber Co., Woodbine; *for mail*, 572 Columbia Ave., Millville, N.J.
Ritchings, Frank A., Jr. (J'46) (KOS), 16 Jane St., New York 14, N.Y.
Ritchings, Robert Hemsworth (J'35) (ZKSY), Engr., Research Dept., Goodyear Tire & Rubber Co., Akron 16, Ohio.
Rittelmeyer, John Mosal ('23; '35) (KLSZ), Pres., Rittelmeyer & Co., 816 Bona Allen Bldg., Atlanta 3, Ga.
Ritter, Carlisle A. (J'43) (ZAL), Application Engr., Taylor Instrument Cos., Inc., 95 Ames St., Rochester 1; *for mail*, 51 Bellevue Dr., Rochester 7, N.Y.
Ritter, George T., Jr. (J'40), 1320 Walnut St., Williamsport, Pa.
Ritter, Harold Peter ('25; '26) (B), Partner, Oberhelman-Ritter Fdy. Co., 3323 Colerain Ave., Cincinnati 25, Ohio.
Ritter, Herman (J'38), Sewing Mch. Technician, Singer Mfg. Co., Trumbull St., Elizabeth; *for mail*, 115 Mohawk Dr., Cranford, N.J.
Ritter, Kurt (J'29) (CM), Mch. Designer, Am. Can Co., 499 Alabama St., San Francisco 10; *for mail*, 846 Acacia Dr., Burlingame, Calif.
Ritter, Paul Alex. ('25) (EB), Retired; 16 Helen St., Melbourne, Fla.
Ritter, Walter T. ('24; '32) (GDCZYK), Mech. Engr., Chicago Carton Co., 4200 S. Crawford Ave., Chicago 32; *for mail*, 707 William St., River Forest, Ill.
Ritterbusch, Harry F. (J'37) (MCN), Gen. Branch Engr., Am. Optical Co.; *for mail*, 137 Lebanon St., Southbridge, Mass.
Rittman, W. F. ('19), Cons. Engr., 6112 Alder St., Pittsburgh, Pa.
Rivello, Robert Matthew (J'46), Instr., College of Engrg., Univ. of Md., College Park, Md.
Rivera-Rodriguez, Denjiro (J'41) (TMCSD), Mech. Engr., Puerto Rico Devel. Co., 55 Allen, San Juan; *for mail*, 2 Orbeta St., Santurce, P.R.
Rives, Thomas McDowell, Jr. (J'46) (BHK), Research Engr., Worthington Pump & Mchy. Corp., Harrison; *for mail*, 331 W. Scott Ave., Rahway, N.J.
Rivinius, George Ambrose, Jr. (J'39) (CWMO DN), Propr., F. C. Rivinius & Co., 148 State St., Boston 9; *for mail*, 346 Main St., Winchester, Mass.
Rivoira, Edilio J. (J'34) (CMR), Elec. Engr., Cincinnati Milling Mch. Co.. Oakley St., Cincinnati; *for mail*, 3822 Thornton Dr., Silverton, Ohio.
Rizzo, Joseph F. (J'41) (ACM), Pres., Managing Engr., Am. Tech. Inst., 305 N. 15th St., Philadelphia, Pa.
Roach, Jack W. (J'36) (CKL), Asst. Supt., Okmulgee Refinery, Phillips Petroleum Co.; *for mail*, Box 2351, Okmulgee, Okla.
Roach, Jere Henry (J'38) (CJNM), Plant Engr., Penn Brass & Copper Co.; *for mail*, R.D. 3, 1105 Arlington Rd., Erie, Pa.
Roane, Eugene Stuart, Jr. (J'42) (SKFAHC), Capt., U.S.M.C.R., Overseas; *for mail*, 603 Montvale Ave., Richmond 22, Va.
Robb, Budd R. (J'40), Aviation Machinist's Mate, 1st Class, Assembly & Repair, Bldg. 676, Naval Air Sta., Pensacola, Fla.
Robb, Charles A. ('14; '35) (SHC), Mech. Engrg. Dept., McGill Univ., Montreal, Que., Can.
Robb, Harry Walter ('40) (CJ), Stands. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Robba, William H. ('25; '30) (OSNKEB), Indus. Engr., 3231—213th St., Bayside, N.Y.

Robbins, Albert L. (J'45) (NJYBML), Jr. Engr., Design, Wyrstapes Co., 200 Hudson St., New York; *for mail*, 417 Ave. F, Brooklyn 18, N.Y.
Robbins, Harold E., Jr. (J'37) (ABC), Serv. Rep., Pratt & Whitney Aircraft Div., United Aircraft Serv. Corp., East Hartford; *for mail*, 47 Granada Terrace, New London, Conn.
Robbins, Irving Paul (J'32) (NOHSBC), Mech. Engr., Haughton Elev. Co., 671 Spencer St., Toledo 9, Ohio.
Robbins, John L. ('39), 314 Main St., Great Barrington, Mass.
Robbins, Lloyd Wilton (J'44), Route 1, Orange, Calif.
Robbins, Louis (J'42) (ANBSFO), Aero. Engr., Aero. Eng. Lab., Naval Base, Philadelphia 12; *for mail*, 2609 N. 33rd St., Philadelphia 32, Pa.
Robbins, Vernon Ellsworth (J'44) (NMBHK), 1536 N. Lorel Ave., Chicago 51, Ill.
Robbins, William Francis ('20; '35) (CSD), Plant Engr., Graton & Knight Co., 356 Franklin St., Worcester 4; *for mail*, 37 Forest, Worcester 5, Mass.
Robert, James Marshall ('20) (BCS), *Manager*, '43-'45, *Director-at-Large*, '45-'46; Dean of Engrg., Tulane Univ., New Orleans 15, La.
Robert, John ('20; '26) (CDMNSW), Ch. Engr., Globe Roofing Products Co., Inc., 2207 Schrage Ave., Whiting, Ind.
Robert, Jules Henry (J'18) (BHS), Prof. Applied Mechanics, Instr., Kan. State College, Manhattan, Kan.
Robert, L. W., Jr. ('14; '23), 706 Bona Allen Bldg., Atlanta, Ga.
Roberts, A. Perry ('21; '25; '30) (JOC), Mech. Engr., 233 Broadway, New York 7, N.Y.; *for mail*, 132 Tenafly Rd., Tenafly, N.J.
Roberts, Arthur ('42), 2870 Woodbury Rd., Shaker Heights, Ohio.
Roberts, Arthur, Jr. ('36; '40) (BCDHM), Ch. Engr., Lynchburg Fdy. Co., Lynchburg, Va.
Roberts, Arthur Reginald ('17) (NBSKFA), Prof. & Chmn., Dept. of Mech. Engrg., Faculty of Engrg., McGill Univ., Sherbrooke St. W., Montreal, Que., Can.
Roberts, Cary Russell ('33; '35) (ABJFNG), Capt., Army Air Forces, Aircraft Maint. Officer, Goodfellow Field, San Angelo, Tex.; *for mail*, 813 Elberon Ave., Dayton 3, Ohio.
Roberts, Chapin ('14; '21) (OS), Constr. Mgr., Sears, Roebuck & Co., 925 E. Homan Ave., Chicago; *for mail*, 317 N. Euclid Ave., Oak Park, Ill.
Roberts, Charles Lewis (J'43) (A), 122 Fairlawn Rd., Louisville 7, Ky.
Roberts, Charles P. ('26; '35) (AE), Assoc. Prof., Ohio State Univ., Columbus, Ohio.
Roberts, Colin Anson (J'46) (BHJ), 95 Washington Ave., Providence, R.I.
Roberts, Cyril Herbert ('42) (BMRNCF), M.M., Blaw-Knox Co., Main St., Martins Ferry; *for mail*, Box 149, R.D. 2, Dillonvale, Ohio.
Roberts, Edward Drayton ('26) (BDL), Mech. Designer, Distillers Co., Ltd., 21 St. James Sq.; *for mail*, BM/WSVN London, W.C. 1, England.
Roberts, Edward E. (J'38) (ALM), c/o Frank Parris, 1635 N. Main Ave., Scranton, Pa.
Roberts, Edward H. (J'36) (ABJMY), Engr., Refrigerator Cabinet Engrg. Dept., Gen. Elec. Co., 2901 E. Lake Rd., Erie, Pa.
Roberts, Frank Bostwick (J'41) (CFLSZ), Indus. Engr., E. I. du Pont de Nemours & Co., Wilmington; *for mail*, 1411 Hamilton St., Wilmington 73, Del.
Roberts, Frank Kendall (J'38) (L), Mech. Engr., Anaconda Wire & Cable Co.; *for mail*, 28 Whitman St., Hastings-on-Hudson 6, N.Y.
Roberts, Fred Gordon ('41) (CDHJMN), Div. Works Mgr., Natl. Carbon Co., Inc., Cleveland 7; *for mail*, 1114 Homewood Dr., Lakewood 7, Ohio.
Roberts, Frederick Farr (J'45) (NHS), Demonstrator, Univ. of Toronto, Toronto 5; *for mail*, 99 King St. N., Waterloo, Ont., Can.
Roberts, Geoffrey D. (J'41) (M), 39 Kirkland St., Cambridge 38, Mass.
Roberts, George Samuel (J'41) (LJKB), Design Engr., Ethyl Corp., Box 341, Baton Rouge 1; *for mail*, 1520 Olive St., Baton Rouge 11, La.
Roberts, Hugh Edward (J'43) (SRBAFJ), Test Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.; *for mail*, 37 Washington Ave., Whitefish, Mont.
Roberts, James E. (J'45), U.S. Army, 2810 Glenhurst Ave., Minneapolis 16, Minn.
Roberts, James Frank ('20; '27; '35) (HSJ), Mgr., Hyd. Dept., Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Roberts, James L. ('25) (SJK), Sponsor Engr., Turbine Engrg. Dept., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 1552 Regent St., Schenectady 8, N.Y.
Roberts, John ('34) (R), Managing Dir., Natl. Rys. Munitions, Ltd., 360 Sabastapole St., Montreal, Que., Can.
Roberts, John Robert (J'46) (CDK), 3232 Central Ave., Indianapolis 5, Ind.
Roberts, Joseph Harry Newton ('22; '27) (ONM DB), Sr. Engr., Joseph H. Roberts & Associates, Box 1525, 77 Bank St., Waterbury 89, Conn.
Roberts, Jules Darius ('27) (AEFK), V.P., Mountain Fuel Supply Co., 36 S. State St., Salt Lake City 10, Utah.
Roberts, Karl Davis ('41) (NT), Plant Engr., Plymouth Cordage Co., Plymouth, Mass.
Roberts, Kenneth Clark ('41) (LDHJ), Head Struc. Engr., Tenn. Valley Authority, Knoxville, Tenn.
Roberts, Melvin Robert (J'39) (YMNCHW), Ch. Engr., Precision Specialties Mfg. Co., 210 N. Western, Los Angeles 4; *for mail*, 1641 Crenshaw Blvd., Los Angeles 6, Calif.
Roberts, Montague H. ('12; '17), 124 Linden Ave., Englewood, N.J.
Roberts, Ralph Tanfield ('45) (SFKD), Asst. Div. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Roberts, Reginald Thomas (J'44) (BCS), Field Engr., Westinghouse Elec. Corp., 10 High St., Boston; *for mail*, 24 Parker St., Saugus, Mass.
Roberts, Richard Francis ('15; '35) (ZLJS), Indus. Sales Engr., Brown Instruments Div., Minneapolis-Honeywell Regulator Co., 221—4th Ave., New York 3, N.Y.; *for mail*, 136 Cleveland St., Orange, N.J.
Roberts, Samuel B. ('19; '25; '29) (SFENCT), Ch. Engr., Celanese Corp. of Am., 180 Madison Ave., New York 16, N.Y.; *for mail*, Anderson Rd., Greenwich, Conn.
Roberts, W. Stewart (J'40) (ANS), Sr. Project Engr., Allison Div., Gen. Motors Corp., Indianapolis; *for mail*, 2947 N. Meridian St., Indianapolis 8, Ind.
Roberts, Walter H. ('19; '33) (EHSF), Sr. Engr., U.S. Engrs., Box 97, Memphis 1, Tenn.
Roberts, Webster Charles (J'42) (AEM), Ensign, U.S.N.R., 947 S. Marengo, Pasadena, Calif.
Robertshaw, Clarence Wilfred (J'29) (ABL), V.P. in Research, Dir., Robertshaw Thermostat Co., Youngwood; *for mail*, 524 Harrison Ave., Greensburg, Pa.
Robertson, Alexander Martin (J'37) (NDJYLM), Draftsman, Natl. Biscuit Co., 449 W. 14th St., New York 11, N.Y.
Robertson, Baxter ('28) (FHKL), Mech. Engr., Aetna Engrg. Co., 716 Columbus Ave., Boston 20; *for mail*, 25 Hawthorne Ave., Arlington, Mass.
Robertson, Billy Perkins (J'43) (BAKHNO), Research Engr. (Lubrication), Research Labs., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Robertson, Burton J. ('21) (EFA), Prof., Internal Combustion Engs., Univ. of Minn., 2031 University Ave., S.E., Minneapolis 14, Minn.
Robertson, Charles A. ('23; '35) (SEFBKJ), Engr., Steam Turbine Dept., Allis-Chalmers Mfg. Co., 1113 S. 73rd St., West Allis 14, Wis.
Robertson, Eugene N. (J'42) (BGNWY), Test. Engr., Rohm & Haas Co., Bristol; *for mail*, 18 S. Chancellor St., Newtown, Pa.
Robertson, Gay Autricht ('27) (EFK), Ch. Engr., Atlantic Co., Box 1417, Atlanta, Ga.
Robertson, J. Douglas ('20; '26; '33) (ACMNT), Pres., Treas., Mt. Hope Mchy. Co., 42 Adams St., Taunton, Mass.
Robertson, James Mueller (J'42) (BHAK), Asst. Prof. Engrg. Mechanics, The Pa. State College, State College, Pa.
Robertson, John Marshall ('21; '33) (FJYZMC), Mgr., Indus. Dept., Houston Natural Gas Corp., Box 1188, Houston 1, Tex.
Robertson, Norman Finch (A'30) (CAHNZJ), Pres., Gen. Mgr., John Robertson Co., Inc., 133 Water St., Brooklyn 1, N.Y.
Robertson, Ralph A. (J'39), Ensign, U.S.N.R., N.Y. Naval Shipyard, Brooklyn; *for mail*, 1325 York Ave., New York, N.Y.
Robertson, Robert James ('44) (BHS), Ch. Designer, Pump Dept., Thompson's Engrg. Co.; *for mail*, 18 Myring St., Castlemaine, Victoria, Australia.
Robertson, Robert John ('45) (CFSKHB), Supt., La. Sta., Gulf States Utilities Co., 340 Florida St.; *for mail*, 6130 Government St., Baton Rouge, La.
Robertson, Robert Roy ('19) (HORS), Retired; 2291 Moreno Dr., Los Angeles 26, Calif.
Robertson, Roy C. (J'41) (EF), Box 211, Dante, Va.
Robertson, Stewart Fisher ('38; '40) (OCSL EH), Partner, Charge of Mech. Engrg., Whitman, Requardt & Associates, 1304 St. Paul St., Baltimore 2, Md.
Robertson, William Tennant ('44) (CEMO), Design Engr., Humble Oil & Refining Co., Houston 1, Tex.
Robeson, A. M. ('95), Retired; c/o Standard Bank of S. Africa, 10 Clements Lane, London, E.C. 4, England.
Robeson, Philip Baker (J'41) (MZCL), 204 Westminster Ave., Merchantville, N.J.
Robie, Theodore Mead ('43) (ESRCF), Gen. Diesel Sales Div., Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Robin, Philip Theodore ('40) (NDJC), V.P., Exec. Mgr., Clamshell Bucket Sales Corp., 122 E. 42nd St., New York 17; *for mail*, 126 Wilson Rd., Valley Stream, N.Y.
Robins, Alexander (J'43) (ACO), Draftsman, DeWitt Griffin Assoc., 303 N.E. 4th, Portland; *for mail*, Apt. 7596, 10817 Lake, Vanport, Ore.
Robins, John Wall ('42) (ZA), Devel. Engr., Mason-Neilan Regulator Co., 1190 Adams St., Boston 24; *for mail*, 412 Hartford St., Westwood, Mass.
Robins, Vern Thomas ('44) (FMP), Asst. Gen. Maint. Foreman, Pure Oil Refinery; *for mail*, Cabin Creek, W.Va.
Robinson, Albert William ('39) (SL), Chicago Dist. Mgr., Farr Co., 2729 W. 19th St., Chicago; *for mail*, 166 N. Scoville Ave., Oak Park, Ill.
Robinson, Almon L. ('19; '35), Engr., Charge of Tool Design, L. C. Smith Typewriter Co., 701 E. Washington St.; *for mail*, 109 Southview Rd., Syracuse, N.Y.
Robinson, C. Stanley ('24; '31; '35) (SFECLD), Power Engr., E. I. du Pont de Nemours & Co., Nemours Bldg., Wilmington 98; *for mail*, 16 Cragmere Rd., Cragmere, Wilmington, Del.
Robinson, Charles H. ('26; '42) (BJS), 1436 Easton Ave., Bethlehem, Pa.
Robinson, Charles S. L. ('37; '44) (BHS), Engrg., Shipbldg. Div., Bethlehem Steel Co., Quincy; *for mail*, 213 Waltham St., West Newton 65, Mass.
Robinson, Curville J. (J'37) (KLS) Sales & Test Engr., Croll-Reynolds Engrg. Co., Inc., 17 John St.; *for mail*, 2475 Palisade Ave., Spuyten Duyvil, New York, N.Y.
Robinson, Darrell N. (J'37) (CLP), 1119 Huntington Dr., South Pasadena, Calif.
Robinson, Dwight P. ('19), Pocasset, Cape Cod, Mass.
Robinson, Edmund M. ('27; '35) (SYKOBZ), Res. Engr., Travelers Indemnity Co., 120 N. Michigan Ave., Saginaw, Mich.
Robinson, Edward Arthur ('41), Cons. Engr., 208-353rd St., Nicholas Bldg., Montreal, Que., Can.
Robinson, Emerson Jos. (J'35) (AJM), Mech. Engr., New England Butt Co., 304 Pearl St., Providence, R.I.; *for mail*, 31½ Forest St., Milford, Mass.
Robinson, Eric ('44) Superheater Co., Ltd., St. Margaret's Rd., Bowdon, Altrincham, Cheshire, England.
Robinson, Ernest Leffert ('23) (SBCEHA), *Melville Medallist*, '44; Struc. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Robinson, Ernest W., Jr. (J'42) (AJM), Jr. Prod. Engr., Goodyear Aircraft Corp., Akron; *for mail*, Box 232, Route 9, Akron 1, Ohio.
Robinson, George Elliott (J'45) (BJA), Ch. Test Engr., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge; *for mail*, 48 N. Pearl St., Brockton 55, Mass.
Robinson, Harold Irwin (J'42) (CDHSLM), V.P., Charge of Sales, Pittsburgh Pipe Cleaner Co., 311 N. 20th St., Philadelphia 3, Pa.
Robinson, Harry Delmer, Jr. (J'45) (MCKO), Asst. Mgr., C. L. Robinson Ice & Cold Storage Co., N. Cameron St.; *for mail*, 514 Amherst St., Winchester, Va.
Robinson, Harry Heistand, Jr. (J'44), 621 Manor Rd., Staten Island 2, N.Y.
Robinson, Henry Meyer ('27; '31) (FCH), Engr. (P-4), Ft. Worth Regional Office, Fed. Power Comm., 412 Neil P. Anderson Bldg., Ft. Worth 2; *for mail*, 2142 Kessler Pkwy., Dallas 8, Tex.
Robinson, Howard Addison ('45), Ch. Physicist, Research Lab., Armstrong Cork Co., Lancaster, Pa.
Robinson, I. V. ('42), Carisbrooke, Walton-on-Thames, England.
Robinson, James Milton (J'44), Spence-Rigolo, 677—5th Ave., New York, N.Y.
Robinson, James P. (J'43) (CJMN), Cpl., Army Air Forces, Sqd. F, 4000th Base Unit (AB), Wright Field, Dayton, Ohio; *for mail*, 120 Fort Hill Ave., Lowell, Mass.
Robinson, James R. ('05), Pres., Robinson Vent. Co., Zelienople, Pa.
Robinson, John L. (J'27), V.P., Mgr., W. Mich. Consumers Co., 706 Hackley Bank Bldg., Muskegon, Mich.
Robinson, John Thomas (J'45), Res. Engr. & Mgr., Vinco Corp., 9 Rockefeller Plaza, New York 20, N.Y.
Robinson, John W. (J'45) (BNHMC), Devel. Engr., Skilsaw, Inc., 5033 Elston Ave., Chicago 30; *for mail*, 4859 N. Wolcott Ave., Chicago 40, Ill.
Robinson, John William (J'40) (CEHLSZ), Dist. Mgr., Leeds & Northrup Co., 116 New Montgomery St., San Francisco 5, Calif.
Robinson, Joseph ('23) (CMROJY), Pres., Joseph Robinson, Inc. & Robinson Connector Co., 1907 Park Ave., New York 35, N.Y.
Robinson, Louis Gilmore ('05; '13) (JOLZF), Owner, Louis G. Robinson Labs., 219 McFarland St., Cincinnati 1, Ohio.
Robinson, Manuel G. ('40) (AEHBFJ), Mech. Engr., 20 Dorinda Circle, Lynn, Mass.
Robinson, Max Brewster ('35) (MFCR), Dean of Engrg., Fenn College, Cleveland 15, Ohio.
Robinson, P. Arthur ('45) (CEFJNS), Traffic Mgr., Freihofer Baking Co., 20th & Indiana Ave., Philadelphia 32; *for mail*, 112 Decatur Rd., Havertown, Delaware Co., Pa.
Robinson, Parke D. ('42) (SFA), Research Engr., Stainless & Steel Products Co., 1000 Berry Ave., St. Paul 4; *for mail*, 5533 Pillsbury Ave. S., Minneapolis 9, Minn.
Robinson, Parker Morrill ('45) (SOEC), Cons. Engr., 2523 Pacific Ave., San Francisco 15, Calif.
Robinson, Paul Clark (J'38) (NGBW), Gen. Supt., Arkell & Smiths; *for mail*, R.F.D. 1, Canajcharie, N.Y.
Robinson, Philip James ('42) (SFKEO), Steam Turbine Specialist, Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Robinson, Robert Frank (J'45) (ABKZ), Pvt., 1st Class, Corps of Engrs., U.S.A., (Assigned to Clinton Labs., Monsanto Chem. Co., as Asst. Instrument Engr.), Manhattan Dist., Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.; *for mail*, R.F.D. 3, West Salem, Ohio.

Robinson, Robert R. ('23; '30) (BCM), Partner, Charge of Design & Mfr., W. D. Howze Sales Co., 422 Alpine St.; *for mail,* 2402 Victoria Ave., Los Angeles, Calif.

Robinson, Samuel Murray (H'41), Admiral, U.S.N., Retired, Administrator, Webb Inst., Webb & Sedgwick Ave., Bronx 63, N.Y.

Robinson, Stephen Bernard, Jr. (J'43) (BJC), Stress Analyst, Aerojet Engrg. Corp., 285 W. Colorado St.; *for mail,* 2305 Ridgeway Rd., San Marino 9, Calif.

Robinson, Theodore Bliss (J'43) (CMOD), Electronics Technician's Mate, 2nd Class, U.S.N.R., Radar Maint. Sch. for Radar Operators, Point Loma, San Diego, Calif.

Robinson, Walter (J'41) (EFKBCS), Managing Engr., Ulster Fuel Oil, Heat & Power Co., Box 204, Kingston, N.Y.

Robinson, Walter E. ('28; '35) (ACM), Promotion Mgr., *Machinery,* Indus. Press, 148 Lafayette St., New York, N.Y.

Robinson, Walter Percil ('13; '26) (CLMD), Pres., McCarthy & Robinson Ltd., 2 Neville Park Blvd., Toronto 8, Ont., Can.

Robinson, Ward M. ('28) (CMDGT), V.P., Gen. Mgr., Talon, Inc., Arch St., Meadville, Pa.

Robinson, Watson Vernon (J'44) (NMCB), Mch. Designer, SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34, Pa.

Robinson, Wilford J. ('45) (SKFCJ), Watch Engr., Potomac Elec. Power Co., 10th & E Sts., Washington, D.C.; *for mail,* 4729—16th St. N., Arlington, Va.

Robinson, William Alfred (J'40) (ASHJKE), 1515 Guardian Bldg., Cleveland, Ohio.

Robson, Charles John ('45) (NOBMZJ), Mech. Engr., Research Lab., Pittsburgh Plate Glass Co., Creighton; *for mail,* 535—7th St., Oakmont, Pa.

Robson, W. J. (J'35), 40 Colin, Toronto, Ont., Can.

Roby, C. F. ('22; '26; '35) (ACM), Managing Dir., Cincinnati Milling Mchs. Ltd., Woodlands Farm Rd., Tyburn, Birmingham, England.

Rochester, William L. ('42) (JMC), Pres., Rochester Ropes, Inc., Culpeper, Va.

Rockefeller, Harry E. ('27; '36), Mgr., Process Devel. Dept., Linde Air Products Co., 30 E. 42nd St., New York, N.Y.

Rockett, Harold Challinor ('41) (CMS), Comdr., U.S.N., Regional Supvr., Terminal Island, San Pedro, Calif.; *for mail,* 511 Forest Ave., Rye, N.Y.

Rocklein, George Washington, Jr. (J'39) (LNBKSG), Mech. Engr., Designer, Charles Pfizer & Co. Inc., 11 Bartlett St., Brooklyn 6; *for mail,* 44—72nd St., Brooklyn 9, N.Y.

Rockwell, Ralph Alvin ('45) (HPZ), Dir. of Engrg., Mason-Neilan Regulator Co., 1190 Adams St., Boston 24; *for mail,* 20 Park View Dr., Hingham, Mass.

Rockwell, Robert L. ('23) (EMS), Regional Engr., Smaller War Plants Corp., 4544 Stuart Bldg., Seattle 1; *for mail,* Port Madison, Winslow, Wash.

Rockwell, Samuel Forbes ('12) (CT), Pres., Treas., Davis & Furber Mch. Co., North Andover, Mass.

Rockwell, Theo. F. ('41) (FOS), Cons. Engr., Glenover Pl., Pittsburgh 15, Pa.

Rockwell, Willard F. ('13; '19; '21) (CFMNOZ), Chmn. of Bd., Pres., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.

Rockwood, Charles H. (J'41) (NZBJMY), Asst. Engr., Elec. Control Div., Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1; *for mail,* 1003 S. 76th St., West Allis 14, Wis.

Rockwood, George I. ('91; F'36), *Manager,* '03-'06, *Vice President,* '24-'25; Chmn., Bd. of Trustees, Worcester Poly. Inst.; *for mail,* 2 Military Rd., Worcester 5, Mass.

Roddis, Louis Harry, Jr. (J'44) (ACH), 200 Old Gulph Rd., Wynnewood, Pa.

Roddy, Fred M. (J'28), Cumberland Engrg. Co.; *for mail,* Box 216, Providence, R.I.

Rodenbaugh, Donald Irvin ('32; '42) (FKZDMC), Furnace Engr., Design & Constr., Hazel-Atlas Glass Co., Market St., Zanesville, Ohio.

Rodenbaugh, Henry Nathan ('22) (RCSOJM), Vice-Chmn., Price Adjustment Bd., Philadelphia Ord. Dist., Army Serv. Forces, 150 S. Broad St., Philadelphia; *for mail,* Seven Pines, Elbow Lane, Haverford, Pa.

Rodenburg, Carl Euler ('43) (SKFHO), Engr., Design & Opera., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.; *for mail,* 427 Center St., Wood-Ridge, N.J.

Roderick, Edward M. ('27; '35; '35) (CSOFMK), Capt., Ord. Dept., U.S.A., Asst. Ch., Supply Branch, Ord. Office, Hdq., 2nd Serv. Comd., Governors Island, New York 4; *for mail,* Carteret Hotel, 208 W. 23rd St., New York 11, N.Y.

Rodgers, Arthur Cecil (J'36) (MSZ), Engr., New England Tel. & Tel. Co., 50 Oliver St., Boston; *for mail,* 342 Prospect St., Norwood, Mass.

Rodgers, Edward G. (J'38) (CJL), Engr., Constr. Supt., Gen. Chem. Co.; *for mail,* Box 348, Green Ridge, Pa.

Rodgers, Oliver Eliot (J'41) (BAMCEJ), Mgr., Gas Turbine Design Engrg., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Essington; *for mail,* 535 Riverview Ave., Swarthmore, Pa.

Rodgers, Walter Anthony ('44) (CBL), Pres., Rodgers, Keefe & Associates, 115 Broadway, New York 6; *for mail,* 20 S. Lakeside Dr., Baldwin, L.I., N.Y.

Rodin, Max Bernhard (J'41) (LJZACB), Co. Engr., Rodin Bros., 6039 Cottage Grove, Chicago 19, Ill.; *for mail,* 920 W. Jeff Blvd., South Bend, Ind.

Rodman, Nicholas ('22; '26; '35), Ch. Mar. Inspr., N.Y., New Haven & Hartford R.R., 132nd St. & Willis Ave., New York, N.Y.

Rodman, Wilmot Lee (J'42) (ABC), Sr. Structures Engr., Design, Analysis, Ranger-Lark Div., Fairchild Eng. & Airplane Corp., 184-10 Jamaica Ave., Jamaica; *for mail,* 114-40—220th St., St. Albans 9, L.I., N.Y.

Rodolf, Carl Francis ('44) (RSHEF), Water Supply Inspr., So. Pacific Co., Rm. 1035-B, 65 Market St., San Francisco 5, Calif.

Rodriguez, Charles Julian (J'45) (ERJN), Project Engr., Sperry Products, Inc., 1505 Willow Ave., Hoboken, N.J.; *for mail,* 38 E. 98th St., New York 29, N.Y.

Rodriguez, Zonzalo G. (J'41) (BMS), 27 Arzuaga St., Rio Piedras, P.R.

Roe, Joseph W. ('02; F'41) (MCD), *Melville Medallist,* '29; Retired; Southport, Conn.

Roe, Kenneth Andrew (J'45), Burns & Roe, Inc., 233 Broadway, New York; *for mail,* 168 Westervelt Ave., Baldwin, N.Y.

Roe, Ralph Coats ('32) (EKSO), Pres., Burns & Roe, Inc., 233 Broadway, New York, N.Y.; *for mail,* 1 N. Brae Court, Tenafly, N.J.

Roedel, John Kennedy ('38) (BHKSLJ), Stand. Oil Co. of Ind., Casper, Wyo.

Roeder, Carl Henry (J'37) (FEKBNY), Automotive Engr., Natl. Bur. of Stands., Van Ness & Connecticut Aves., Washington 25, D.C.

Roeder, George Philip (J'38) (SKFMJA), Sr. Draftsman, Fed. Shipbldg. & Drydock Co., Kearny; *for mail,* Mountain Ave., Murray Hill, N.J.

Roehm, Jack Moody (J'34) (CABHJ), Dir. of Engrg., Buehler & Co., 1607 Howard St., Chicago, 26, Ill.

Roehm, Perry R. (J'38) (CAMNYS), Sales Mgr., Barden Corp., E. Franklin St., Danbury, Conn.

Roehm, William F., Jr. (J'41) (ER), 1243 Chicago Ave., Evanston, Ill.

Roemer, Herbert (J'35), 1st Lt., Coast Arty. Corps, Coast Arty. Sch., Ft. Monroe, Va.

Roemmele, Herbert Frederick ('25; '36) (SEKFLZ), Assoc. Prof. Mech. Engrg., The Cooper Union, 41 Cooper Sq., New York 3, N.Y.

Roesch, Daniel ('25) (EAB), Prof. Automotive Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago 16; *for mail,* 2136 W. 108th Pl., Chicago 43, Ill.

Roesch, Frank Pierrie ('19) (FR), Retired; 1413 E. 50th St., Chicago 15, Ill.

Roesel, Arno Fritz ('26; '34; '35) (STBK), Project Engr., Am. Bemberg Corp.; *for mail,* 611 N. Main St., Elizabethton, Tenn.

Roesel, Louis Christopher ('22; '26; '35), Ch. Engr., Ga. Ice Co., 431 Harmon St.; *for mail,* 1301 Washington Ave., Savannah, Ga.

Roesler, David George (J'45), 732—8th Ave., S.W., Rochester, Minn.

Roessle, Rudolph Bliesath, Jr. (J'41) (CDM), 424 Vine St., Hudson, Wis.

Roessler, Edward William ('44) (CJK), Sec. Hdq., Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.

Roets, James Byron Pyre (J'42) Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford; *for mail,* 1654 Main St., Glastonbury, Conn.

Roetzer, Alfred Arnold (J'36) (EFJKS), 79 Garland St., Everett 49, Mass.

Roffman, Kenneth A. (J'42) (KNZSEH), Westerlin & Campbell Co., 1113 W. Cornelia Ave., Chicago 13, Ill.

Rogers, A. Scobey, Jr. (J'41) (MNCJDB), Partner, Rogers Engrg. Serv., 2808 Westwood Ave., Nashville 5, Tenn.

Rogers, Alfred William ('41) (CFS), Div. Engr., E. I. du Pont de Nemours & Co., Orange; *for mail,* 5148 Lewis Rd., Port Arthur, Tex.

Rogers, Arlester B. ('27) (FRS), Fuel Consultant, United East. Coal Sales Corp., 1132 Lincoln-Alliance Bank Bldg., Rochester 4, N.Y.

Rogers, Benjamin F. ('22; '24) (CFKS), Results Engr., Luzerne County Gas & Elec. Corp., 247 Wyoming Ave.; *for mail,* 650 Rutter Ave., Kingston, Pa.

Rogers, Donald Atwater ('24; '33) (BDHKLS), Advisory Engr., Consultant, Solvay Process Co., Hopewell, Va.

Rogers, Edward Louis (J'37) (OMCBHS), Plant Engr., S. Morgan Smith Co., Lincoln & Hartley Sts., York, Pa.

Rogers, Eugene Lawrence (J'43) (CMAKO), Box 707, Jackson, Miss.

Rogers, Frank H. ('17) (BHM), Sales Mgr., I. P. Morris Dept., Baldwin Loco. Works, Paschall Sta. P.O., Philadelphia, Pa.

Rogers, Fred Stillman ('40) (NBMJ), Prof., Mch. Design, College of Engrg., Cornell Univ.; *for mail,* 110 W. Court St., Ithaca, N.Y.

Rogers, George A. (J'34) (JKR), Lt., S(A), U.S.N.R., Naval Air Sta., Navy 128, Box 1358, F.P.O., San Francisco, Calif.

Rogers, George A. (J'42) (CKS), Lt. Comdr., U.S. Maritime Serv.; *for mail,* Apt. 2, Park Plaza Apts., 632 Courtland Park Ave., Yonkers, N.Y.

Rogers, J. Frank (J'45) (DJM), Sales Engr., George J. Fix Co., 2413 Commerce St., Dallas 1, Tex.

Rogers, Joseph Elsworth ('24; '33; '35) (CAEM), V.P., Rogers Radio Broadcasting Co., 37 Bloor St. W., Toronto, Ont., Can.

Rogers, Lewis Chubb ('12) (OFAESD), Cons. Engr., Wolverine Shoe & Tanning Corp., Rockford; *for mail,* 824—28th St., S.W., Grand Rapids 9, Mich.

Rogers, Ralph W. ('14), Tidioute, Pa.

Rogers, Walter E. (J'38) (CJS), Swett & Crawford Ins., 1065 Stuart Bldg.; *for mail,* 1919 Broadway N., Seattle, Wash.

Rogers, Wayne Carlton (J'37) (BFJ), Design Engr., Riley Stoker Corp., 9 Neponset St.; *for mail,* 143 W. Boyleston St., Worcester 5, Mass.

Rogers, Willard Lewis (J'43) (SBKENA), Instr. in Mech. Engrg., Iowa State College; *for mail,* 200 E. 13th St., Ames, Iowa.

Rogers, Wm. D. (J'37), Asst. Engr., Larrowe Milling Co., Div. of Gen. Mills, Inc., Rossford; *for mail,* 2415 Pickle St., Toledo, Ohio.

Rogerson, John B. ('42) (CFEY), Refinery Supt., Lion Oil Co., El Dorado, Ark.

Rohlin, Victor Alfred ('28; '35) (S), Works Mgr., Cochrane Corp., 17th & Allegheny Ave., Philadelphia; *for mail,* 311 Bickley Rd., Glenside, Montgomery Co., Pa.

Rohrbough, Jack Curtis (J'41) (ACM), 804 S. 51st St., Omaha, Neb.

Rohrer, Albert L. ('96), Retired; 307 Wyoming Ave., Maplewood, N.J.

Rohrer, Josiah Hershey ('41) (SKECO), Engr., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2; *for mail,* 3903 Vaux St., Philadelphia 29, Pa.

Rohrhurst, William ('27; '34; '35) (SFCKZ), Asst. Mgr., Power Div., Calco Chem. Div., Am. Cyanamid Co.; *for mail,* 537 Watchung Rd., Bound Brook, N.J.

Rohrich, Harold Arthur ('29) (BGHMN), Mch. Designer, Lehmann Mch. Co., 3560 Chouteau Ave.; *for mail,* 1433 Pine St., St. Louis, Mo.

Rohsenow, Warren M. (J'41) (KBESHF), Asst. Prof., Mech. Engrg. Dept., Mass. Inst. of Tech., Cambridge, Mass.

Roig, J. Adalberto ('24; '30; '35) (C), Partner, Antonio Roig, Sucesores S. en C., Humacao, P.R.

Rojeski, Frank Robert (J'46), Ensign, U.S.N.; 348 Wildwood Ave., Piedmont 11, Calif.

Roland, John ('42) (FBK), 2412 E. 16th St., Indianapolis 2, Ind.

Roland, P. W. ('16; '27; '35) (CLM), Asst. Supt., Arkell Safety Bag Co., 67 N. 11th St., Brooklyn; *for mail,* 27-14—167th St., Flushing, N.Y.

Rolland, George Alexander ('21; '35) (TSHFED), Engr., Warwick Mills, 77 Franklin St., Boston 10; *for mail,* 163 Milton St., Boston 24, Mass.

Rolle, Carl ('36; '41) (BJM), Mech. Engr., Internatl. Nickel Co., Inc., 67 Wall St., New York, N.Y.

Roller, Charles August (J'46), 77 Dartmouth St., Forest Hills, N.Y.

Rollins, Fitzhugh Smith, Jr. (J'30) (AEBZC), Research Engr., Calif. Research Corp., Richmond; *for mail,* 1111 Milvia St., Berkeley 7, Calif.

Rollins, James Thaydas (J'46) (CE), Box 426, Selman City, Tex.

Rollins, John Pletcher (J'39) (SKBNR), Instr. in Mech. Engrg., Rensselaer Poly. Inst., Troy, N.Y.

Rollins, Lewis Mortimer ('16; '19) (KYSFDB), Owner, Firm of Lew M. Rollins, 455 Summit Ave., Eau Claire, Wis.

Rollman, Martin Edw. (J'32), Jr. Engr., Prod., Cincinnati Milling Mch. Co., Oakley; *for mail,* 1734 Andina Ave., Cincinnati, Ohio.

Rollo, William Smith ('44), Asst. Engr., Pur. Dept., Atlantic Refining Co., 260 S. Broad St., Philadelphia, Pa.

Rollow, Douglas K. (J'38) (ACMN), Engrg. Designer, Lockheed Aircraft Corp., Burbank; *for mail,* 4918 Placidia Ave., North Hollywood, Calif.

Rollow, James Grady ('15; '22) (SEFB), Supvg. Engr., So. Calif. Gas Co., Box 3249, Terminal Annex, Los Angeles 54, Calif.

Rolls, L. Stewart (J'44) (AEH), 696 S. 3rd St., San Jose 12, Calif.

Romaine, Millard ('22; '35) (CAG), Asst. to Pres., Cincinnati Milling Mch. Co., 4701 Marburg Ave., Cincinnati 9, Ohio.

Romanach, Juan Angel ('23; '35) (BDGHRY), Loco. Shops Supt., United Rys. of Havana, Box 450, Havana, Cuba.

Romanchuk, Peter Alexis (J'44) (NLJAOB), Mech. Design Engr., Reeves Sound Labs., 215 E. 91st St., New York 28, N.Y.; *for mail,* 7829 Bergenline Ave., North Bergen, N.J.

Romanelli, Otto Charles (J'46) (BGS), 9026 Farragut Rd., Brooklyn, N.Y.

Romann, John H. ('16) (JNTMOL), V.P., Cons. Engr., U.S. Engrg. & Mfg. Co., 188 W. Randolph St., Chicago 1, Ill.

Romanowich, Richard Ramon (J'41) (AEJKS), Lt. (j.g.), U.S.N.R., Engr. Officer, Air Corps, VB-100, F.P.O., San Francisco, Calif.; *for mail,* 4522 Midway Ave., Dayton 7, Ohio.

Romeiser, Adam Henry (J'38) (CBMNSD), Asst. to Factory Supt., Am. Laundry Mchy. Co., 1800 N. Spaulding Ave.; *for mail,* 2302 N. Kedzie Blvd., Chicago 47, Ill.
Romie, Fred E. (J'43) (KLA), Research Engr., Univ. of Calif.; *for mail,* 2063 Los Angeles Ave., Berkeley 7, Calif.
Romigh, Orin L. (J'34) (JBHA), Contact Metallurgist, Chrysler Corp., Detroit; *for mail,* 1272 Princeton, Berkley, Mich.
Rondo, Anthony Andrew ('45) (ACOLJF), Designer, Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 56 Rockdale St., Mattapan 26, Mass.
Ronkanen, Vaino Alexander ('24; '31; '35) (BWTONM), Factory Inspr., Dept. of Labor, State of N.Y., State Office Bldg., Albany 1; *for mail,* 41 Notre Dame St., Glen Falls, N.Y.
Roome, Bertram Reinagle, Jr. (J'44) (CMSK), Ensign, U.S.N.R., U.S.S. *Chester,* "M" Div. Officer, Philadelphia Gp., 16th Fleet, Naval Base, Philadelphia 12, Pa.; *for mail,* 127 Midland Ave., Arlington, N.J.
Roome, George R. S. (J'44) (FKS), 138 South St., Freehold, N.J.
Roome, Hugh Reinagle (J'41) (SMCBHD), 4905 Rodman St., N.W., Washington 16, D.C.
Rooney, Sidney Crawford (J'42) (WMNL), Engr., Pulp Mill Design, Pulp Div., Bloedel, Stewart & Welch Ltd., 16 E. Hastings St., Vancouver; *for mail,* 532 E. 11th St., North Vancouver, B.C., Can.
Roop, Frank Sidney, Jr. (J'35) (SFERAK), Asst. Prof. Mech. Engrg., Va. Poly. Inst.; *for mail,* Box 459, Blacksburg, Va.
Roose, Robert W. (J'40) (ACJ), Maj., 410-7 Kearney, Ft. Leavenworth, Kan.
Roosman, Robert Q. (J'39) (ABM), Design Engr., Lockheed Aircraft Serv., Div. of Lockheed Aircraft Corp., 1705 Victory Pl., Burbank; *for mail,* 3223 Perlita Ave., Los Angeles 26, Calif.
Rooste, Ernest Edward ('26; '35) (JAYC), c/o Australian Gen. Elec. Pty. Ltd., 93-95 Clarence St., Sydney, N.S.W., Australia.
Root, Albert B. ('41) (R), V.P., Hunt-Spiller Mfg. Corp., 383 Dorchester Ave., South Boston 27, Mass.
Root, Charles Stanley (A'45) (JNY), Developing Engr., Charles S. Root Mfg. Co., 92 Dimmick St.; *for mail,* 53 Cloran St., Springfield 9, Mass.
Root, Ernest L. ('29), New England Mgr., Permutit Co., 700 Statler Bldg., Boston; *for mail,* 61 Whittier Rd., Wellesley Hills, Mass.
Root, Frederick J. ('18; '35) (CMGJY), 223 Kenilworth Rd., Ridgewood, N.J.
Root, Frederick Victor (J'37) (CMRJ), Time Study Engr., Stand. Stoker Co., Inc., 1701 Gaskell Ave., Erie, Pa.
Root, Joseph James, Jr. ('14) (RCMDSO), V.P., Mech. Dept., Union Tank Car Co., 228 N. La Salle St., Chicago 1, Ill.
Root, Marshall Jewell ('40) (NBYSEF), Ch. Engr., Am. Wringer Co., Inc., 212 Clinton St.; *for mail,* 17 Cold Spring Pl., Woonsocket, R.I.
Root, Virgil Adelbert ('43) (CHMJRE), Mgr., Mch. Tool Dept., China Corp., 1319 F St., N.W., Washington 4; *for mail,* c/o Wardman Park Hotel, Washington 8, D.C.
Rootes, Harold Alfred (J'42) (JMHY), Lab. Engr., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3; *for mail,* 13912 Potomac Ave., Cleveland 12, Ohio.
Roper, Charles Garland ('46) (Z), Elec. Engr., Manning, Maxwell & Moore, Inc., Bridgeport; *for mail,* 36 Green Acre Lane, Fairfield, Conn.
Roper, Edward H. (J'37) (CJW), Asst. to Mgr., Tech. Sales Div., Air Reduction Sales Co., 60 E. 42nd St., New York 17, N.Y.
Rorick, Joseph Arthur (J'43), Jr. Engr., Draftsman Designer, Eastman Kodak Co., Kodak Park; *for mail,* 194 Magee Ave., Rochester 13, N.Y.
Rosanowski, Walter Peter (J'41) (ASHJ), Draftsman & Designer, Kohler System Co., 159 E. Chicago Ave., Chicago; *for mail,* 245 S. 13th Ave., Maywood, Ill.
Rosborough, Caldwell Robert ('13; '21) (CBA LYS), Pres., Gen. Mgr., Moline Tool Co., 102—20th St.; *for mail,* 1235—11th Ave., Moline, Ill.
Rosborough, James Graham, Jr. ('38) (CDK LS), Acting Ch. Engr., Anheuser-Busch, Inc., St. Louis 18, Mo.
Roscher, Alfred M. (J'33) (BNTC), Engr., Design, Welin Davit & Boat Corp., 500 Market St., Perth Amboy; *for mail,* 115 Mt. Prospect Ave., Belleville 9, N.J.
Roscoe, Harry W. ('37) (CDL), Mech. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 5 Columbus Ave., Montclair, N.J.
Rose, Andrew H., Jr. (J'36) (KHCLNY), Cons. Engr., H. B. Berry Labs., 5122 Alhambra Blvd., Los Angeles 32; *for mail,* 1123 S. Termaine Ave., Los Angeles 6, Calif.
Rose, Bennie Albert ('40) (DJRO), Dir. of Engrg., Heyl & Patterson, Inc., 50 Water St., Pittsburgh; *for mail,* 603 Woodside Rd., Pittsburgh 21, Pa.
Rose, Carl G. (J'31) (JBZYKN), Research Sec. Head, Carborundum Co., Buffalo Ave., Niagara Falls, N.Y.
Rose, Fred W. ('13; '17) (SAKEFY), 2151 Carroll Ave., St. Paul 4, Minn.
Rose, Harold James ('45) (FKR), V.P., Dir. of Research, Bituminous Coal Research, Inc., 719 Oliver Bldg., Pittsburgh 22, Pa.
Rose, John H. (A'07), Retired; 33 Chautauqua Pl., Bradford, Pa.
Rose, Lawrence Milton (J'43) (ASH), Mech. Engr., Gen. Elec. Co., Schenectady, N.Y.; *for mail,* 12543 Lake Ave., Lakewood 7, Ohio.
Rose, Leonard Julian ('24; '35) (NJMSEO), Mech. Engr., Chem. Warfare Serv., War Dept., Washington 25; *for mail,* 3011 Rodman St., N.W., Washington, D.C.
Rose, Reed A. ('21; '30) (AEF), Lt. Comdr., U.S.N., U.S.S. *Kilauea,* c/o Postmaster, New York, N.Y.
Rose, Robert Lawrence (J'43) (KHSB), Instr., Mech. Engrg. Dept., Ill. Inst. of Tech., 3300 S. Federal St., Chicago 16; *for mail,* 726 Erie St., Oak Park, Ill.
Rosebrough, John D. (J'42), c/o Am. Blower Co., 6000 Russell St., Detroit, Mich.
Rosebrugh, Crawford Monroe ('29) (EHCSOB), Ch. Engr., Houston Pipe Line Div., Gulf Oil Corp., P.O. Drawer 2100, Houston 1; *for mail,* 2036 Harvard St., Houston 8, Tex.
Rosecky, George A. (J'39), 2853 N. 41st St., Milwaukee, Wis.
Roseman, Theodore (J'45), 1607 Ruscomb St., Philadelphia, Pa.
Rosemont, Leland W. (J'45) (BDN), Sales & Serv. Engr., Ahlberg Bearing Co., 3025 W. 47th St., Chicago 32, Ill.
Rosen, Albert (J'44) (BCD), Project Engr., Frankford Arsenal, Bridesburg, Philadelphia; *for mail,* 1727 N. 57th St., Philadelphia 31, Pa.
Rosen, Carl G. A. (16; '21; '35) (BEK), Dir. of Research, Caterpillar Tractor Co., Peoria 8, Ill.
Rosen, Saul (J'46) (CEM), 2002 Ave. J, Brooklyn 10, N.Y.
Rosenbaum, Arthur (J'41) (ZMCS), Radio Technician, 2nd Class, U.S.N., U.S.S. *Buckingham* APA-141, F.P.O., New York, N.Y.; *for mail,* 2301 N. Stanton St., El Paso, Tex.
Rosenbaum, Arthur G. ('43) (SFCK), Supt. Steam Prod., Louisville Gas & Elec. Co., 311 W. Chestnut St., Louisville 2, Ky.
Rosenberg, Edgar N. (J'45) (AEM), Mech. Engr., Charge of Mech. Design Sec., Electronics Lab., U.S.N., Point Loma; *for mail,* 3725 Leland St., San Diego, Calif.
Rosenberg, Edwin Charles ('25; '35) (BMDG CN), Exec. V.P., Ch. Engr., Charge of Engrg. & Mfg., F. Rosenberg Elev. Co., 3745 N. Richards St.; *for mail,* 3356 N. Humboldt Ave., Milwaukee 12, Wis.
Rosenberg, Heyman ('27), V.P., Parker-Kalon Corp., 200 Varick St., New York 14, N.Y.
Rosenberg, Irving Sumner (J'40) (ANOS), Mgr., Perry Elec. Co., 20 Congress Ave.; *for mail,* 65 Cottage St., Chelsea 50, Mass.
Rosenberg, Joseph Harry (J'43) (MZYNCA), Mech. Engr., Rosenberg Bros., 625 W. 55th St., New York 19; *for mail,* 348 Beach 146th St., Rockaway Beach, N. Y.
Rosenberg, S. ('20), 8440 Broadway, New York, N.Y.
Rosenberger, Albert John ('45) (ZBSLFN), Ch. Design Engr., Republic Flow Meters Co., 2240 Diversey Ave., Chicago 47; *for mail,* 807 Chestnut Ave., Wilmette, Ill.
Rosener, Leland S. ('27) (CDE), Cons. Engr., 233 Sansome St., San Francisco, Calif.
Rosenfeld, Maurice Sidney (J'40) (ABJ), Aero. Engr., Aircraft Matl. Lab., Naval Air Matl. Center, Naval Shipyard, Philadelphia 12; *for mail,* 6608 N. 12th St., Philadelphia 26, Pa.
Rosengarten, George Mott ('46) (EJK), Area Supvr., Carbide & Carbon Chemicals Corp., South Charleston, W.Va.
Rosenkrans, Joseph Robert (J'41 (NBJY), Ensign, U.S.N.R., Asst. Engrg. Officer, U.S.S. *Grafton,* APA-109, F.P.O., San Francisco, Calif.; *for mail,* Wray, Colo.
Rosenthal, Paul (J'43) (MCJBZ), Supvr., Quality Control Dept., Cent. Scien. Co., 1700 Irving Park Rd.; *for mail,* 739 Irving Park Rd., Chicago 13, Ill.
Rosenzweig, Siegfried ('14) (BEY), Pres., Treas., Korfund Co., Inc., 48-15—32nd Pl., Long Island City 1, N.Y.
Roshong, Raymond George ('40) (HBOCAN), Mgr., Crane Packing Co., 511 W. Pico St., Los Angeles 15; *for mail,* 4855 Gentry Ave., North Hollywood, Calif.
Rosing, William H. V. ('96); Retired; St. Louis, Mo.
Roslund, Arthur Elford (J'43) (MAZCJ), Prod. Engr., Carl L. Norden, Inc., 80 Lafayette St., New York 13; *for mail,* 141-19—78th Rd., Flushing, N.Y.
Rosness, George Edward (J'43) (ACE), Lt. (j.g.), U.S.N.R., Ground Trng. Sqd. 3, Naval Auxiliary Air Sta., Corry Field, Pensacola, Fla.
Ross, Alvin Morton (J'45), Apprentice Seaman, U.S.N., 40 Gifford Ave., Jersey City, N.J.
Ross, Carroll Adams ('21; '26; '39) (OMN), Cons. Engr., 113 Frontenac Ave., Buffalo 16, N.Y.
Ross, Chandler C. (J'37) (CHM), 817 N. Madison St., Pasadena, Calif.
Ross, Cleland Coldwell ('15; '22; '29) (NHBS CM), Propr., Ross Engrs., 82 St. Paul St., Rochester 4; *for mail,* 121 Penfield Rd., Rochester 10, N.Y.
Ross, Don R. (J'40) (DBNFLS), Owner, Mgr., Firm of Don R. Ross, Sand & Gravel, Colfax, Wash.
Ross, Edward Robert ('42) (EKS), Mech. Engr., Bevington-Williams, Inc., 730 K. of P. Bldg., Indianapolis, Ind.
Ross, Frank ('38), Supt. M. P., Elec. Engr., Terminal Railroad Association, Rm. 413, Union Sta.; *for mail,* 5621 Lisette, St. Louis, Mo.
Ross, Gilbert I. ('44), Consultant Engr., Ross & Co., 441 Lexington Ave., New York 17, N.Y.
Ross, Grant Duncan ('43) (HLS), Ch. Engr., Marathon Paper Mills of Can., Ltd., 320 Bay St., Toronto 1, Ont., Can.
Ross, Harold J. M. ('26; '34; '35) (AHS), Chief, Turbine Inspc. Div., Gilbert Associates, Inc., 412 Washington St.; *for mail,* 1349 Mineral Spring Rd., Reading, Pa.
Ross, Hugh Landers ('29; '46) (HSJN), Ch. Engr., Pump Dept., Allis-Chalmers Mfg. Co., Milwaukee; *for mail,* 2303 E. Olive St., Milwaukee 11, Wis.
Ross, James Harvey ('41) (CDLMOS), Engrg. Consultant, 2906 Robinson Ave., Austin 22, Tex.
Ross, John A., Jr. ('16) (ACE), Pres., Clarkson College of Tech., 51-63 Main St.; *for mail,* 71 Pierrepont Ave., Potsdam, N.Y.
Ross, John Ogg ('20), Pres., Ross Industries Corp., 350 Madison Ave., New York 17, N.Y.
Ross, Otto Carl, Jr. (J'40) (SEBKHZ), Asst. Engr., Stand. Oil Co. of Calif., El Segundo; *for mail,* 217—30th St., Manhattan Beach, Calif.
Ross, Richard Holbert (J'44) (SFE), Terminal Supt., Tex. Petroleum Co., Apartado 249, Puerto Limon, Costa Rica, C.A.
Ross, Robert Cleland (J'41) (CM), Project Engr., Eastman Kodak Co., Kodak Park, Lake Ave., Rochester; *for mail,* 121 Penfield Rd., Rochester 10, N.Y.
Ross, Robert Simon (J'42) (ABNHZY), Ch. Wind Tunnel Research Engr., Daniel Guggenheim Airship Inst., 1300 Triplett Blvd., Akron 6; *for mail,* 3228 E. Overlook Rd., Cleveland Heights 18, Ohio.
Ross, Thurston H. ('26) (CDM), Capt., U.S.N.R., c/o Dr. T. H. Ross, Jr., Veteran's Hospital, Walla Walla, Wash.
Ross, William Clifford (J'46), 48 Warder St., Dayton 5, Ohio.
Ross, William Davis (J'45) (CM), Admin. Aide, Project 22, Northwest. Tech. Inst.; *for mail,* 1922 Orrington St., Evanston, Ill.
Ross, William Tyler (J'43) (C), Asst. Trainmaster, Norfolk & West. Ry. Co., Roanoke; *for mail,* 712 Mt. Vernon Rd., Roanoke 15, Va.
Rossetto, Louis (J'36) (BMS), Lt. Col., Ord. Dept., Army Serv. Forces, Frankford Arsenal, Philadelphia 37, Pa.
Rossheim, David B. ('35) (ABEFJK), Ch. Engr., M. W. Kellogg Co., Foot of Danforth Ave., Jersey City 3; *for mail,* 229 Van Buren Ave., Teaneck, N.J.
Rossi, Boniface Ernest (J'41) (AOJBCM), Matls. & Processes Engr., Columbia Aircraft Corp., Sunrise Highway, Valley Stream; *for mail,* 36 Irving Pl., Baldwin, L.I., N.Y.
Rossmann, Peter Franz ('43) (AMR), Gen. Mgr., Devel. Div., Curtiss-Wright Corp.; *for mail,* Box 30, Bloomfield, N.J.
Rossner, Bertram J. (J'43) (CMKOS), Lt. (j.g.), U.S.N.R., Engrg. & Salvage Officer, U.S.S. *Laysan Island,* F.P.O., San Francisco, Calif.; *for mail,* 610 W. 110th St., New York 25, N.Y.
Roswell, Irving (J'46) (EHJSM), Lt., U.S.N.R., 2013 Fair Oaks Ave., South Pasadena, Calif.
Roth, Arthur Jacob (J'44) (NJBMYC), Mech. Engr., Submarine Sig. Co., 160 N. Washington St., Boston; *for mail,* 12 Sumner Rd., Cambridge, Mass.
Roth, Henry ('24; '35) (NHJBZ), Design Engr., Hamilton Watch Co., Columbia Ave.; *for mail,* 721 Columbia Ave., Lancaster, Pa.
Roth, Herbert ('29; '35) (A), Drawer B, Hartsdale, N.Y.
Roth, Paul V. ('21; '38), Shop Engr., Leeds & Northrup Co., 4901 Stenton Ave.; *for mail,* 4931 N. 16th St., Philadelphia, Pa.
Roth, Robert Julius (J'44) (LCMZAY), Research Engr., Bur. of Visual Sci., Am. Optical Co., Mechanic St., Southbridge, Mass.
Roth, W. E. ('42) (EFS), 45 Herman St., Hackensack, N.J.
Rothe, Hugo Frederick ('36) (L), Mech. Design Engr., Wyandotte Chemicals Corp., Wyandotte; *for mail,* 1069 Emmons Blvd., Lincoln Park 25, Mich.
Rothemich, Edmund Fredrick (J'41) (SFAZ), Field Serv. Engr., Riley Stoker Corp., 9 Neponset St.; *for mail,* 11 Intervale Rd., Worcester, Mass.
Rothenberg, George S. (J'42), Philco Corp., Tioga & C Sts., Philadelphia 34, Pa.
Rothermel, Earl Harold (J'44), 6532 Malabar St., Huntington Park, Calif.
Rothgeb, Ross McKinley ('28; '31) (CFSOBK), Gen. Mgr., N.C. Hospitals Bd. of Control, 509 Revenue Bldg., Raleigh, N.C.
Rothmaler, Oswald ('26) (CLME), Secy., Ertel Engrg. Corp., 40 W. 48th St., New York 19; *for mail,* 197 Rugby Rd., Brooklyn 26, N.Y.
Rothman, Nathan Matthew ('44) (NCYAWT), Ord. Engr., Design & Supvg., N.Y. Naval Shipyard, Brooklyn; *for mail,* 145 Lincoln Rd., Brooklyn 25, N.Y.
Rotondo, Alfred (J'38) (CDFMST), Draftsman, Engrg. Dept., Am. Woolen Co., Inc., 1 Mill St., Lawrence, Mass.
Rottersman, Henry (J'45) (ABKN), Designer, Sam Tour Engrg. Co., 44 Trinity Pl., New York, N.Y.; *for mail,* 726 E. 22nd St., Paterson 4, N.J.

Rottersmann, Maurice Henry (J'40) (KBNESZ), Ch. Engr., Advance Oven Co., 700 S. 18th St., St. Louis; *for mail*, 437 Woodlawn Ave., Webster Groves, Mo.

Roubik, Joseph Rudolph (J'45) (MNBJAH), Jr. Mch. Designer, Research & Devel., Kearney & Trecker Corp., 6784 W. National Ave., Milwaukee 14; *for mail*, 1606 S. 33rd St., Milwaukee 4, Wis.

Rouch, Ernest A. ('24; '34; '35) (BHM), Checker, Engrg. Dept., Cameron Pump Div., Ingersoll-Rand Co., Phillipsburg, N.J.; *for mail*, R.D. 3, Bangor, Pa.

Roudebush, R. E. ('36) (CDWYA), Assoc. Prof. Mech. Engrg., Iowa State College, Ames, Iowa.

Rougemont, Rene Louis ('44) (NMCY), Ch. Engr., Reed-Prentice Corp., Worcester 4; *for mail*, 6 Baxter St., Worcester 2, Mass.

Roulton, John A. ('39) (SFLE), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.; *for mail*, 85 Washington St., East Orange, N.J.

Rouse, Daniel Henry, Jr. (J'39), 2449 Broadway, Quincy, Ill.

Rouse, Robert Wilson (J'43), 3426 Brown St., N.W., Washington 10, D.C.

Roushey, Ray Ezra (J'44) (SLOCDF), Ch. Engr., Harrington & Co., Dushore, Pa.

Rousku, Aaro William (J'44) (SKCLNA), 816—42rd St., Brooklyn 32, N.Y.

Rouy, Auguste Louis Marie Antoine ('44) (YKBAZS), Cons. Engr., Lord Mfg. Co., Erie, Pa.; *for mail*, 95 Christopher St., New York 14, N.Y.

Row, Ronald Boswell (J'42) (CAMJDO), Outside Prod. Buyer, Boeing Aircraft Co., Georgetown Sta., Seattle; *for mail*, 3354—59th, S.W., Seattle 6, Wash.

Rowan, Adrian McMore (J'42) (ESM), Ensign, U.S.N.R., Naval Shipyard, Portsmouth, N.H.; *for mail*, 89 Poultney St., Whitehall, N.Y.

Rowan, Edward D. (J'41) (JEFH), *Student Award*, '40; Project Engr., Portland Gas & Coke Co., Pub. Serv. Bldg., Portland; *for mail*, 3602 S.E. Stark St., Portland 15, Ore.

Rowan, Robert Bruce ('38; '44) (FHK), Mech. Engr., U.S. Bonneville Power Admin.; *for mail*, 7604 S.E. 18th Ave., Portland, Ore.

Rowan, Robert L. ('26; '35) (FKS), Pres., Houston Interlocking Tile Co., 6006 Ardmore St.; *for mail*, 6024 Fordham St., Houston, Tex.

Rowand, Will Haines ('36; '43) (SFEBKC), Ch. Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.

Rowbotham, Charles Fisher (J'43) (MCJBL), Ensign, U.S.N.R.; 7704 Cregier Ave., Chicago 49, Ill.

Rowe, Gilbert Thornton (J'45) (KA), Designer, Air Conditioning, Douglas Aircraft Co., Inc.; *for mail*, 2105 Idaho Ave., Santa Monica, Calif.

Rowe, Harrison Gerard (J'44) (CMDJBS), Lt. (j.g.), U.S.N.R., Hull Inspr., c/o Asst. to Indus. Mgr., Foot of Market St., San Francisco; *for mail*, 2321 LeConte, Berkeley 4, Calif.

Rowe, Hartley ('16) (ABDORS), V.P., United Fruit Co., 1 Federal St., Boston, Mass.

Rowe, Henry Atkinson (J'44) (ABC), Res. Engr., Bldg. 32, Mass. Inst. of Tech.; *for mail*, 5 Crawford St., Cambridge 39, Mass.

Rowe, William Henry (J'42) (MLSJFC), Admin. Asst. to Asst. Dir. of Operas., Gen. Chem. Co., 40 Rector St., New York 6, N.Y.

Rowell, Edward Sargent (J'44) (RSFKOC), Contract Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.

Rowell, J. Kirk (J'37) (CNMDBE), Pur. Agt., Jeffersonville Boat & Mch. Co., Jeffersonville, Ind.; *for mail*, 122 Sherrin Ave., Louisville, Ky.

Rowell, John William (J'38) (ECFJ), Sales Engr., Cummins Eng. Co.; *for mail*, 624½ Franklin St., Columbus, Ind.

Rowell, Kendall Bailey ('30; '35) (REAC), Engr., Diesel Loco. Design, Am. Loco. Co., Schenectady 5, N.Y.

Rowell, Lester James ('43) (SFJYZG), Effic. Engr., Chicago Dist. Elec. Generating Corp., Box 65, Hammond, Ind.; *for mail*, 7850 S. Shore Dr., Chicago 49, Ill.

Rowell, Richard M. (J'38), Ensign, U.S.N.R., Fighting Sqd. 3, Naval Air Sta., San Diego, Calif.

Rowland, David J. ('23) (S), Owner, Sales Engr., Rowland & Burns, 39 Cortlandt St., New York, N.Y.

Rowland, Richard H. ('33) (CMY), V.P., Gen. Mgr., Gould Commercial Div., Natl. Battery Co., 35 Neoga St., Depew, N.Y.

Rowland, Robert A., Jr. (J'41) (CMO), Pres., Associated Builders, Inc.; *for mail*, 602 Hackberry Ave., McAllen, Tex.

Rowland, Roger W. ('25; '28), Pres., New Castle Refractories Co., Box 471, New Castle, Pa.

Rowland, William G. (J'46), Instr., Tex. Agric. & Min. College; *for mail*, Box 5657, College Station, Tex.

Rowland, William Platt (J'44), Lt., U.S.N.R., U.S.S. *War Hawk*, AP-168, F.P.O., San Francisco, Calif.; *for mail*, 291 Lincoln St., New Britain, Conn.

Rowley, Alfred Williams (J'45) (DJK), Kelley Springfield Tire Co.; *for mail*, Allegheny Grove, R.D. 1, Cumberland, Md.

Rowley, Frank B. ('11; '13; '18) (ASE), Prof., & Head, Mech. Engrg. Dept., Univ. of Minn., Minneapolis; *for mail*, Chaska Rd., Excelsior, Minn.

Rowley, Louis N., Jr. ('31; '42) (ESF), Managing Editor, *Power*, McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.

Rowley, Millard Corey (J'24) (ABN), Design Engr., Hamilton Stand. Propellers Div., United Aircraft Corp., Main St., East Hartford 8; *for mail*, 2 Gerard St., Manchester, Conn.

Rowley, Ridgway Lloyd ('10; '16; '22) (HYEW), Dir., Charge of Engrg. & Fire Ins. Depts., Johnson & Higgins of Calif., 311 California St., San Francisco 4; *for mail*, 30 La Noria, Route 2, Orinda, Calif.

Rowley, Robert Donald (J'41) (CMDAJ), Lt. Comdr., U.S.N.R., Operas. Officer, Navy 91, Box 2, F.P.O., San Francisco, Calif.; *for mail*, 7721 Ridge Blvd., Brooklyn 9, N.Y.

Rowley, William Rogers (J'46), Ensign, U.S.N.; 62 Beech St., East Orange, N.J.

Rowse, William Crapo ('12; '16) (SFK), Engr. Steam Design, Dept. of Water & Power, City of Los Angeles, 207 S. Broadway, Los Angeles 12; *for mail*, 1880 Hill Dr., Los Angeles 41, Calif.

Roy, Nereus Hubert ('41) (RBSZY), Dir. of Research, Waugh Equip. Co., 420 Lexington Ave., New York 17, N.Y.

Roy, Robert Hall ('41) (CG), Dir., Maint. Div., Waverly Press, Inc., Mt. Royal & Guilford Aves., Baltimore 2; *for mail*, Ruxton, Baltimore 4, Md.

Royce, Robert Fredrick (J'45), 328 E. Ann St., Ann Arbor, Mich.

Royer, Dan Long ('29; '35) (SFKC), Ch. Engr., Mgr., Engrg. Dept., Ocean Accident & Guarantee Corp., 1 Park Ave., New York 16, N.Y.

Royle, Vernon E. ('05; '17) (GJNWY), Retired; 399—15th Ave., Paterson 4, N.J.

Roys, Francis William ('21) (BJMA), Dean of Engrg., Head, Dept. of Mech. Engrg., Worcester Poly. Inst., Boynton St., Worcester 2, Mass.

Roys, Lawrence ('07; '21) Pres., West. Struc. Co.; *for mail*, c/o Moline Pub. Library, Moline, Ill.

Roysdon, Joe Page (J'42) (BCM), Mech. Engr., P-4, The Engr. Bd., Ft. Belvoir; *for mail*, 811—14th St., New Alexandria, Va.

Rozenberg, Henry W. ('26; '35), Ch. Engr., J. K. Mosser Leather Corp., 500 Arch St., Williamsport, Pa.

Rozett, William, Jr. ('33; '45) (YPC), Mgr., Tech. Products Dept., Shell Oil Co., Inc., 441 Stuart St., Boston 16; *for mail*, 11 Valley Rd., Winchester, Mass.

Rubel, Paul (J'45) (BHS), 3131 W. Highland Ave., Milwaukee, Wis.

Rubendunst, Robert Frank (J'39) (BMNCJ), Tool Engr., Knowlton Engrg. Co., 2524 Leslie Ave., Norwood 12; *for mail*, 2977 Lischer Ave., Cincinnati 11, Ohio.

Rubenkoenig, Harry ('17; '35) (ERS), Prof. Ry. Mech. Engrg., Purdue Univ., Lafayette; *for mail*, 1120 Northwestern Ave., West Lafayette, Ind.

Rubin, Jacob Carl (J'45) (NMBJKC), Mech. Designer, U.S. Television Mfg. Corp., 106—7th Ave., New York 11; *for mail*, 1248 St. Marks Ave., Brooklyn 13, N.Y.

Rubin, Maurice L. (J'38) (BKS), Mech. Engr., Cent. Tech. Dept., Bethlehem Steel Co., Quincy; *for mail*, 76 W. Cedar St., Boston 14, Mass.

Rubinoff, Abraham Lipman (J'37) (MCANSB), Instr. in Engrg. Drawing, Ajax Div., Univ. of Toronto; *for mail*, Arbor Lodge, Ajax, Ont., Can.

Rubinstein, Martin Alexander (J'45) (AKY), Jr. Layout Engr., Chance Vought Aircraft Div., United Aircraft Corp., Stratford; *for mail*, 144 Washington Ave., Bridgeport, Conn.

Ruch, Alan Joseph (J'27) (SFK), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail*, 80 Beechwood St., Port Washington, N.Y.

Ruck, George ('15; '35) (CMJD), Pres., Gen. Mgr., Columbia Steel Equip. Co., Lincoln-Liberty Bldg., Philadelphia 7; *for mail*, Huntington Valley, Montgomery Co., Pa.

Ruck, Thomas Frederick (J'45) (CMKNDS), Mech. Engr., Corning Glass Works; *for mail*, 52 E. 3rd St., Corning, N.Y.

Ruckman, John H. ('21; '35), Lt. Col., Ord. Dept. U.S.A., Exec. Officer, Seneca Ord. Depot, Romulus, N.Y.

Rud, George Franklin (J'40) (SFED), Draftsman, Froehlich & Emery Engrg. Co., 510 Toledo Trust Bldg., Toledo 4; *for mail*, 2424 Rosewood Ave., Toledo 6, Ohio.

Rudbarg, Frank (J'42) (NTSL), Engrg. Draftsman, Am. Viscose Corp., 35 S. 9th St., Philadelphia, Pa.; *for mail*, 2105 Jefferson St., Wilmington 258, Del.

Rudd, William C. ('23) (CFHS), Mech. Engr., City Engr.'s Office, 1000 Griswold Bldg., Detroit 26, Mich.

Ruddick, Robert Bell (J'42) (JBLY), Research Engr., Eastern Lab., E. I. du Pont de Nemours & Co., Gibbstown; *for mail*, 367 Morris St., Woodbury, N.J.

Rude, Robert Lee ('34) (CPS), Mfg. Dept., Sun Oil Co., 1608 Walnut St., Philadelphia 3, Pa.

Ruder, Winfield Scott (J'41) (SCFDZK), Gen. Delivery, Lexington, N.Y.

Rudgers, Anthony J. (J'29) (BEJMNA), Chief, Fed. Specifications Div., Procurement Div., U.S. Treasury Dept., 7th & D Sts., S.W., Washington 25, D.C.; *for mail*, 304 S. Ivy St., Arlington, Va.

Rudick, Milton Joel (J'44), Lt., U.S.A., Ord. Dept., A.P.O. 11983, c/o Postmaster, New York, N.Y.; *for mail*, 274 Otis St., West Newton 65, Mass.

Rudiger, Bernhard W. (J'37) (JEFM), Mech. Engr., Irvington Smelting & Refining Works, Nye Ave., Irvington; *for mail*, 22 Sheridan Ave., West Orange, N.J.

Rudin, William ('27) (SEB), Div. Engr., Charge of Struc. Engrg., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail*, 705 Charles St., Bellmore, L.I., N.Y.

Rudolph, Frederic Charles ('37) (OLBKH), Cons. Engr., 1841 Broadway, New York 23, N.Y.

Rue, Harold E. ('23; '27) (LMBC), Ch. Engr., Pabst Brewing Co., 917 W. Juneau Ave., Milwaukee 1; *for mail*, 7914 Warren Ave., Wauwatosa 13, Wis.

Rue, John Davison ('30) (LOKFZW), Asst. to Ch. Engr., Hooker Electrochem. Co., 4701 Buffalo Ave.; *for mail*, 3061 Dorchester Rd., Niagara Falls, N.Y.

Rued, Fred H. (J'35) (BHM), Jr. Engr., Pelton Water Wheel Co., 2929—19th St., San Francisco, Calif.

Ruemelin, Richard ('20) (K), Pres., Ch. Mech. Engr., Ruemelin Mfg. Co., 3860 N. Palmer St., Milwaukee 12; *for mail*, 4939 N. Cumberland Blvd., Milwaukee 11, Wis.

Ruess, Max Emil ('36; '43) (NSEK), Designer, Nordberg Mfg. Co., 3073 S. Chase, Milwaukee 7, Wis.

Ruetschi, Robert R. (J'36) (HKL), U.S. Navy, 332 E. Main St., Greenfield, Ind.

Ruettinger, Thurman Otto (J'39) (BHNAZ), Test Engr., Research Lab., Bendix Aviation Corp., 4855—4th Ave., Detroit 1, Mich.

Ruff, Alonzo William ('44) (BCEFJN), V.P., Charge of Engrg., York-Shipley, Inc., Jessop Pl. & Pa. R.R.; *for mail*, 538 W. Spring Ave., York, Pa.

Ruff, Herbert ('20; '35) (KJF), Supt. Maint. & Repair, Mar. Dept., Atlantic Refining Co., Box 7709, Philadelphia; *for mail*, 207 Owen Ave., Lansdowne, Pa.

Rugge, Geo. J. ('40), Moody Engrg. Co., 140 Cedar St., New York, N.Y.

Ruggles, Robert Raymond (J'45) (CMABLD), Aero. Research Engr., Daniel Guggenheim Airship Inst., 1300 Triplett Blvd., Akron; *for mail*, 8320 Bellevue Ave., Cleveland 3, Ohio.

Rugh, John M. ('33), Mech. Engr., Geneva Mill, Wanskuck Co., 1117 Douglas Ave., Providence, R.I.

Ruhloff, F. Carl ('25), Sales Engr., Bucyrus-Erie Co.; *for mail*, 1193 Fairview Ave., South Milwaukee, Wis.

Ruiter, Ray N. ('42) (FDSWZM), Engr., Rust Engrg. Co., 2109—5th Ave. N., Birmingham 5, Ala.

Ruiz, Albert Louis ('30; '41) (BAR), Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.

Ruiz, Joseph J. ('40) (BDFHKL), Hotel Commodore, Euclid & Mayfield, Cleveland 6, Ohio.

Rule, Perrin ('18), Gen. Supt., Youngstown Sheet & Tube Co., 9355 S. Kreiter Ave., Chicago 17, Ill.

Ruley, Bryce Townsend (J'44) (BCNAOZ), Stress Analyst, SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34; *for mail*, 918 The Kenilworth, Alden Park, Philadelphia 44, Pa.

Rulfs, Carl H. ('29) (SFEKBH), Design Engr., Union Elec. Co. of Mo., 315 N. 12th Blvd., St. Louis 1, Mo.

Rumball, Caswell John Hanna (J'45) (CHO), 1 Marjory Ave., Toronto, Ont., Can.

Rumble, Virgil Alden ('25; '34; '35) (EFHKSZ), Mgr., San Francisco Office, Bailey Meter Co., 424 Sharon Bldg., San Francisco 5, Calif.

Rumfelt, Henry F. C. (J'38) (BHM), Capt., 1862 Guilford Rd., Columbus, Ohio.

Rumpf, H. E. (J'39) (LKFH), Plant Engr., Red Star Yeast & Products Co., 325 N. 27th St., Milwaukee 8; *for mail*, 8330 Avon Court, Wauwatosa 13, Wis.

Rumpp, Emile T., Jr. (J'38) (BEM), Engr., Scripps Motor Co., 5817 Lincoln St., Detroit; *for mail*, 2435 Burns Ave., Detroit 14, Mich.

Rumsey, Clayton Selah (J'40) (RESABK), Loco. Engrg. Div., Gen. Elec. Co., E. Lake Rd.; *for mail*, 668 Silliman Ave., Lawrence Park, Erie, Pa.

Rundell, Harwood Faville ('42) (AHMY), Sales Engr., Aeroquip Corp., 300 S. East Ave.; *for mail*, 604 McBride St., Jackson, Mich.

Rundlett, Donald Carlisle (J'45) (DHS), 8 Shepard St., Cambridge, Mass.

Runquist, Ernest Martin (J'41) (ACDEHO), Devel. Engr., Consltd. Vultee Aircraft Corp., Lindbergh Field, San Diego; *for mail*, 5128 Foothill Blvd., San Diego 9, Calif.

Runyon, Malcolm Eagles ('27; '33; '35) (OFHSEK), Jr. Partner, Runyon & Carey, Cons. Engrs., 31 Fulton St., Newark 2; *for mail*, 15 New England Ave., Summit, N.J.

Ruoff, Frederic L. ('39) (FKSCO), Mgr., Natl. Constr. Dept., Plibrico Jointless Firebrick Co., 1840 Kingsbury St., Chicago 14, Ill.

Rupinski, Waldemar (J'42) (FKS), Research Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail*, 4914 Tennessee, Chattanooga 9, Tenn.

Rupp, Manning E. ('09; '13; '19) (CMBAO), Plant Engr., East. Air Devices, Inc., 585 Dean St., Brooklyn 17; *for mail*, 710 Ave. V, Brooklyn 23, N.Y.

Rusanowski, Walter Peter (J'41) (ASHJ), 245 S. 13th Ave., Maywood, Ill.

Rush, Charles Daniel (J'42) (ABH), Box 907, Amagansett, N.Y.

Rush, John Edward (J'45) (NOAMGY), Mch. Designer, Armstrong Cork Co., Liberty & Mary Sts., Lancaster, Pa.; *for mail,* R.D. 1, Phillipsburg, N.J.
Rush, Ralph M. ('40) (CKS), Mgr., Indus. Dept., Dravo Corp., 300 Penn Ave., Pittsburgh, Pa.
Rushing, Frank Crofford ('40) (BCN), Mgr., Motor Engrg., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Rusie, Emil F. (J'42) (BDEF), Navy Dept., Rm. 1007, 983 Main St., Hartford, Conn.
Rusk, Glenn Orde ('45) (KOS), Mgr., Rusk Engrg. Associates, 816 W. 5th St., Los Angeles; *for mail,* Box 396, R.F.D. 1, Orinda, Calif.
Ruskin, Philip ('45) (EOP), Automotive Engr. & Writer, 28 E. 31st St., New York 16, N.Y.
Rusnack, Victor Arthur (J'40), Draftsman, Buhl Stamping Co., 2730 Scotten Ave., Detroit; *for mail,* 16545 Ward, Detroit 27, Mich.
Russ, Daniel Carver, Jr. (J'45), 53-12 Junction Blvd., Elmhurst, N.Y.
Russ, Daniel George (J'40) (ABS), Aero. Engr., Naval Air Exper. Sta., Naval Air Matl. Center, Naval Shipyard, Philadelphia 12, Pa.; *for mail,* 114 Edison Ave., Erlton, N.J.
Russ, John Megginson ('32) (GNMLC), Prof. Engrg. Drawing, State Univ. of Iowa, 208 Elec. Engrg. Bldg., Iowa City, Iowa.
Russell, Andrew ('44) (JKEB), Engrg. & Devel. Dept., Imperial Oil Ltd., S. Christina St.; *for mail,* 324 London Rd, Sarnia, Ont., Can.
Russell, Arthur O. ('41) (BJT), Sylvania Indus. Corp.; *for mail,* 300 Caroline St., Fredericksburg, Va.
Russell, Floyd L. ('23), Constr. Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark, N.J.
Russell, Frank Edwin ('24) (EHKNRS), Retired; 1750 Vallejo, San Francisco 23, Calif.
Russell, James R. (J'40) (AES), Asst. Engr., Office of Chief of Ord., War Dept., The Pentagon, Washington 25, D.C.; *for mail,* 1061 S. Edison St., Arlington, Va.
Russell, James Scotti (J'38) (AEFJS), Lt., U.S.N.R., Air Ord. Officer, U.S.S. *Hornet,* F.P.O., San Francisco, Calif.
Russell, John Holmes (J'43), Pvt., 1st Class, U.S.A., 818th Military Police Co.; *for mail,* 23 Gibson St., Needham 92, Mass.
Russell, John J. ('33) (BS), Mech. Engr., Courter & Co. Inc., 204 Lafayette St., New York 12, N.Y.; *for mail,* 1605 Lemoine Ave., Fort Lee, N.J.
Russell, Kenneth F. ('29; '40) (CBHJMZ), Gen. Mgr., Vortox Mfg. Co., 121 S. Alexander; *for mail,* 665 W. 10th St., Claremont, Calif.
Russell, R. M., Jr. (J'40) (BMS), Shelby County Sch. of Aero., Whitehaven, Tenn.
Russell, Robert A. (J'40) (BFS), Mech. Engr., Black & Veatch, 4706 Broadway, Kansas City; *for mail,* 2219 Fieldston Rd., Kansas City 3, Kan.
Russell, Robert Jackson ('28; '42) (LKF), Hardinge Co., Inc., York, Pa.
Russell, Samuel (J'37) (SZF), Asst. Mech. Engr., Engrg. & Maint. Dept., Eastman Kodak Co., Kodak Park; *for mail,* 261 Merrill St., Rochester, N.Y.
Russell, Thomas Frederick ('44) (CBMN), Partner, Highway Equip. Co., 2150 Langdon Farm Rd., Cincinnati 12, Ohio.
Russo, Gennaro (J'45), 1060 Stell Pl., Bronx, N.Y.
Russo, James N. (J'45), 63 Van Winkle St., Dorchester, Mass.
Rust, Albert D, III ('37) (SKLCFM), Asst. Ch. Engr., Dow Chem. Co.; *for mail,* Angleton, Tex.
Rust, George M. ('31; '41) (BCSZ), V.P., Rust Engrg. Co., Exchange Bldg., Birmingham 3, Ala.
Rust, Mack Donald ('27; '35) (JYDH), Rust Cotton Picker, Box G, Casa Grande, Ariz.
Rust, Stirling Murray, Jr. ('34; '43) (COLSWJ), Pres., Rust Engrg. Co., Clark Bldg., Pittsburgh 22, Pa.
Rutherglen, John Alfred ('39; '45) (RSEMB), Dist. Trans. Engr., Gen. Elec. Co., 920 S.W. 6th Ave., Portland; *for mail,* 2535 N.E. 30th Ave., Portland 12, Ore.
Rutishauser, Hans (J'40) (BEFA), Ch. Research Engr., Aerojet Engrg. Corp., Colorado St., Pasadena; *for mail,* 1795 Sonoma Dr., Altadena, Calif.
Rutkovsky, Hyman David (J'40) (JLMBS), Engr., Picatinny Arsenal; *for mail,* 51 Berry St., Dover, N.J.
Rutledge, Eric A. (J'37), Research Engr., Sales, Rensselaer Valve Co.; *for mail,* R.F.D. 2, Leversee Rd., Troy, N.Y.
Rutz, William Edward ('39) (CHMNY), V.P., Works Mgr., Giddings & Lewis Mch. Tool Co., 142 Doty St.; *for mail,* 290 Sheboygan St., Fond du Lac, Wis.
Ruzga, Joseph Albert (J'39) (MBCLYJ), Engr., Mech. Mfg., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore 24; *for mail,* 219 Alpine Rd., Baltimore 10, Md.
Ryan, Beverly Edward (J'32) (OSH), Capt., Adjutant, U.S.A., Hdq. 3rd Student Regt., Armored Sch., Ft. Knox, Ky.; *for mail,* 203 Midland Ave., Little Rock, Ark.
Ryan, Clifton Miles (J'45) (BHJ), Taylor Instrument Cos., Rochester; *for mail,* c/o Smith, Fenton Rd., Rochester 11, N.Y.
Ryan, David Gerald ('39) (MBK), Prof. Mech. Engrg., Univ. of Ill., 115 Trans. Bldg., Urbana, Ill.
Ryan, Edmund J. (J'40), 914 Argonne Dr., Baltimore, Md.
Ryan, Francis M. ('19; '25; '35) (ABL), Foreign Dept., Norton Co., Worcester, Mass.
Ryan, Frederick James, Jr. (J'41) (FELO), Sales Engr., Phillips Petroleum Co., 15th & Locust Sts., Philadelphia; *for mail,* Washington Lane & Noble Rd., Jenkintown, Pa.
Ryan, James Jay ('26; '42) (NBSAC), Assoc. Prof., Mech. Engrg. Dept., Univ. of Minn., Minneapolis 14, Minn.
Ryan, James W. (J'41) (JLMZYA), Ch. Metallurgist, Eclipse Mch. Div., Bendix Aviation Corp., Oakwood Ave. & 18th St.; *for mail,* 401 Maple Ave., Elmira, N.Y.
Ryan, John Edward ('34; '45) (BKNJZ), Research Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 2190 Grand Blvd., Schenectady 8, N.Y.
Ryan, John Michael (J'43) (RFSEK), Seaman, 1st Class, U.S.N., Rm. 121, Bldg. 601, Naval Trng. Center, Bainbridge, Md.
Ryan, John Thomas, Jr. ('44) (CFJ), Gen. Mgr., Mine Safety Appliances Co., 201 N. Braddock Ave., Pittsburgh 8, Pa.
Ryan, Robert Edwin (J'45) (JNGB), 101 McKinley Ave., Lansdowne, Pa.
Ryan, Warren W. (J'40) (ADE), Staff Sgt., 1601 E. 5th St., Dayton 3, Ohio.
Ryan, William Francis ('17; '24; F'38) (SBLFE), Asst. Engrg. Mgr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Ryan, William John ('12; '36) (SF), Engr., Water Serv. Labs., 423 W. 126th St., New York 27; *for mail,* 403 W. 115th St., New York 25, N.Y.
Ryan, William R. (J'34) (ACS), Ch. Engr., Marine Div., Bendix Aviation Corp., 106 Nostrand Ave., Brooklyn 5, N.Y.
Rybkin, I. Z. (J'33), Apt. 50, 4 Machinostroenia St., Moscow 88, U.S.S.R.
Ryder, Fred. W. ('39) (BDFKSZ), Sales Mgr., Ryder Equip. Co., 4165 Flora Blvd., St. Louis 10, Mo.
Ryder, George (J'42) (NYBM), Mech. Engr., Devel., Telautograph Corp., 16 W. 61st St., New York 23; *for mail,* 1693 E. 22nd St., Brooklyn 29, N.Y.
Ryder, William L. (J'39) (CM), Ensign, U.S.N.R., Design Unit, Bur. of Ord., Navy Dept., Calif. Inst. of Tech., Pasadena 4, Calif.
Ryding, Herbert Charles ('00), Retired; Rhodes Circle, Birmingham 5, Ala.
Rynda, Joseph Theodore, Jr. ('32) (ABJKOS), Elec. Engr., Power Heat, Commander-Larabee Milling Co., Boulevard Ave.; *for mail,* 202 S.W. Elm Ave., Montgomery, Minn.

S

Saalfrank, R. Bartlett ('46) (O), Cons. Engr., Box 37, Gulfport 7, Fla.
Saari, Thomas Arvo (J'41) (NYDMHA), Design Engr., Defiance Mch. Works, Inc., Perry & 3rd; *for mail,* 731 W. High, Defiance, Ohio.
Saathoff, George W. ('22) (SOFE), V.P., Elec. Advisers, Inc., 60 Wall Tower, New York 5, N.Y.
Sabersky, Rolf Heinrich (J'42) (HKABS), Devel. Engr., Aerojet Engrg. Corp., 332 Irwindale Rd., Azusa, Calif.
Sabol, Alexander (J'46), 305 Wayne Ave., Lansdowne, Pa.
Sachs, George ('43) (JBOA), Prof. Physical Metal., Case Sch. of Applied Sci., Cleveland 6, Ohio.
Sachs, Henry (J'37) (BCHMLK), Mgr., Abco Metalcraft, 1008 N. Highland, Los Angeles 38; *for mail,* 1320 N. Alta Vista Blvd., Los Angeles 46, Calif.
Sachs, Joseph ('11) (ACJ), Research, Engrg. Devel., Invention & Pat. Consultant, 1900 Albany Ave., West Hartford 5, Conn.
Sachs, Rudolph (J'43) (BHK), 601 W. 113th St., New York 25, N.Y.
Sack, Melvin ('37; '41) (KSLBJN), Mgr., Heat Exchanger Div., Henry Vogt Mch. Co., 10th & Ormsby Sts.; *for mail,* 2333 Gladstone, Louisville, Ky.
Sackett, Robert Lemuel ('15; F'36) (COEH), *Manager,* '32-'35, *Vice-President,* '35-'37; *Assistant to Secretary,* A.S.M.E.; *for mail,* 303 Lexington Ave., New York 16, N.Y.
Sacks, Martin Murray (J'45) (HYSFZM), Process Devel. Engr., Spec. Engr. Det., Oak Ridge, Tenn.
Sackson, Murray (J'42) (ABE), Brewster Aero. Corp., Long Island City; *for mail,* 190 Waverly Pl., New York, N.Y.
Saco, Felix, Jr. (J'40) (KESABY), Research & Devel. Engr., Maxim Silencer Co., 85 Homestead Ave.; *for mail,* 404 Farmington Ave., Hartford, Conn.
Sadler, Cornelius Robinson ('13) (KMS), 426—4th Ave., Patterson Heights, Beaver Falls, Pa.
Sadler, J. H. ('23; '35) (BS), Mech. Supt., Steam Div., Duke Power Co., Box 338, Mt. Holly, N.C.
Sadowsky, Michael Alexander ('43) (B), Assoc. Prof. Math., Ill. Inst. of Tech., 3300 Federal St., Chicago 16, Ill.
Sadwith, Howard Marvin (J'38) (CLMDK), V.P., Indus. Washing Mch. Corp., 289 Burnet St., New Brunswick; *for mail,* 735 Pemberton Ave., Plainfield, N.J.
Saenger, David Mangold (J'46) (CFK), 607 San Miguel, Berkeley, Calif.
Saenger, George Waldemar ('19; '35) (CMLDF), Managing Dir., Maize Products Pty. Ltd., Footscray, W. 11, Victoria, Australia.
Saff, Leonard Carl (J'46) (CJM), Jamestown Fabricated Steel Co., 135 Hopkins Ave.; *for mail,* 321 E. 5th St., Jamestown, N.Y.
Safford, Joseph Frederick (J'41) (SCFZHK), Mech. Engr., E. I. du Pont de Nemours & Co., Waynesboro, Va.
Sage, Darrow ('15) (SCFKE), Mech. Oper. Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Sager, E. H. ('23; '35) (CJM), Lt. Col., Chief, Prod. Serv., St. Louis Dist. Ord. Office, U.S. Court & Customs House, St. Louis; *for mail,* 1020 N. Harrison Ave., Kirkwood, Mo.
Sager, Norbert William ('27; '33; '35) (SFK), Ch. Steam Plant Engr., Pub. Serv. Dept., 172 W. Magnolia; *for mail,* 1027 E. Grinnell, Burbank, Calif.
Saginor, Sidney Victor ('38; '45) (OKM), Gen. Mgr., Davey Compressor Co., 266 N. Water St.; *for mail,* 900 Byrce Rd., Kent, Ohio.
Sagstetter, W. H. ('41) (R), Ch. Mech. Officer, Denver & Rio Grande West. R.R. Co., 1531 Stout St., Denver 1, Colo.
Saharoff, A. V. ('25) (EL), Cons. Engr., Compressor Div., Worthington Pump & Mchy. Corp., Clinton & Robert Sts., Buffalo, N.Y.
Sahlstrom, Roy (J'46) (ESZ), J. W. Murphy Co., 10 N. Clark St., Chicago; *for mail,* 3236 Balmoral Ave., Chicago 25, Ill.
Sahmel, Viggo Karl ('14) (BEOSZ), Elec. Engr., F. L. Smidth & Co., 11 W. 42nd St., New York 18, N.Y.
Sailliard, John Hector ('37) (MBZC), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14, N.Y.
St. Clair, Clinton Draper ('20; '26; '35) (CSM), V.P., Manning, Maxwell & Moore, Inc., 5 Watson St., Boston 65; *for mail,* 157 Fuller St., West Newton 65, Mass.
St. Clair, David William (J'44), U.S. Army; Box 863, Staunton, Va.
St. Clair, Frederick Grant ('31; '41) (SFKE), Supvg. Engr., Washington Univ., St. Louis 5, Mo.
St. Clair, Oscar Allen ('30) (CMGO), Prof. & Head, Dept. of Indus. Engrg. & Engrg. Drawing, Tex. Tech. College, Lubbock, Tex.
St. Eve, Edward J. (J'39) (JLMNCH), Indus. Engr., Roller Bearing Co. of Am., 4030 Chouteau Ave., St. Louis 10, Mo.
St. John, Elbert D. ('31; '35) (SFCKHO), Kan. Gas & Elec. Co., Box 208, Wichita 1, Kan.
St. John, Stuart B. ('29; '35) (CMBHND), Sales Engr., Heald Mch. Co., New Bond St., Worcester 6; *for mail,* 7 Hapgood Way, Shrewsbury, Mass.
St. Lawrence, John ('17) (CLMN), Asst. Works Mgr., Gen. Elec. Co., 2901 E. Lake Rd., Erie, Pa.
Sait, Alfred John (J'43) (ABMCDK), Supercharger Engrg. Div., Bldg. 2-32, Gen. Elec. Co., Lynn, Mass.
Salathe, Lloyd Walter (J'46), Ensign, U.S.N.R.; 1623 Lesseps St., New Orleans, La.
Saldin, Harvey B. (J'39) (BANEZ), Sec. Engr., Aviation Gas Turbine Lab., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 627—8th Ave., Prospect Park, Pa.
Salecker, Anton (J'37) (ZLMJ), Engr., Bell Tel. Labs., 463 West St., New York; *for mail,* Roosevelt Ave., Syosset, L.I., N.Y.
Salemme, Vincent James (J'43) (MNBJCD), Asst. Engr., West Side Ave. Plant, West. Elec. Co., Inc., Jersey City, N.J.; *for mail,* 4348—196th St., Flushing, N.Y.
Salibian, Mihran Hampartzoom (J'45) (CDMTYS), Engr., Dadourian Export Corp., 25 Elizabeth St., New York 13; *for mail,* 110-40—70th Ave., Forest Hills, L.I., N.Y.
Salimbene, Rocco C. (J'40) (BHS), 2nd Lt., U.S.A., 16th Engr. Bn. (A), Ft. Knox, Ky.
Salisbury, Allen (J'29) (YAKLZC), Engr., Dow Chem. Co.; *for mail,* 1414 Ashman St., Midland, Mich.
Salisbury, Chester Riley (J'44) (MSNHC), Ensign, U.S.N.R., Engrg. Officer, LST-484, F.P.O., San Francisco; *for mail,* 788 N. 9th St., Colton, Calif.
Salisbury, Donald Worcester ('42) (KFEL), Pres., Southwest. Engrg. Co., Inc., 90 West St., New York 6, N.Y.
Salisbury, Harold G. (J'32) (ZAG), Designer, Gen. Engrg. Lab., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail,* 238—5th Ave. N., Troy, N.Y.
Salisbury, J. Kenneth ('30; '42) (ESFABK), *Melville Medallist,* '42; Gen. Elec. Co., Schenectady 5; *for mail,* 2112 Niskayuna Dr., Schenectady 8, N.Y.
Salisbury, Robert William ('16) (BJR), Sr. Mech. Engr., Rail Div., Office of Pres., Trans. Corps Bd., U.S.A., N.Y. Port of Embarkation, 58th St. & 1st Ave., Brooklyn; *for mail,* 270 College Ave., Staten Island 2, N.Y.
Salkeld, Alan Bertram ('44) (BCJK), Devel. & Sales Engr., Steel Processing Co., 212 Wood St., Pittsburgh 21; *for mail,* 1731 Jamestown Pl., Pittsburgh 22, Pa.
Salkin, Max ('43) (MJB), Sr. Engr., San Francisco Ord. Dist., 100 McAllister St., San Francisco 1; *for mail,* 901 Portola Dr., San Francisco 16, Calif.

Sallmann, Gerard (J'37) (FEZK), Process Design Engr., Lummus Co., 420 Lexington Ave., New York 17; *for mail,* 235 E. 22nd St., New York 10, N.Y.
Salls, David M. (J'41) (EFKSRA), Box 212, Howard, Ohio.
Salma, Emanuel A. ('32; '41) (BSE), Instr. in Mech. Engrg., Cooper Union, Cooper Sq., New York 3; *for mail,* 230 E. 71st St., New York 21, N.Y.
Salminen, Adolph August (J'42) (MJCDBL), Tech. Asst., Am. Steel & Wire Co., 238 Fairmount Ave., New Haven, Conn.; *for mail,* 6 Watch St., Rochdale, Mass.
Salmon, Fred A., Sr. ('45) (CBK), Ch. Engr., Simplicity System Co.; *for mail,* 416 Sioux Trail, Chattanooga, Tenn.
Salmon, Joseph Harold ('24; '31) (CFE), Asst. to Pres., Shell Oil Co., Inc., 50 W. 50th St., New York 20; *for mail,* 625 The Parkway, Mamaroneck, N.Y.
Salmon, Philip Alexander ('25; '35; '35) (SFCEHM), Asst. Engr., Elec. Engrg. Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1; *for mail,* 16 Tulip Lane, Short Hills, N.J.
Salmonsen, Robert ('21; '27; '32) (BCESDF), Asst. to V.P., F. L. Smidth & Co., Suite 2400, 11 W. 42nd St., New York 18, N.Y.
Salo, Eric August ('45) (FSKCBZ), Asst. Mech. Engr., N.Y. State Elec. & Gas Corp., 62 Henry St., Binghamton; *for mail,* R.D. 2, Endicott, N.Y.
Salomonson, John Elmer (J'38) (CA), Asst. Engr., Pacific Tel. & Tel. Co., 74 Otis St., San Francisco; *for mail,* Apt. 402, 2655 Polk St., San Francisco 9, Calif.
Salter, Milo Junior (J'43), West Bend, Wis.
Salter, Tom (J'42) (AC), Cons. Engr., Cessna Aircraft Co., Wichita; *for mail,* 140 N. Clifton, Wichita 8, Kan.
Saltz, Fred (J'44) (NB), Designer, R. Hoe & Co., 138th St., Bronx; *for mail,* 6541 Saunders St., Forest Hills, N.Y.
Saltzer, Bertram Hanson ('40) (CAE), Engrg. Staff Asst., Wright Aero. Corp., Beckwith Ave., Paterson 3, N.J.
Salvage, Maurice (J'39) (BFKS), Asst. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* 1101 Union St., Brooklyn 25, N.Y.
Salveson, Melvin Erwin (J'43) (CMNHBL), Lt. Comdr., U.S.N.R., Asst. Force Engr., Submarine Force, Atlantic Fleet, F.P.O., New York, N.Y.; *for mail,* Box 53, R.F.D. 1, Old Lyme, Conn.
Salzman, Carl Ernest (J'33) (MOEBDJ), Plant Engr., Harding Glass Co.; *for mail,* 1023 S. 17th St., Ft. Smith, Ark.
Salzman, William Boyd (J'41) (SKF), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 412 Wa-Pella Ave., Mt. Prospect, Ill.
Samans, Walter ('20) (OBSDEL), Admin. Engr., Sun Oil Co., 1608 Walnut St., Philadelphia 3, Pa.
Sambach, Warren Austin (J'43) (OHKNCA), Mech. Engr., Design, with Otto J. Sambach, Cons. Engr., 51 Chambers St., New York; *for mail,* 2882 Valentine Ave., New York 58, N.Y.
Samburoff, Serge N. (J'41) (ASBH), Assoc. Engr., Aircraft Design, Engrg. & Research Corp., Riverdale; *for mail,* 4207 Tuckerman St., Hyattsville, Md.
Sammis, Edward A. (J'40) (ABZ), Project Engr., Research Labs., Sperry Gyroscope Co., Inc., Garden City; *for mail,* R.F.D., Roslyn, N.Y.
Samoiloff, L. A. (J'43), 45 Remsen St., Brooklyn 2, N.Y.
Samp, Carl Fred (J'37) (CMSFJ), Lt., U.S.N.R., Inspr. of Naval Matl., U.S.N., Niels Esperson Bldg., Houston 2; *for mail,* 3405 N. Shepherd, Houston 8, Tex.
Sample, Sam Steele ('44) (ESKL), 115 N. Old Orchard Ave., Webster Groves 19, Mo.
Sampson, Edwin M. (J'19) (ACD), Prod. Mgr., Felber Biscuit Co., Grant Ave. & McCoy St., Columbus, Ohio.
Sampson, Harold S. (J'41) (CEF), 131 W. Elliott Ave., Springfield, Ill.
Sampson, Merritt B. (J'40), Asst. Ch. Engr., Motch & Merryweather Mchy. Co., Penton Bldg., W. 3rd & Lakeside Ave., Cleveland 13; *for mail,* 11843 Lake Ave. 12, Lakewood 7, Ohio.
Sampson, Robert George (J'46), 514 Wyoming Ave., Millburn, N.J.
Sampter, Herbert C. ('16; '22; '35) (CDW), Factory Mgr., Blaisdell Pencil Co., 52 Church Lane, Philadelphia 44, Pa.
Sams, Bruce Jones ('25; '35) (DEFKMO), Dist. Engr., So. Cotton Oil Co., Lathrop Ave.; *for mail,* 526 E. 41st St., Savannah, Ga.
Sams, James Hagood, Jr. ('32; '37) (AEH), Prof. Mech. Engrg., Sch. of Engrg., Clemson Agric. College; *for mail,* 113 N. Clemson Ave., Clemson, S.C.
Samson, James Brunton ('45) (CJM), Works Mgr., Ransome & Marles Bearing Co. Ltd.; *for mail,* 35 Winchilsea Ave., Newark-on-Trent, Nottingham, England.
Samuel, Alan Julian (J'43) (KBMNAJ), Apt. 4, 510—24th Ave., San Francisco, Calif.
Samuel, Edward, Jr. (J'41) (LKH), Engr., Hercules Powder Co., Hattiesburg, Miss.
Samuel, Hubert D., Jr. (J'40) (AJM), Mech. Design Engr., W. J. Voit Rubber Corp., 1600 E. 25th St., Los Angeles 11; *for mail,* 424 N. Edinburgh Ave., Los Angeles 36, Calif.
Samuels, Sidney ('45) (BSKHF), Pres., Sidney Samuels, Inc., 165 Amsterdam Ave., New York 23; *for mail,* 245 W. 107th St., New York 25, N.Y.
Samuels, T. William (J'32) (CDM), Bardstown, Ky.
Sanborn, Elmer Edward ('31; '35) (AEFSYK), Automotive Engr., Natl. Carbon Co., Inc., 30 E. 42nd St., New York 17, N.Y.; *for mail,* 3429 Lenox Rd., N.E., Atlanta, Ga.
Sanclemente, Amaury Hipolito (J'45) (EFHNCR), Engr., Marketing Dept., Tropical Oil Co., Box 570, Bogotá, Colombia, S.A.
Sandberg, Ray Alfred (A'45) (BMLN), Ch. Engr., Oakes N. Chicago Div., Houdaille Hershey Corp., Foss Park Ave., North Chicago, Ill.
Sandbrook, John Wayne (J'40) (CELKJM), 2920 W. Lloyd St., Pensacola, Fla.
Sanders, Albert Michael (J'41) (CMAD), Lt., Army Air Forces, Matagorda Peninsula, Foster Field, Tex.; *for mail,* 67-70 Yellowstone Blvd., Forest Hills, N.Y.
Sanders, Jas. Corbin ('23; '35) (MRT), Pres., Treas., J. C. Sanders Cotton Mill Co., Inc., Box 1296, Mobile, Ala.
Sanders, John Claytor (J'37) (AEKFJ), Mech. Engr., Natl. Advis. Com. for Aero., Mun. Airport, Cleveland; *for mail,* 17422 Lake Ave., Cleveland 7, Ohio.
Sanders, Leon Herning (J'39) (CNSJMD), Staff & Plant Engrg., Babcock & Wilcox Co., Sterling St.; *for mail,* 611 Charles Ave., Barberton, Ohio.
Sanders, Murray Allan (J'45), Engr., Publix Metal Products, Inc., 100—6th Ave., New York; *for mail,* 67-70 Yellowstone Blvd., Forest Hills, N.Y.
Sanders, Victor David (J'43) (KHB), Research Engr., Rm. 207, Calif. Hall, Univ. of Calif., Berkeley; *for mail,* 2317 Le Conte Ave., Berkeley 4, Calif.
Sanders, Walter C. ('17; '21; '35) (RCJMB), Gen. Mgr., Ry. Div., Timken Roller Bearing Co., 1835 Dueber Ave., Canton 6, Ohio.
Sanderson, Robert R. ('22; '35) (AE), 181 Sussex Dr., Manhasset, L.I., N.Y.
Sanderson, Victor Louis ('41) (STL), Partner, Sanderson & Bradford, 1710 Walnut St., Philadelphia 3, Pa.
Sandgren, Marvin Arthur (J'42) (BMAY), 4147 Emerson Ave. N., Minneapolis 12, Minn.
Sandiford, Arthur (J'39) (CRSJ), Technician, Gibbs & Cox, Inc., 1 Broadway, New York 4; *for mail,* 38 E. 75th St., New York 21, N.Y.
Sandison, Alexander G. S. ('40) (KNSWZ), Asst. Ch. Engr., Waterous, Ltd.; *for mail,* 65 Brant Ave., Brantford, Ont., Can.
Sandland, Clifford Milner (J'41) (KLEHF), Process Engr., C. F. Braun & Co., 1000 S. Fremont Ave., Alhambra; *for mail,* 2422 S. Oak Knoll, San Marino 9, Calif.
Sandor, George (J'41) (GCNMY), Ch. Engr., with Harry W. Faeber, Ptg. Equip. Consultant, 551—5th Ave., New York 20, N.Y.
Sandorff, Paul Edwin (J'43) (ABJYMG), Research Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 3460 Rowena Ave., Los Angeles 27, Calif.
Sandowski, Wladyslaw (J'44), Asst. Ch. Engr., Equi-Flow, Inc., 1841 Broadway, New York; *for mail,* 755 Ocean Ave., Brooklyn 26, N.Y.
Sands, William Robert, Jr. ('46) (CKS), Field Power Consultant, Edge Moor Plant, E. I. du Pont de Nemours & Co.; *for mail,* 6 Corinne Court, Villa Monterey, Wilmington, Del.
San Fanandre, Albert John (J'44), 45 Ella St., Valley Stream, N.Y.
Sanford, Leigh Russell ('21), 4420 Brandywine St., N.W., Washington 16, D.C.
Sangster, William ('94; '02) (HJNW), Engr., DeLaval Co. Ltd., 113 Park St., Peterborough, Ont., Can.
Santaella, Jose Antonio (J'42), Box 269, Ponce, P.R.
Santamaria, Ildefonso Jorge (J'29) (FKS), Ch. Engr., Guantanamo Sugar Co., Apartado A, Guantanamo; *for mail,* Calle G, No. 411, Altos-Vedado, Havana, Cuba.
Santos, Anibal ('45) (COS), Engr., c/o Compania Colombiana de Electricidad, Barranquilla, Colombia, S.A.
Santry, Joseph V. ('35) (FKS), Pres., Dir., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Santucci, Edward Domenic (J'45), Engr., H. A. Wilson Co., 105 Chestnut St., Newark; *for mail,* 690 S. 19th St., Newark 3, N.J.
Santucci, Lawrence F. (J'37) (ABC), Engr., Republic Aviation Corp., 42 Broadway, New York; *for mail,* 485 E. 180th St., Bronx 57, N.Y.
Saper, Martin Leo (J'41), Lt., Hdq. Sqd., 51st Serv. Gp., A.P.O. 629, New York, N.Y.
Sappet, Charles Leo (J'41) (AHJ), Engr., Bell. Tel. Labs., Whippany; *for mail,* 458 Park Ave., East Orange, N.J.
Saracino, Frank Ernest ('31; '45) (SKFCL), Plant Mgr., Manistee Salt Works, 1501 Main St.; *for mail,* 500—2nd St., Manistee, Mich.
Sargent, Elbert Ritz (J'45) (ABHKEF), Research Engr., Helicopter Div., McDonnell Aircraft Corp., Lambert Field, St. Louis; *for mail,* 310 Thoroughman, Ferguson 21, Mo.
Sargent, Ralph ('28) (SKFD), Partner, Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Sargent, Ralph Nelson, Jr. (J'36) (SOMNCE), Power Engr., Johns-Manville Corp., Manville; *for mail,* 121 Myrtle Ave., Plainfield, N.J.
Sargent, Richard Boynton ('45), Gen. Mgr., Hale Fire Pump Co., 708 Spring Mill Ave., Conshohocken, Pa.
Sarlat, Irwin M. (J'39) (COMA), Engr., Edward Ermold Co., 652 Hudson St., New York 14; *for mail,* Apt. 1, 408 W. 57th St., New York 19, N.Y.
Sarles, Canneth D. (J'39) (K), Research Engr., Sunbeam Elec. Mfg. Co., 225 W. Morgan St., Evansville 7; *for mail,* 1816 Marie Ave., Evansville 11, Ind.
Sarto, Jorma Olavi (J'42) (EAHZFB), Student Engr., Chrysler Corp., 12800 Oakland, Detroit; *for mail,* 7560 Honeysuckle Rd., Route 1, Walled Lake, Mich.
Sartore, Arthur Charles (J'46), 756 N. Garland St., Memphis 7, Tenn.
Sartorius, William ('28; '35) (OLD), Chief, Constr. & Plant Engrg., Cent. Engrg. Div., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City, N.J.
Sar Vant, Wilbur Nason ('46) (CJL), Mech. Engr., Treadwell Constr. Co., Midland, Pa.
Sass, Clifton H. ('25) (CBNYKH), Mech. Supt., Simplex Wire & Cable Co., 79 Sidney St., Cambridge 39, Mass.
Sattler, Carl S. ('42; '43) (SWFK), Mech. Engr., Weyerhaeuser Timber Co., P.O. Drawer 1645, Tacoma, Wash.
Sattler, Seymour (J'45) (OMN), Exper. Engr., Chrysler Corp., Detroit; *for mail,* 150 W. Euclid, Detroit 2, Mich.
Sauby, Wesley Orville (J'41) (NBEJZY), 2nd Lt., Sig. Corps, A.U.S., R.L. Area, Ft. Monmouth, N.J.; *for mail,* 3845—18th Ave. S., Minneapolis 7, Minn.
Sauerbrunn, Edward S. ('38), Dist. Mgr., Babcock & Wilcox Co., 2511 Carew Tower, Cincinnati, Ohio.
Sauermann, W. Otto (J'37) (LKDS), Design Engr., Armour 31st St. Auxiliaries, 1355 W. 31st St., Chicago 9; *for mail,* 2032 W. 103rd St., Chicago 43, Ill.
Sauerwald, Joseph ('42) (CDM), Supvr., Gen. Elec. Co., 6901 Elmwood, Philadelphia 43; *for mail,* 4529 Spruce St., Philadelphia 39, Pa.
Saulson, Saul ('27) (EFHKLO), Mech. Engr., Albert Kahn Associated Archs. & Engrs., Inc., 345 New Center Bldg., Detroit 2; *for mail,* 12524 Broadstreet Blvd., Detroit 4, Mich.
Saunders, C. B. ('33; '35), 229 N. Maple St., Delphos, Ohio.
Saunders, Ed. F. (J'42), Lt. (j.g.), U.S.N.R., 176 Gardner St., Vallejo, Calif.
Saunders, Fred Quinn ('40; '45) (FSCE), Ch. Engr., Div. of the Budget, Commonwealth of Va., State Capitol, Richmond, Va.
Saunders, Fred S. ('18; '26) (JNB), Tool & Exper. Engr., Metal Auto Parts Co. Inc., 1428 W. Henry St., Indianapolis; *for mail,* 762 N. Riley Ave., Indianapolis 1, Ind.
Saunders, Ivan Lucian (J'40) (KMZ), Engr., Kaiser Co., Inc., Vancouver, Wash.; *for mail,* 2801 N. Kilpatrick, Portland 3, Ore.
Saunders, Ward Bishop, Jr. (J'43) (CM), 340 Mapleton Ave., Hollister, Calif.
Saunders, William H., Jr. ('18; '24; '35) (DLA), Pres., Gen. Mgr., Internatl. Lubricant Corp., Box 390, New Orleans 1, La.
Saunier, William P. ('29) (SBFK), Project Mgr., Jackson & Moreland, 31 St. James Ave., Boston 16; *for mail,* 36 Dana St., Cambridge 38, Mass.
Saurwein, George Karl ('15; '19) (SFDEBM), Lombard Governor Corp., Ashland; *for mail,* 639 Boston Post Rd., Weston, Mass.
Saus, Frank John (J'37) (CNY), Secy.-Treas., Superior Tool & Mch. Co., 1508 Kenmore Blvd.; *for mail,* 1115 Carey Ave., Akron 14, Ohio.
Sausele, George John Henry (J'45) (BCH), Research Engr., Intertype Corp., 360 Furman St., Brooklyn; *for mail,* 8623—111th St., Richmond Hill 18, N.Y.
Sauter, Wm. V. ('19; '35) (FDHSMC), Chmn., Exec. Com., Am. Engrg. Co., Aramingo Ave. & Cumberland St., Philadelphia 34, Pa.
Sautter, Robert Underwood (J'44) (MJCS), Sales Engr. (Trng.), Bantam Bearings Div., Torrington Co., South Bend 21; *for mail,* Apt. 7, 334 W. La Salle St., South Bend 12, Ind.
Sauvage, John Edwin (J'41) (NJAY), Engr., Sauvage Engrg., 4519 Harrison St., Chicago 24, Ill.
Savacchio, Antonio N. (J'39) (EBNH), Mech. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 1321 Croes Ave., New York 60, N.Y.
Savage, Kennedy (J'43) (ACJ), 18 Park Ave., Lyndhurst, N.J.
Savarese, James Joseph (J'46) (ABW), Aero. Engr., Aero. Eng. Lab., Naval Air Matl. Center, Philadelphia; *for mail,* 2300 S. 11th St., Philadelphia 48, Pa.
Savaro, V. Gregory (J'45) (ELS), 6207 Wayne Ave., Philadelphia 44, Pa.
Saveland, Walter Thomas, Jr. (J'37) (BCJ), c/o Allis-Chalmers Mfg. Co., Norwood, Ohio.
Savko, Joseph (J'41) (SFKNBE), Engr., Power & Fuel Engrg. Div., Carnegie-Ill. Steel Corp., Munhall; *for mail,* 862 Lilac St., Pittsburgh 17, Pa.
Savoye, Charles U. ('18; '26) (SFBE), Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* 350 Central Ave., Hackensack, N.J.

Savrda, Charles M. ('27; '35) (CDNJYZ), Ch. Engr., Engrg. & Mch. Div., Atlantic Coast Fisheries Co., 307 Water St., New York; *for mail,* 237 N. Windsor Ave., Brightwaters, L.I., N.Y.
Sawchyn, Stanley (J'46), Detail Engr., Exper. Test Div., Plant 2, Allison Div., Gen. Motors Corp., Indianapolis; *for mail,* Y.M.C.A., 310 N. Illinois Ave., Indianapolis 4, Ind.
Sawdon, Will Miller ('10) (KEFHS), Prof. Exper. Engrg., Emeritus, Cornell Univ.; *for mail,* 1018 E. State, Ithaca, N.Y.
Sawford, Frank ('09) (BENOS), Mech. & Elec. Engr., 510 Credit Foncier Bldg., Vancouver, B.C., Can.
Sawin, Herbert Asa ('30; '35) (CDHMN), Sales Engr., Yuba Mfg. Co., 351 California St., San Francisco 4, Calif.
Sawyer, Alfred J. (J'32) (LCHSZM), Maint. Supvr., Sharples Chemicals Inc., 4700 Biddle Ave., Wyandotte; *for mail,* 2550 Edgemont Rd., Trenton, Mich.
Sawyer, Charles Emlis (J'42) (CDM), 4101 Southern Pkwy., Louisville, Ky.
Sawyer, Clifford Lewis (J'39) (AC), Asst. Project Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3; *for mail,* Lake Trail E., Packanack Lake, N.J.
Sawyer, Fred Albert (J'37) (SFHJKB), Engr., Mech. Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.; *for mail,* 182 Waterman St., Providence, R.I.
Sawyer, George Norman (J'44), 8905 Hollis Court Blvd., Queens Village 8, N.Y.
Sawyer, Gordon Trowbridge (J'45) (EFCHM), Asst. Ch. Engr., Novo Eng. Co., Porter St., Lansing 5, Mich.
Sawyer, Harold Timberlake (J'40) (SZHL), Mgr., Seattle Dist. Office, Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio; *for mail,* 2430—42nd Ave. N., Seattle 2, Wash.
Sawyer, J. G. (J'41) (AHK), 79 Oakgrove, Williamsville, N.Y.
Sawyer, Robert Thomas ('30; '35) (ERSAKJ), Am. Loco. Co., 30 Church St., New York 8, N.Y.
Sawyer, Willits H. ('28; F'43) (ORS), Exec. Engr., 30 Broad St., New York 4, N.Y.
Saxby, Lewis Ellis ('17; '21) (CEFNMS), Francisco & Jacobus, 511—5th Ave., New York, N.Y.; *for mail,* 563 Park St., Upper Montclair, N.J.
Saxe, John Burtis ('43) (ORS), Engr., Gibbs & Hill, Inc., Pa. Sta., New York 1, N.Y.; *for mail,* 528 Birch Ave., Westfield, N.J.
Sayer, John Samuel (J'40) (LSZC), Field Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail,* 18-B Cedar Dr., Baltimore 20, Md.
Sayers, William W. ('01; '06) (D), 2373 E. 70th St., Chicago 49, Ill.
Sayles, Bertram J. (A'31) (JLC), Pres., Calorizing Co., 400 Hill Ave., Wilkinsburg, Pa.
Saylor, David Comly ('32) (FBHKSZ), Assoc. Prof. Mech. Engrg., Carnegie Inst. of Tech., Schenley Park, Pittsburgh 13, Pa.
Sayre, Charles Raymond (J'42), Ensign, LCT-723, Gp. 64, Flotilla 22, F.P.O., San Francisco, Calif.
Sayre, Mortimer F. ('22) (BNSCEO), Prof. Applied Mechanics, Dir. of Extension, Union College, Schenectady 8, N.Y.
Sayre, Richard L. ('26; '38) (CMS), Plant Mgr., Heekin Can Co., 435 New St., Cincinnati 2, Ohio.
Scagnelli, Henry John (J'45) (CKHNAM), Ensign, U.S.N.R., Gunnery-Ord. & Stores Officer, U.S.S. *Hancock,* CU-19, F.P.O., San Francisco, Calif.; *for mail,* 4429 Carpenter Ave., New York 66, N.Y.
Scaife, J. Verner, Jr. ('40), Scaife Co., Oakmont; *for mail,* Woodland Rd., Pittsburgh, Pa.
Scanland, Boyd (J'39), Designer, Hutchinson Fdy. & Steel Co., D & Washington Sts.; *for mail,* 307 W. 18th St., Hutchinson, Kan.
Scanlon, John Joseph ('42), Lt., U.S.N.R., Asst. Power Supt. (Mech.), N.Y. Naval Shipyard, Brooklyn; *for mail,* 941 Brayton Pk. Pl., Utica, N.Y.
Scarano, Ralph Michael (J'43) (CMN), V.P., Pyramid Elec. Co., 415 Tonnele Ave., Jersey City, N.J.
Scavullo, Joseph J. (J'41) (BCH), Instr., Economics Dept., Stevens Inst. of Tech., Hoboken, N.J.; *for mail,* 2212 Ave. M, Brooklyn, N.Y.
Scelba, Anthony Joseph ('42) (MJNC), Ch. Tool Engr., Leslie Co., Grant Ave., Lyndhurst; *for mail,* 8 Palmer St., Passaic, N.J.
Schaaf, George C. (J'36), Aviation Radio Technician, 1st Class, U.S.N.R., Naval Air Tech. Trng. Center, Barracks 8, Ward Island, Corpus Christi, Tex.
Schaaf, Jule Walter (J'45) (BDL), Plant Engrg. Designer, RCA Victor Div., Radio Corp. of Am.; *for mail,* 410 Grant St., Camden, N.J.
Schaal, Norbert J. ('25; '35) (ONWYL), Cons. Engr., 905—2nd Ave. Bldg., Seattle 4, Wash.
Schabtach, Carl (J'36) (BJNKSR), Turbine-Generator Engrg. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1918 Mayfair Rd., Schenectady 8, N.Y.
Schachnow, Abraham Simon (J'44), Assoc., Army Air Forces Cent. Procurement Dist., Packard Motor Car Co.; *for mail,* 2752 Tyler, Detroit 6, Mich.
Schade, Albert, III (J'45) (COSKZF), Mech. Engr., Power Div., United Engrs. & Constructors Inc., 1401 Arch St., Philadelphia 5; *for mail,* 200 Walnut Lane, Ambler, Pa.
Schaedle, Leo John ('44) (BJLNOS), Supvg. Engr., Stone & Webster, 5th & Grand, Los Angeles; *for mail,* 255—14th St., Santa Monica, Calif.
Schaefer, C. T. ('19; '30) Tech. Editor, *Automobile Digest,* 22 E. 12th St., Cincinnati, Ohio; *for mail,* 5722 Neosho St., St. Louis, Mo.
Schaefer, Conrad B. (J'27) (CDS), 407 Harrison Ave., Roselle, N.J.
Schaefer, Duane Dee (J'42) (SKBOZ), Supvg. Engr., Westinghouse Elec. Corp., 309 S.W. 6th Ave., Portland 4; *for mail,* 2515 S.E. 28th Pl., Portland 2, Ore.
Schaefer, Earl H. (J'41) (CMBJNZ), Prod. Engr., Elgin Natl. Watch Co., 107 National St.; *for mail,* 630 Congress St., Elgin, Ill.
Schaefer, Frederick LeRoy ('33) (CLMBDS), Plant Engr., Andrew Jergens Co., 2535 Spring Grove Ave.; *for mail,* Box 375, Route 4, Cincinnati 5, Ohio.
Schaefer, Fred'k R. ('11; '25) (CDJ), Gen. Mgr., Niles Steel Products Div., Republic Steel Corp., Niles, Ohio.
Schaefer, Jacob Wernli (J'41) (BZN), Capt. Ord. Dept., U.S.A., Devel. Officer, Frankford Arsenal, Philadelphia 37, Pa.
Schaeffer, Emile Joseph (J'38) (ABM), Aero. Engr., Assembly & Repair Engrg. Div., Mar. Corps Air Sta., Cherry Point, N.C.
Schaeffer, Philip Austin ('37; '46) (CDMO), Engr., Motch & Merryweather Mchy. Co., 715 Penton Bldg., Cleveland 13; *for mail,* 1500 Caswell Court, Euclid 17, Ohio.
Schaeuble, Clifford H. (J'31) (FLSZ), Power Engr., Andrew Jergens Co., 2535 Spring Grove Ave., Cincinnati 22; *for mail,* 1610 W. Belmar Pl., Cincinnati 24, Ohio.
Schafer, Robert Adam ('43) (BHMN), Ch. Devel. Engr., Natl. Automatic Tool Co., Inc.; *for mail,* 522 S.W. A St., Richmond, Ind.
Schafer, Sidney Paul ('24; '25; '35) (BNRJAH), East. & Export Mgr., Racine Tool & Mch. Co., State & Carlisle, Racine, Wis.; *for mail,* 39 N. Fullerton Ave., Montclair, N.J.
Schafer, Theodore W. D. (J'41) (CLT), Asst. to Yarn Supt., Atlantic Mills Div., A. D. Juilliard & Co., Inc., 120 Manton Ave., Providence; *for mail,* 92 Ferncrest Ave., Cranston, R. I.
Schafer, Vernon E., Jr. (J'40) (EBNSZ), Application Engr., Detroit Diesel Eng. Div., Gen. Motors Corp., 13400 W. Outer Dr., Detroit 23; *for mail,* 16587 Murray Hill, Detroit 27, Mich.
Schaff, Frederic Alan ('15; '26; F'45), Pres., Superheater Co., 60 E. 42nd St., New York 17, N.Y.
Schaffer, Lyle Edwin (J'42) (LZSEK), Design Engr., Stand. Oil Co. of Ind., Whiting, Ind.; *for mail,* 7808 Colfax, Chicago 49, Ill.
Schaffert, George A. ('40; '42) (TDG), Dir. of Maint., Burlington Mills Corp., Greensboro, N.C.
Schaffner, John William ('31) (DBMCNS), Matl. Handling Engr., Rome Mfg. Co. Div., Revere Copper & Brass Corp.; *for mail,* Apt. 3-C-C, 319 E. Bloomfield St., Rome, N.Y.
Schaidt, Leander, Jr. (J'40) (ACS), Project Weight Engr., Glenn L. Martin Co., Middle River; *for mail,* 2512 Talbot Rd., Baltimore 16, Md.
Schainker, Arnold (J'35) (JEMNSF), Serv. Engr., Shell Oil Co., Inc., Shell Bldg., St. Louis; *for mail,* 7900 Gannon St., University City 5, Mo.
Schalla, Clarence August (J'41) (MBHSCN), 2117 W. 62nd St., Cleveland 2, Ohio.
Schaller, Alwin ('11; '21) (ESB), V.P., Gen. Mgr., McEwen Bros., 1 N. Main St., Wellsville, N.Y.
Schaller, Philip H. ('44) (CKS), Steam Specialist, Westinghouse Elec. Corp., 207 W. 3rd St., Cincinnati 2; *for mail,* 4331 Ridgeview Ave., Cincinnati 5, Ohio.
Schalscha, Walter Gilbert ('43) (CMJH), Exec. V.P., Lidgerwood Mfg. Co., Elizabeth B; *for mail,* R.R. 1, Far Hills, N.J.
Schanes, Alan (J'45) (SLRHKA), Indus. Engr., Chambers Works, E. I. du Pont de Nemours & Co., Penns Grove, N.J.; *for mail,* 1200 Gilpin Ave., Wilmington 12, Del.
Schantz, Edward Arthur (J'42) (CYLMNA), Draftsman, Engr., Shaw Insulator Co., 160 Coit St.; *for mail,* 34 Greene Terrace, Irvington 11, N.J.
Schanz, Joseph Leo (J'45) (KSCNMH), Mech. Engr., Eli Lilly & Co., 740 S. Alabama, Indianapolis, Ind.
Schaper, Harry Charles ('45) (BDHKN), Div. Engr., Brass Mill, West. Cartridge Co., East Alton; *for mail,* 1206 McPherson Ave., Alton, Ill.
Schapiro, Paul (J'43) (MJNYLC), Asst. Ch. Mech. Engr., Lewyt Corp., 60 Broadway, Brooklyn 11; *for mail,* 1184 Ocean Ave., Brooklyn 30, N.Y.
Schapiro, Sylvan Benjamin ('26; '45) (CFLKOS), Asst. Gen. Plant Supt., Pan Am. Refining Corp., Box 401, Texas City, Tex.
Scharfenstein, Vernon Lewis (J'43), Lt., Hdq. & Serv. Co., 838 Engr. Bn., A.P.O. 528, c/o Postmaster, New York, N.Y.
Scharnagel, Herman J. ('19) 235 W. 22nd St., New York, N.Y.
Scharnberg, Lester N. (J'22) (CKHZ), Asst. Steam Engr., Gulf Oil Corp., Gulf Bldg., Pittsburgh 30, Pa.
Scharpenberg, Charles C. ('17; '35) (OHFES), 2300—18th St., Bakersfield, Calif.
Schatz, Warner S. (J'42) (SKEF), Lt. (j.g.), 2nd Asst. Engr., U.S. Maritime Serv.; *for mail,* 56 Ft. Washington Ave., New York 32, N.Y.
Schaub, Herman William ('40; '45) (HCMN), Works Engr., Worthington-Gamon Meter Co., 296 South St., Newark 5; *for mail,* 6 Swaine Pl., West Orange, N.J.
Schaum, Otto W. ('94) (BMNTW), Chmn. of Bd., Fletcher Works, Inc., Glenwood Ave. & 2nd St., Philadelphia 40, Pa.
Schear, Abraham (J'39) (NMBDCG), Mech. Engr., Office of Supvr. of Shipbldg., U.S.N., Cramp Shipbldg. Co., Richmond & Norris Sts., Philadelphia 25; *for mail,* 2401 Bigler Terrace, Philadelphia 45, Pa.
Schechter, Jack Eli (J'42) (OKLSBM), Air Conditioning Engr., James H. Martin Inc., 503 W. 43rd St., New York 18; *for mail,* 733 Howard Ave., Brooklyn 12, N.Y.
Schechter, John P. (J'35) (CDSLY), Mech. Engr., Austin Co., 227 Curtis Bldg., Detroit 2; *for mail,* 1812 Burns Ave., Detroit 14, Mich.
Schechter, Robert (J'41) (MZCDKO), Draftsman, Designer, Jelrus Co., 136 W. 52nd St., New York 19; *for mail,* 245 W. 107th St., New York 25, N.Y.
Scheck, Fred William (J'45) (BJHNGC), Mch. Designer, Whiting Corp., 157th St. & Lathrop Ave., Harvey; *for mail,* 10024 St. Lawrence Ave., Chicago 28, Ill.
Scheckenbach, John A. V. ('17; '21) (CJR), V.P., Mfg., Am. Car & Fdy. Co., 30 Church St.; *for mail,* 501 W. 113th St., New York, N.Y.
Scheel, Henry Van Riper ('09; '13) (OCMTWY), Cons. Engr., 630—5th Ave., New York 20, N.Y.
Scheer, Lloyd Francis (A'45) (EFS), Boiler Inspr., Dist. of Columbia, Dist. Bldg., Washington 4, D.C.; *for mail,* 18 E. Rosemont Ave., Alexandria, Va.
Schefe, Fred K. ('41), Plant Ch. Engr., Gary Sheet & Tin Mill, Carnegie-Ill. Steel Corp.; *for mail,* 636 Grant St., Gary, Ind.
Scheffel, Maxwell Lewis (J'45) (AFK), 7 Henry St., Hespeler, Ont., Can.
Scheib, Louis (J'42) (NYLBCG), Engr., Cent. Engrg. Div., U.S. Rubber Co., 1230 Ave. of Americas, New York 20; *for mail,* 1730 Carroll St., Brooklyn 13, N.Y.
Scheibe, Elias Wesley (J'40) (NEBKH), Layout Draftsman, Diesel Equip. Div., Gen. Motors Corp., Grand Rapids; *for mail,* 419 Burton St., S.W., Grand Rapids 7, Mich.
Scheibel, Albert H. ('19; '35) (FKS), Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Scheidemantle, Herbert Spenler ('39; '43) (FJSCBY), Mech. Engr., Erie Forge & Steel Co.; *for mail,* 1036 W. 10th St., Erie, Pa.
Scheidt, Harry John (J'36) (SR), Engr., Engrg. Dept., No. Equip. Co., 1945 Grove Dr.; *for mail,* 1116 Hartt Rd., Erie, Pa.
Scheidt, Karl Henry, Jr. (J'43) (SCWHYL), Maint. Engr., Brown-Forman Distillers Corp., 1908 Howard, Louisville 10; *for mail,* 113 S. 42nd St., Louisville 12, Ky.
Scheiner, Carl Joseph (J'42), Design Engr., Gen. Elec. Co., Thompson Rd., Syracuse; *for mail,* 124 Luddington St., Syracuse 6, N.Y.
Scheinman, Burton (J'45) (ANMCBO), Mech. Engr., Republic Aircraft Corp., Farmingdale; *for mail,* 265 Beach 135th St., Rockaway Beach, N.Y.
Scheirman, William Lynn (J'43) (BEJA), Ensign, U.S.N.R., ATR-13, F.P.O., San Francisco, Calif.
Schell, Albert E. ('22; '35) (MCJY), Design Engr., Stromberg-Carlson Co., 100 Carlson Rd., Rochester 3; *for mail,* 129 Stone Rd., Rochester 12, N.Y.
Schell, Erwin Haskell ('13; '21; '35) (CDO), Charge, Dept. of Business & Engrg. Admin., Mass. Inst. of Tech., Cambridge, Mass.
Schell, Frederic B., Jr. ('21; '28; '35), Comptrollers Dept., Charge of Costs, Firestone Tire & Rubber Co.; *for mail,* 1344 Jefferson Ave., Akron, Ohio.
Schell, Horace B. ('41) (NMDBH), Ch. Engr., Cole Mchy. Mfg. Corp., 227 Fulton St., New York 7; *for mail,* 216 Park Pl., Brooklyn 17, N.Y.
Schell, Philip Clifford (J'42) (ELR), Lt., Officer in Charge, Workshop Stores & Tool Sec., Royal Canadian Elec. & Mech. Engrg., Longue Pointe Ord. Depot, Montreal, Que.; *for mail,* 963 Avenue Rd., Toronto 12, Ont., Can.
Schell, William A. ('42) (CMNDSZ), Asst. Mech. Engr., Plant Engr.'s Office, Camera Works, Eastman Kodak Co., 333 State St., Rochester 4; *for mail,* 191 Valley Rd., Rochester 10, N.Y.
Scheller, Oscar Arthur (J'39) (EKYSHA), 85-25—151st St., Jamaica, L.I., N.Y.
Schellhase, Frank August (J'40) (MJAEFS), Tool Designer, Boeing Aircraft Co., Seattle 8; *for mail,* 6406 S. 120th, Seattle 88, Wash.
Schenck, Charles ('02; '16) (ABCDEF), Retired; Box 236, Coopersburg, Pa.
Schenk, Einar Sigurd (J'40) (CKOSZ), Owner, Weinrich Fence, Bronze & Iron Works, 337 Ward St., Honolulu 42; *for mail,* 3305 E. Manoa Rd., Honolulu 15, T.H.

Schenk, Everett M. (J'36) (BCDHJN), Devel. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail,* 75 Grove St., Montclair, N.J.
Schenk, Frank Edward (J'45) (CES), Ensign, U.S.S. *Bingham,* APA-225, F.P.O., San Francisco, Calif.; *for mail,* 16 Bennett Ave., Roosevelt, L.I., N.Y.
Schenk, Jan Martin (J'37) (BAES), Sr. Research Engr., Turbo Engrg. Corp., Hancock & Lalor, Trenton; *for mail,* 455 W. State St., Trenton 8, N.J.
Schenke, Kurtis Paul (J'44) (ACFEMH), Engrg. Mgr., Application Co., 205 E. 42nd St., New York 17, N.Y.
Scher, Geo. ('25), Pres., Gen. Mgr., Weber & Scher Mfg. Co., Inc., 263 Sussex Ave., Newark, N.J.
Scherer, Francis R. ('23), Arch., Supt. Sch. Bldg., Bd. of Education, 13 Fitzhugh St. S., Rochester, N.Y.
Scherer, Herman Andrew ('19; '22; '27) (BCH), Ch. Engr., Ivers-Lee Co., 215 Central Ave., Newark 4; *for mail,* 71 Pine Grove Ave., Summit, N.J.
Scherer, Walter George ('41) (KLM), Ch. Engr., Leader Iron Works, Inc., 2200 N. Jasper St., Decatur 60; *for mail,* 327 E. Kellar Lane, Decatur 46, Ill.
Scheringer, Emil W. (J'46) (AE), Test Dept., Wright Aero. Corp., Paterson; *for mail,* 195 N. Vivyen St., Bergenfield, N.J.
Scherner, John ('17; '20) (Y), Plant Engr., U.S. Rubber Co., 94 Grove St., Chicopee Falls, Mass.
Schettler, Wesley William ('44) (CEFRS), Ch. Engr., Diesel Eng. Div., Fairbanks, Morse & Co., Lawton Ave.; *for mail,* 717½ Harrison Ave., Beloit, Wis.
Scheuble, Philip A., Jr. (J'39) (CMAJ), Lt. Comdr., U.S.N.R., Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail,* 147 E. 45th St., New York, N.Y.
Schey, Ira M., Jr. (J'41) (JMNOWY), Pres., SKT Industries Inc., 94 Bowery, New York 13; *for mail,* 340 W. 71st St., New York 23, N.Y.
Schick, D. Frederick, Jr. ('31; '41) (FHS), Engr., Philadelphia Elec. Co., 900 Sansom, Philadelphia 5; *for mail,* 4517 Locust St., Philadelphia 39, Pa.
Schick, Elliot (J'44) (CDM), Indus. Engr., Charge of Project Layout Methods & Stands., Emerson Radio & Phonograph Corp., 111—8th Ave., New York; *for mail,* 910 West End Ave., New York 25, N.Y.
Schick, Herman L. (J'34), Sr. Engr., Ch., Research Div., York Safe & Lock Co., York, Pa.
Schickedanz, Louis H. (J'16) (FSDCNK), Draftsman, Engr., U.S. Coal & Coke Co.; *for mail,* Box 158, Gary, W.Va.
Schickel, Norbert H. ('40) (O), 423 Oak Ave., Ithaca, N.Y.
Schier, Oscar Bernhardt, II ('32; '40) (COS), Meetings & Divisions Assistant, A.S.M.E., 29 W. 39th St., New York 18, N.Y.
Schiff, Henry (A'45) (CDMYBT), Ch. Engr., Installations, George S. May Co., 122 E. 42nd St., New York 17; *for mail,* 541 Bronx River Rd., Yonkers, N.Y.
Schiffel, Joseph William (J'41) (BKDMCN), Lt. Comdr., U.S.N.R., Bur. of Ships, Navy Dept., Washington; *for mail,* 139 Joliet St., S.W., Washington 20, D.C.
Schildhauer, Edward ('11), Retired; 223 Bronwood Ave., Los Angeles 24, Calif.
Schiller, Frank Mathew (J'43) (AEM), 324 Cooper St., Utica 2, N.Y.
Schiller, William Alfred ('20; '35) (CDJLMW), Indus. Engr., 2320 W. 25th St., Los Angeles, Calif.
Schilling, Bernhard John ('22; '35) (EF), Pres., Mgr., Newburgh Oil Co., Inc., Box 349, Newburgh, N.Y.
Schilling, William Henry (J'44) (MBCKLD), Dist. Engr., De Vilbiss Co., 1280 W. Washington Blvd., Chicago 7; *for mail,* 8701 Austin Ave., Morton Grove, Ill.
Schillinger, Christian ('22) (FKSB), Pres., Coxe Stoker Engrg. Co., 1109 Markle Bank Bldg., Hazleton, Pa.
Schimpf, Arthur James Dyer (J'43) (EKNSC), Lt. (j.g.), U.S.N.R.; *for mail,* Demarest Ave., West Nyack, N.Y.
Schimpf, Frederick John, Jr. (J'39) (BEARSZ), Diesel Test Engr., Baldwin Loco. Works, Eddystone; *for mail,* 662 S. Highland Ave., Merion Station, Pa.
Schindeler, John William, Jr. (J'45) (SEFKNB), Test & Field Engr., Buell Engrg. Co., Inc., 70 Pine St., New York 5, N.Y.; *for mail,* 802 Park Ave., Hoboken, N.J.
Schindler, Alma Johann ('44) (NMBSHR), Designing Engr., Christensen Mch. Co., Salt Lake City; *for mail,* 2900 Kenwood St., Salt Lake City 5, Utah.
Schindler, David Blair (J'39) (CJM), Application Engr., B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., 1506—1st Natl. Bank Bldg., Pittsburgh; *for mail,* 1414 Tolma Ave., Pittsburgh 16, Pa.
Schindler, Leon W. (J'43) (BSA), Mech. Engr., Gustin-Beacon Mfg. Co., 1412 W. 12th St., Kansas City 7; *for mail,* 4746 Bernice St., Kansas City 3, Kan.
Schinnerer, Roy Louis (J'39) (ANBHKJ), Design Engr., Airesearch Mfg. Co., 9851 Sepulveda Blvd., Los Angeles 45; *for mail,* 1559 Linden Ave., Long Beach 6, Calif.
Schirokauer, Henry ('44) (BCMLNH), V.P., Shirgun Corp., 136 W. 52nd St., New York 19, N.Y.
Schlachter, Carl H. ('06), Retired; 12 Crestmont Rd., Montclair, N.J.
Schlack, Bruno ('38), Engr., Philadelphia Elec. Co., Edison Bldg., 9th & Sansom Sts., Philadelphia; *for mail,* 1415 Robinson Ave., Manoa, Upper Darby, Pa.
Schlafke, Albert Paul, Jr. (J'44) (RS), Ensign, U.S.S. *Minneapolis,* CA-36, F.P.O., San Francisco, Calif.; *for mail,* 238—94th St., Brooklyn 9, N.Y.
Schlage, Ernest L. (J'38) (BJMZ), Supvr. Devel. Engrg., Librascope, Inc., 133 E. Santa Anita Ave., Burbank; *for mail,* 808 Burlingame Ave., Burlingame, Calif.
Schlanger, Arthur Lee (J'45), Jr. Engr., Clinton Engr. Works, Tenn. Eastman Corp.; *for mail,* Rm. 236, Clayton Hall (M-7), Oak Ridge, Tenn.
Schlatter, Hugo (J'41) (BLM), Design Engr., Hercules Powder Co., Port Ewen; *for mail,* 197 Bruyn Ave., Kingston, N.Y.
Schlatter, Rudolf ('10) (BEK), Ch. Mech. Engr., Busch-Sulzer Bros.-Diesel Eng. Co., 3300 S. 2nd St., St. Louis 18, Mo.
Schlatter, Rudolph Paul (J'39) (EBCFAJ), Mech. Engr., Serv. Dept., Busch-Sulzer Bros.-Diesel Eng. Co., 3300 S. 2nd St., St. Louis 18; *for mail,* 3955 Connecticut St., St. Louis 16, Mo.
Schlatter, Willard F. (J'42) (CMS), Ensign, U.S.N.R.; 3955 Connecticut St., St. Louis 16, Mo.
Schleser, Erich (J'36) (ZLHKSE), Design Engr., Pan Am. Refining Corp.; *for mail,* Box 927, Texas City, Tex.
Schlesinger, Georg C. ('05) (BCDJLM), Consultant, 145 Ashby Rd., Loughborough, Leics, England.
Schlichter, Frederick William ('37) (BCM), Pres., Ch. Engr., Works Mgr., Hamilton Tool Co., 9th & Hanover Sts.; *for mail,* RR. 4, Hamilton, Ohio.
Schlick, Louis Frederick ('40), Lt. Comdr., E-M, U.S.N., c/o Postmaster, New York, N.Y.; *for mail,* 115 Morse Ave., Rutherford, N.J.
Schlieder, Harold Augustus ('42) (SBHJKZ), Engr., No. Equip. Co., 1945 Grove Dr.; *for mail,* 632 Sassafras St., Erie, Pa.
Schlieper, E. Jerome (J'44) (MDNCWJ), Prod., Design & Maint. Engr., Union Iron Works, 600 E. William, Decatur 30, Ill.
Schlink, Frederick John ('17; '19; '24) (ZFBK), Tech. Dir., Consumers' Research, Inc., Washington, N.J.
Schlink, Norman H. (J'40) (MD), Waterford, Conn.
Schlitt, J. L. ('19; '41) (SK), Petroleum Chemicals, Inc., Box 1637, Baltimore; *for mail,* 112 Park Dr., Catonsville 28, Md.
Schlitzkus, Robert Frederick (J'39) (MNCBYZ), Ch. Engr., Prod. Engrg. Co., 9th & Dwight Way, Berkeley, Calif.
Schlobach, George F. (J'35) (CJM), Supvr. Plant Maint., Plant 4, Internatl. Business Mchs. Corp.; *for mail,* 15 Forbus St., Poughkeepsie, N.Y.
Schlub, Carl F. (J'38) (BL), Jr. Mech. (Project) Engr., Army Air Forces, Wright Field, Dayton; *for mail,* Route 2, South Charleston, Ohio.
Schluderberg, Christian C. ('22; '35) (KLS), Mech. Engr., U.S. Indus. Chemicals Inc., 5101 Distillery Ave., Baltimore; *for mail,* 5920 Burgess Ave., Baltimore 14, Md.
Schluderberg, Donald C. (J'39) (FKS), 63-50 Wetherale St., Rego Park, L.I., N.Y.
Schmall, Henry Herman (J'42) (ZSLEHF), Serv. Engr., Bailey Meter Co., 424 Sharon Bldg., San Francisco 5, Calif.
Schmarje, Clarence F. (J'33) (NCJM), Partner, Schmarje Tool & Engrg. Co., Muscatine, Iowa.
Schmeisser, Ernest G. ('17), 3333 N. Charles St., Baltimore, Md.
Schmeisser, Wilbur J. (J'35) (JY), Engr., Belden Mfg. Co.; *for mail,* 1301 W. Main St., Richmond, Ind.
Schmelz, Willard F. (J'37), Design Engr., Ward Leonard Elec. Co., 31 South St., Mt. Vernon; *for mail,* 2527 Valentine Ave., Bronx 58, N.Y.
Schmid, Alfred W. ('30), Tool Engr., Robbins & Myers, Inc.; *for mail,* 2141 Harshman Blvd., Springfield, Ohio.
Schmid, Ben Jacob (J'39) (SHNKBE), Mech. Engr., Buffalo Niagara Elec. Corp., 300 Elec. Bldg., Buffalo 3; *for mail,* 79 Orchard Dr., Kenmore 17, N.Y.
Schmid, Carl G. ('43) (CMNO), Ch. Engr., Maint. & Prod. Engrg. Dept., Naval Ord. Plant, 21st & Arlington Ave., Indianapolis; *for mail,* 1431 Pleasant St., Indianapolis 3, Ind.
Schmid, W. E. ('36), 11 rue de Beaumont, Geneva, Switzerland.
Schmid, William Andrew (J'26), V.P., City Ice & Fuel Co., 500—5th Ave., New York 18, N.Y.
Schmidlin, Albertus E. (J'39) (BKSAFZ), Devel. Engr., Walter Kidde & Co., Inc., 675 Main St., Belleville 9; *for mail,* 7-R Prospect St., Caldwell, N.J.
Schmidt, Albert Gottfried ('21) (DO), Rep., Harnischfeger Corp., 4439 Santa Fe Ave., Los Angeles 11, Calif.
Schmidt, Alfred Carl (J'45) (LMCDH), 325 White Oak Lane, Winnetka, Ill.
Schmidt, Alfred Otto ('44) (BCMN), Research Engr., Charge Metal Cutting, Kearney & Trecker Corp., National Ave., Milwaukee 14, Wis.
Schmidt, Arthur A. ('14; '21) (CBN), Cons. Engr., West. Precipitation Corp., 1016 W. 9th St., Los Angeles 15; *for mail,* 2201 Hill Dr., Los Angeles 41, Calif.
Schmidt, Chester William (J'41), Exec. Student, Internatl. Harvester Co., Beuter & Pontiac St., Ft. Wayne; *for mail,* 507 E. Faber St., Ft. Wayne 5, Ind.
Schmidt, Elmer F. E. ('19; '29) (CEFKSY), V.P., Lone Star Gas Co., 301 S. Harwood, Dallas 1, Tex.
Schmidt, Eugene Arthur (J'25) (CMLDSW), Asst. Indus. Engr. Officer, Norfolk Naval Shipyard; *for mail,* 102 Smith Ave., Portsmouth, Va.
Schmidt, Fredk. J. (J'36), 1508 Park Ave., Ft. Wayne, Ind.
Schmidt, Fredrick William ('35; '43) (BNEMJ), Sr. Engr., Charge of Engrg. Dept., Wyrstapes Co., 200 Hudson St., New York 13; *for mail,* 257 Linden St., Brooklyn 27, N.Y.
Schmidt, George Gerald ('13; '16; '35) (EFKLOS), Sr. Partner, Connolly & Schmidt, 10 W. 37th St., New York 18, N.Y.
Schmidt, Harold August (J'41) (SOLKJ), Asst. Field Engr., Commonwealth Edison Co., Chicago; *for mail,* 6111 Nassau Ave., Chicago 31, Ill.
Schmidt, Harry ('38) (ABFJKS), Contract Engr., Carrier Corp., 405 Lexington Ave., New York 17, N.Y.; *for mail,* 130 Westville Ave., Caldwell, N.J.
Schmidt, Harry Paul ('30; '38) (BZAHNK), Prof. & Head, Dept. of Engrg., Hillyer Jr. College, Pearl & Jewell Sts., Hartford, Conn.
Schmidt, Jack Frank (J'45) (KBHJCF), Student Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1, N.Y.; *for mail,* 5917 S. Emerald Ave., Chicago 21, Ill.
Schmidt, John H. ('30; '36) (CJBDLM), Asst. to Ch. Engr., Natl. Tube Co., 1714 Frick Bldg., Pittsburgh 30, Pa.
Schmidt, Knute M. ('33; '39) (TLCZ), Research Engr., N. Am. Rayon Corp., Elizabethton, Tenn.
Schmidt, Ralph ('45), Ch. Mech. Engr., Canadian Loco. Co., Ltd., Kingston, Ont., Can.
Schmidt, Richard B. (J'37), Cons. Engr., Zephyr Hill, Springfield, Ohio.
Schmidt, Roger Wilbur (J'45) (CLZ), Engr., Norden Labs. Corp., 34 Varick St., New York, N.Y.; *for mail,* 539 Grove St., Ridgewood, N.J.
Schmidt, Roy H. (J'41) (HKS), Cpl., Army Air Forces, A.P.O. 12744, c/o Postmaster, San Francisco, Calif.
Schmidt, Warren ('45) (CJMLYD), Pres., Mgr., City Plating Works, Inc., 66 Pine St., Bridgeport 5; *for mail,* 3 Rocky Ridge Dr., Bridgeport 18, Conn.
Schmidt, Warren E. (J'43) (EFM), 17565 San Juan Dr., Detroit, Mich.
Schmidtchen, Robert Paul ('28) (MZYJH), Mech. Designer, Sperry Gyroscope Co., Stewart Ave. & Clinton Rd., Garden City; *for mail,* 37 Sylvan Pl., Valley Stream, N.Y.
Schmitt, Bernard A. ('36) (AMB), 3218 W. Polk St., Chicago, Ill.
Schmitt, George H., Jr. (J'40), Measurement Engr., New Orleans Pub. Serv. Inc., 317 Baronne St.; *for mail,* 4939 S. Galvez St., New Orleans, La.
Schmitt, William Charles (J'44) (JCOMG), Mech. Engr., Flesch & Schmitt, Inc., 118 Brown St., Rochester 6; *for mail,* 590 Beach Ave., Rochester 12, N.Y.
Schmitz, Carl Edward ('45) (HMY), Ch. Engr., Crane Packing Co., 1800 Cuyler Ave.; *for mail,* 1455 Cuyler Ave., Chicago 13, Ill.
Schmock, Carl John (J'42) (BKMNYC), Design Engr., Gen. Elec. Co., 2901 E. Lake Rd., Erie 2; *for mail,* 854 Priestly Ave., Erie, Pa.
Schmoyer, Richard L. (J'37) (BLHN), Mech. Engr., Aldrich Pump Co., 5 Pine St.; *for mail,* 1560 Liberator Ave., Allentown, Pa.
Schmudde, Albert August (J'42) (EA), Exper. Engr., Army Air Forces, Wright Field, Dayton, Ohio; *for mail,* Chamois, Mo.
Schnabel, Julius William (J'43) (ABJKO), Design Engr., Columbia Aircraft Corp.; *for mail,* 88 E. Valley Stream Blvd., Valley Stream, L.I., N.Y.
Schnacke, Richard N. (J'41), Mech. Engr., Aluminum Co. of Am.; *for mail,* 132 Beach St., Massena, N.Y.
Schnackel, Harold Clement (J'43) (SEK), 211 Stiles St., Elizabeth 3, N.J.
Schnaible, Albert Paul (J'41) (AMNHF), Engr., Products Div., Bendix Aviation Corp., South Bend; *for mail,* 1113 Prospect Dr., Mishawaka, Ind.
Schneck, Curtis Elias (J'42) (ABNJ), Aircraft Stress Analyst, G & A Aircraft, Inc., Pitcairn Field, Willow Grove; *for mail,* 1102 Pennypack Circle, Hatboro, Pa.
Schneider, Arthur John Rudolph (J'45) (ABO), Calif. Inst. of Tech., 1201 E. California St., Pasadena; *for mail,* 825 E. Del Mar St., Pasadena 5, Calif.
Schneider, Bernard Robert ('27) (BGHLNT), Ch. Tool Engr., Columbia Mch. Works Inc., 255 Chestnut St., Brooklyn, N.Y.; *for mail,* 621—6th Ave., Lyndhurst, N.J.

Schneider, Carl Albert ('26) (CEFMOS), Sr. Engr., Charge of Engrg. Dept., N.Y. Office, Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Schneider, Fred (J'46) (CMY), 69-81—108th St., Forest Hills, N.Y.
Schneider, Fred B. (J'39) (KRB), Design Engr., Loco. Div., Gen. Elec. Co., 2901 E. Lake Rd., Erie; *for mail,* 3005 Woodlawn Ave., Wesleyville, Pa.
Schneider, George Russell ('41) (HBC), Ch., Engrg. Div., U.S. Engr. Office, 300 Broadway; *for mail,* 711 N. Ash St., Little Rock, Ark.
Schneider, George William (J'42) (KLYSZD), Project Engr., Rohm & Haas Co., Bristol, Pa.; *for mail,* Mercer & Burlington Ave., Ashland, N.J.
Schneider, Gerhart Rudolf (J'44) (CMS), Research Test Engr., Chance Vought Aircraft Div., United Aircraft Corp., Stratford; *for mail,* 35 Alenier St., Hamden 14, Conn.
Schneider, Otto ('44) (OSLKNE), Asst. Ch. Engr., Charge of Engrg., Schwarz Labs., Inc., 202 E. 44th St., New York 17; *for mail,* 102-37—65th Rd., Forest Hills, L.I., N.Y.
Schneider, Raymond Milton (J'44) (DLO), Assoc. Editor, Rock Products Publ. Co., 309 W. Jackson Blvd., Chicago 6, Ill.
Schneider, Robert Edward, Jr. (J'44) (NYB JOC), Cpl., Army Air Forces, Draftsman & Remote Control Turret Mechanic, Branch 6, Sqd. A., 1060th Base Unit, Greensboro, N.C.; *for mail,* 221 S. Genois St., New Orleans 19, La.
Schneider, Robert Frederick (J'43) (MCH), Staff Sgt., Signal Corps, 9463rd Tech. Serv. Unit, Radio Propagation Unit, Holabird Sig. Depot, Baltimore, Md.; *for mail,* 3230 Baker St., San Francisco 23, Calif.
Schneider, Rollin L. (J'45) (AKS), Asst. Supt. of Utilities, Univ. of New Mex., Box 136; *for mail,* 415 Hudson Ave., Albuquerque, New Mex.
Schneider, Rudolph George (J'43) (JMTWRA), Ensign, U.S.N.R., Termination Officer, c/o Naval Inspr. of Ord., Westinghouse Elec. Corp., Sharon, Pa.; *for mail,* 6309 Meis Ave., Cincinnati 24, Ohio.
Schneider, Theodore Anton (J'41) (JSKAEL), Instr., Mech. Engrg. Dept., Newark College of Engrg., 367 High St., Newark 2; *for mail,* 239 Chestnut St., Union, N.J.
Schneider, Wm. (J'41) (BJS), Design Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* 2352 McNary Blvd., Wilkinsburg, Pa.
Schneider, William Charles ('30; '35) (CLM), Asst. Supt. Engrg. & Prod., Waterbury Brass Goods Branch, Am. Brass Co., Box 829, 26 Crane St., Waterbury 88; *for mail,* 245 Country Club Rd., Waterbury 83, Conn.
Schneitter, Lee ('21; '25; '35) (EFSJL), Diesel Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Schnitman, Armand Roy (J'46), 3640 Harvey Rd., Cleveland Heights, Ohio.
Schnitzer, A. J. (J'37) (P), 84-06—109th St., Richmond Hill 18, N.Y.
Schnitzer, Sidney (J'45) (AHL), Ch. Engr., Indus. Visualization Associates, 826 S. Flower St., Los Angeles 14, Calif.
Schoch, Edwin Foresman (J'41) (AJB), Sr. Detailer, McDonnell Aircraft Corp., Lambert Airport, St. Louis; *for mail,* 4529 Holman Lane, St. Louis 14, Mo.
Schock, Harvey Earl, Jr. (J'42) (CMJYTL), Asst. Product Mgr., RCA Victor Div., Radio Corp. of Am., Front & Cooper Sts., Camden, N.J.; *for mail,* 740 N. 63rd St., Philadelphia 31, Pa.
Schock, Lewis L., Jr. (J'40) (BCH), Wrightsville, Pa.
Schocken, Micha Joseph (J'44), c/o Schocken Office, Upper Thalbieh Rd., Jerusalem, Palestine.
Schoen, John Edmund ('25; '35) (BEJKNS), Prof. & Head, Mech. Engrg. Dept., Marquette Univ., 1515 W. Wisconsin Ave., Milwaukee 3; *for mail,* 2941 N. 38th St., Milwaukee 10, Wis.
Schoenfeld, David Morris ('26; '35) (SFKHJB), Asst. Mgr., Mar. Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Schoenfeld, Edward, Jr. (J'36) (EMS), Mar. Surveyor, Am. Bur. of Shipping, 3017 Grant Bldg., Pittsburgh 10; *for mail,* 72 S. Bryant Ave., Bellevue, Pa.
Schoenfeldt, William ('22; '29; '35) (EFKLOZ), Head, Oil Burner Lab., Socony-Vacuum Oil Co., Inc., 412 Greenpoint Ave., Brooklyn 22; *for mail,* 10 Firethorne Lane, Valley Stream, L.I., N.Y.
Schoening, Frederick C. ('20; '23; '35) (BNJ CAM), Ord. Engr., Bur. of Ord., Navy Dept., Washington, D.C.; *for mail,* 2802 Russell Rd., Alexandria, Va.
Schoenthaler, Robert (J'41) (ES), Constr. Engr., Phillips Petroleum Co.; *for mail,* 1534 S. Hickory, Bartlesville, Okla.
Schoerke, Douglas Alexander ('25; '36) (CLM JKO), Ch. Constr. & Maint. Engr., Metal & Thermit Corp., 92 Bishop St., Jersey City 4; *for mail,* 10 Maple Terrace, Maplewood, N.J.
Schoessow, Earl Edward ('46) (BJK), Engr., Babcock & Wilcox Co.; *for mail,* 560 Baird Ave., Barberton, Ohio.
Schoessow, Glen John (J'32) (BSMJE), Engr., Babcock & Wilcox Co., Barberton, Ohio.
Schofield, Felix Bowker (J'42) (CYMJAB), Lt., Royal Elec. & Mech. Engrs., Camden 85, Nottingham Rd., Long Eaton, Derbyshire, England.
Schofield, Robert Charles (J'45) (AEJ), Exper. Test Engr., Allison Div., Gen. Motors Corp.; *for mail,* 3216 Washington Blvd., Indianapolis, Ind.
Schofield, W. Richison ('27; '32) (LMB), V.P., Charge of Engrg., Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44, Pa.
Schoij, Edgar (J'39), 2836 W. 67th St., Chicago, Ill.
Scholes, Daniel R. ('18), V.P., Aermotor Co., 2500 Roosevelt Rd., Chicago 8, Ill.
Scholtz, Herman John ('43; '45) (NOBYZL), Mech. Engr., Preston Labs., Box 847; *for mail,* 908 E. Brady St., Butler, Pa.
Schomburg, Louis (J'40) (ACD), 5530 W. Sunnyside Ave., Chicago 30, Ill.
Schonberg, Harry Eric ('43) (HMN), Devel. Engr., Prest-O-Lite Co., Inc., Indianapolis, Ind.
Schonitzer, Rudolph Ignatius ('29) (COMBNL), Propr., Schonitzer Engrg. Co., 1011 Power Ave., Cleveland 14, Ohio.
Schooley, Harry Eugene (J'40) (JLM), Capt., Hdq., 13th Bn., Field Arty. Reserve Corps, Ft. Bragg, N.C.
Schor, Robert (J'46), Naval Arch., Trans. Corps, U.S.A.; *for mail,* 128 Marine Ave., Brooklyn, N.Y.
Schorling, Henry Fred ('20; '26) (KESJ), Rating Engr., Alco Products Div., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Schrader, Carl G. (J'40) (EKSBH), Mar. Engr., Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* 4112 N. 3rd Rd., Arlington, Va.
Schrader, Herman John ('39) (RBFJ), Research Prof. Theoretical & Applied Mechanics, 103-A Trans. Bldg., Univ. of Ill., Urbana, Ill.
Schrader, Thomas O., Jr. ('36) (FKS), Dist. Sales Agt., Erie City Iron Works, Erie; *for mail,* 711 Investment Bldg., Pittsburgh, Pa.
Schramm, Adolf P. C. ('44) (OMCBZJ), Cons. Engr., 212 E. 48th St., New York 17, N.Y.
Schramm, Charles Suiter (J'42) (BKHCNM), Lt. Comdr., U.S.N.R., Code 940, Rm. 1501, Bldg. 101, Naval Shipyard, San Francisco 24, Calif.
Schranz, Frederick George ('16) (HJMNRY), Cons. Engr., Mfrs. Rep., 90 W. Broadway, New York 7, N.Y.
Schreck, H. ('14) (BEFHJN), Cons. Engr., Am. Loco. Co., 30 Church St., New York 8; *for mail,* 37-15—89th St., Jackson Heights, L.I., N.Y.
Schreiber, Carl Theodore ('31) (FKRS), Research Engr., Dri-Steam Products, Inc., 29 Broadway, New York 6; *for mail,* 86-01—94th St., Woodhaven 21, N.Y.
Schreiber, Herman V. ('17) (CGH), Retired; Maitland, Fla.
Schreiber, John W. ('14; '35) (OCDJM), Ch. Constr. Engr., Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19; *for mail,* 6320 Burchfield Ave., Pittsburgh 17, Pa.
Schreiber, Norman B. ('42), Dailey, Brenner & Schreiber, Bankers Bldg., 105 W. Adams St., Chicago 4, Ill.
Schrengauer, Edwin B. (J'38) (SEFK), Contract Supvr., Babcock & Wilcox Co., Barberton; *for mail,* 248 Highland Ave., Wadsworth, Ohio.
Schrenk, Louis J. ('28) (ACFKS), Gen. Supt., Pub. Ltg. Comm., 174 Atwater St. E., Detroit 26, Mich.
Schrieber, Albert Nathan (J'38) (CMWSY), Graduate Student, Harvard Business Sch., Boston 63; *for mail,* 22 Holden St., Cambridge, Mass.
Schroder, Charles Henry ('42) (CMSD), Maj., Ord. Dept., Office of Ch. of Ord., Washington 25, D.C.; Div. Mgr., Bernite Co., 4700 Frankford Ave., Philadelphia, Pa.
Schroeder, Bernhard ('35) (FOS), Partner, Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Schroeder, Henry ('33) (AJS), Design Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Schroeder, Henry Charles ('45) (FOS), Engr., Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Schroeder, Joseph H. (J'40) (CJM), Secy., Treas., Dauber Co., 199 Marion St., Oshkosh, Wis.
Schroeder, Kenneth Allen (J'42) (LDHKJM), Jr. Engr., E. I. du Pont de Nemours & Co., Charleston, W.Va.
Schroeder, Otto ('35) (CLM), Mech. Engr., Basalt Rock Co., 8th & River Sts., Napa, Calif.
Schroeder, W. C. ('41) (EFS), Chief, Office of Synthetic Liquid Fuels, Bur. of Mines, Washington 25, D.C.
Schroeder, Wilbur C. (J'37) (YB), Mech. Engr., Plaskon Div., L. O. F. Glass Co., 2112 Sylvan Ave., Toledo; *for mail,* 909 Keil Rd., Toledo 7, Ohio.
Schroeder, William ('40) (AJBN), Research Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 9664 Wheatland Ave., Roscoe 2, Calif.
Schroepfer, John Frederick (J'42) (ADFBMO), 4557 Muir Ave., San Diego, Calif.
Schroer, Gerhard Joseph (J'43) (NJMB), Design Engr., Vendo Co., 1907 Grand, Kansas City; *for mail,* 4238 Agnes St., Kansas City 4, Mo.
Schrolucke, Virgil Herman (J'39) (BHMN), Fixture Designer, Natl. Automatic Tool Co., S. 7th & N St.; *for mail,* 88 N. 29th St., Richmond, Ind.
Schubert, Arno G. (J'31) (BKS), Asst. Prof. Mech. Engrg., Rensselaer Poly. Inst., Troy; *for mail,* 1301 Broadway, Watervliet, N.Y.
Schubert, Edward Herman ('21; '25; '35) (CR), Supt., Weir Kilby Corp., 5038 Beech St., Norwood 12, Ohio.
Schubert, Frank J. ('27; '33; '35) (NBA), Design Engr., Sperry Gyroscope Co., Marcus Ave., Lake Success, Nassau Co.; *for mail,* 3 Hewlett Point Ave., East Rockaway, N.Y.
Schubert, William (J'41) (ABCEFK), 424 N. Wilton Pl., Los Angeles 4, Calif.
Schubert, William E. ('21; '25; '29) (FHS), V.P., Gen. Mgr., Wis. Mich. Power Co., 137 W. Mill St.; *for mail,* 213 W. Prospect Ave., Appleton, Wis.
Schueler, Lyle B. ('40; '43) (SKFJEC), Engr., Mech. Engrg. Div., Am. Gas & Elec. Serv. Corp., 30 Church St., New York, N.Y.; *for mail,* 234 Jefferson Ave., Westfield, N.J.
Schuette, Robert W. ('18; '25) (CJB), Gen. Mgr., Crucible Elec. Steel Co., 123 E. 7th Ave., Homestead; *for mail,* 221 Buchanan Pl., Mt. Lebanon, Pittsburgh 16, Pa.
Schuettinger, John George (J'39) (CMLDKO), Asst. Plant Mgr., Sucrest Corp., 280 Richards St., Brooklyn 31; *for mail,* 8422—117th St., Richmond Hill 18, N.Y.
Schuetz, Frederick Faber du Faur ('03; '26) (ZLWD), Solicitor of Pats., 233 Broadway, New York 7, N.Y.
Schug, Kenneth William (J'40) (FEHKLS), Engr., Burns Bros., 415-7 S. Clinton St., Syracuse; *for mail,* R.F.D. 3, East Syracuse, N.Y.
Schuler, John Eugene (J'41) (ACL), Capt., U.S.M.C.R., VMF-914, 9th Mar. Aircraft Wing, Fleet Mar. Force, Cherry Point, N.C.
Schuler, William M. ('38) (CLO), Plant Supt., Jacob Ruppert, 1639—3rd Ave., New York 28, N.Y.
Schulte, Max J. L. ('16; '21; '35) (ACJKS), 123 Avondale Rd., Ridgewood, N.J.
Schulte, William Chester ('42) (JA), Ch. Metallurgist, Propeller Div., Curtiss-Wright Corp.; *for mail,* 30 Bowers Rd., Caldwell, N.J.
Schultz, Alfred William ('31; '42) (NBCMJY), Mech. Engr., Clearing Mch. Corp., 6499 W. 65th St., Chicago 38; *for mail,* 10801 S. Fairfield Ave., Chicago 43, Ill.
Schultz, Herbert Louis ('25; '30) (CLFH), Supt. of Maint., Carborundum Co., Niagara Falls; *for mail,* 278 Paramount Pkwy., Kenmore, N.Y.
Schultz, Joseph A. (J'31) (ACDJMY), Staff Asst. to V.P., Westinghouse Elec. Corp., Union Bank Bldg., Wood & 4th Ave., Pittsburgh; *for mail,* 6744 Penn Ave., Pittsburgh 8, Pa.
Schultz, Kenneth William ('26; '32; '35) (BE MNR), Diesel Eng. Designer, Fairbanks, Morse & Co.; *for mail,* 959 Johnson St., Beloit, Wis.
Schultz, O. Bennett ('43) (JRMNCF), Ch. Metallurgist, Engr. of Tests, Lima Loco. Works, Inc., Lima, Ohio.
Schultz, Oswald Charles ('27; '34) (CMJD), Benton Harbor Malleable Industries, Benton Harbor, Mich.
Schultz, Richard Samuel (A'43) (COM), Dir., Indus. Relations Methods, Inc., 405 Lexington Ave., New York 17, N.Y.
Schultz, Rudolph Henry ('27; '31) (GCMHTY), Pres., Schultz Engrg. Corp., 6-12 Stanwix St., Brooklyn 6, N.Y.
Schultz, William F. (J'32) (CKR), Lt., Hdq. 811, Antiaircraft Arty. Bn. (AW), A.P.O. 953, c/o Postmaster, San Francisco, Calif.
Schultze, George William ('28; '45) (BLNJDK), Engr., Nitrogen Div., Solvay Process Co., Hopewell, Va.
Schulz, Donald D. ('36) (SFK), Results Engr., Scranton Elec. Co., 507-509 Linden St., Scranton; *for mail,* 2011 Capouse Ave., Scranton 9, Pa.
Schulz, Fred J. (J'44), 3505 Cleveland Ave., Dayton 10, Ohio.
Schulze, Erwin Fred C. (J'43) (MBC), Electronics Technician's Mate, 3rd Class, U.S.N., Naval Bks., West Potomac Park, Washington, D.C.; *for mail,* 3745 Belden Ave., Chicago 47, Ill.
Schum, Eugene Charles (J'36) (ECA), Lt., U.S.N.R., Bur. of Ships, Navy Dept., 17th & Constitution Ave., Washington, D.C.; *for mail,* 909 N. Wayne Ave., Arlington, Va.
Schumacher, Edward Burt (J'43) (SKEONM), Property Disposal Officer, 9805th Tech. Serv. Unit, Ohio River Div., Corps of Engrs., Clinton Bldg., Columbus, Ohio; *for mail,* 312 Orient Way, Rutherford, N.J.
Schuman, Frank Joseph ('45), 1644 S. Van Ness Ave., Los Angeles 6, Calif.
Schumann, Alex. P. ('21; '35) (BHJM), Ch. Engr., Sales Mgr., Cincinnati Gilbert Mch. Tool Co., 3366 Beekman St., Cincinnati 23, Ohio.
Schumann, Edward Armin, Jr. (J'35) (SBK), Mech. Engr., Am. Gas & Elec. Serv. Corp., 30 Church St., New York, N.Y.; *for mail,* 436 W. Durham St., Philadelphia 19, Pa.
Schumb, Martin T. ('27), V.P., Charge of Engrg., Boston Gear Works, Inc., Quincy; *for mail,* 27 Garden St., East Milton, Mass.
Schumo, Elmer Mansfield ('42) (CDJR), V.P., Gen. Mgr., Pa. Elec. Steel Casting Co., 3rd & Arch Sts., Hamburg, Pa.
Schupp, Arthur A. ('17; '23; '28) (CL), Exec. Secy., Farmers & Manufacturers Beet Sugar Association, 507—2nd Natl. Bank Bldg.; *for mail,* 1305 Owen St., Saginaw, Mich.

Schupp, Robert William (J'43), 1455 Exposition Blvd., New Orleans, La.
Schuricht, Carl Otto, Jr. (J'26) Pvt., BMC Motor Pool, 20th QM. Gp., A.P.O. 72, c/o Postmaster, San Francisco, Calif.; *for mail,* 192-12—109th Rd., Hollis, N.Y.
Schurman, Isadore Laurence ('43) (CFM), Pres., Consumers Petroleum Co., 1725 W. 31st St., Chicago 8, Ill.
Schuster, Arthur William ('25; '35) (HKLBC), Owner, Firm of Arthur W. Schuster, 344 Powers Bldg., Rochester 4, N.Y.
Schuster, Edwin Conrad (J'45) (BEJ), War Emergency Pipelines, Inc., Box 1638, Cincinnati 1, Ohio; *for mail,* 1625 S. 80th, West Allis 14, Wis.
Schuster, Melvin R. (J'42) (JNCMA), Sales Engr., Aluminum Co. of Am., 2300 Power & Light Bldg., Kansas City 6, Mo.
Schutz, Harold Ralph ('37) (CMJ), Ch. Engr., Libbey Glass Div., Owens-Ill. Glass Co., Ash St., Toledo; *for mail,* 3320 Gallatin Rd., Toledo 6, Ohio.
Schuyler, William A. ('21; '35) (JMN), Firm of W. A. Schuyler, 250 W. 57th St., New York 19, N.Y.
Schwab, Gustav ('43) (FKL), Box 807, West Point Pleasant, N.J.
Schwab, Jerome Joseph (J'44), Graduate Engr., Allis-Chalmers Mfg. Co., Box 512; *for mail,* 2207 W. Kilbourn Ave., Milwaukee, Wis.
Schwabe, Kurt Otto ('43) (SMKDFH), Mech. Engr., Bechtel Bros. McCone Co., 601 W. 5th St., Los Angeles 13; *for mail,* 3535 Dunn Dr., Los Angeles 34, Calif.
Schwalbe, Robert Henry (J'44), 2341 Long Ave., St. Paul 4, Minn.
Schwanda, Theodore Frank (J'43) (BDHNS), 20 Highland Terrace, Stafford Springs, Conn.
Schwanfelder, William (J'44) (CZJMKY), Engr., Supvr. of Product Devel., Research Engrg. Corp., 296 Elm St., New Haven 4, Conn.
Schwanhausser, Edwin J. ('16; '25; '35) (CEH LBS), V.P., Charge of Sales, Worthington Pump & Mchy. Corp., 401 Worthington Ave., Harrison; *for mail,* 311 Glenside Rd., South Orange, N.J.
Schwartz, Alfred (J'36), 256 Liberty Ave., Jersey City, N.J.
Schwartz, Andrew J. ('43) (ABNZ), Retired; 32 Rhode Island Ave., Washington 1, D.C.
Schwartz, Arnold Alexander ('21) (G), Chmn. of Bd., Art Color Ptg. Co., Dunellen; *for mail,* c/o A. A. Whitford, Inc., 705 Park Ave., Plainfield, N.J.
Schwartz, Arthur Adolph ('44) (ABM), Box 88, Hertel Sta., Buffalo, N.Y.
Schwartz, Daniel M. ('42; '45) (NDC), Mech. Engr., Eimco Corp., Box 300, Salt Lake City 8, Utah.
Schwartz, Frank LeRoy ('28; '35) (SEKBF), Assoc. Prof. Mech. Engrg., 231 W. Engrg. Bldg., Univ. of Mich., Ann Arbor, Mich.
Schwartz, Harry Adolph ('07; '13) (JKBR), Mgr. of Research, Natl. Malleable & Steel Castings Co., 10600 Quincy Ave., Cleveland 6, Ohio.
Schwartz, Harry Albert ('42) (CDJMN), Gen. Mgr., Secy., Montpelier Mfg. Co.; *for mail,* 307 E. Washington, Montpelier, Ohio.
Schwartz, Herbert L. (J'42) (MAJCZY), 65 Lord Kitchener Rd., New Rochelle, N.Y.
Schwartz, Ira Arthur (J'42), Seaman, 1st Class (Radio Technician), U.S.N.R., Co. 49, Bldg. 22, Radio Materiel Sch., Treasure Island, San Francisco, Calif.
Schwartz, Kevie Waldemar ('43) (CJLMY), Secy.-Treas., Dover Industries, Inc., 2929 N. Cambell Ave., Chicago 18, Ill.
Schwartz, Nathan (J'43) (HKS), Pvt., Co. C, 1879th Engr. Aviation Bn., A.P.O. 75, c/o Postmaster, San Francisco, Calif.
Schwartz, Perry Lloyd (J'43) (ASF), 2nd Lt., Army Air Forces, Gunter Field, Montgomery, Ala.; *for mail,* 15 E. 196th St., New York 58, N.Y.
Schwartz, Sidney Thompson (J'44) (CFS), Res. Engr., MacDonald Bros., Inc., 10 High St., Boston, Mass.; *for mail,* 445 Beulah Pk., Lake Buelah, Wis.
Schwartz, William Joseph (J'43) (MLSB), Mfg. Engr., Westinghouse Elec. Corp., Sharpsville Ave., Sharon, Pa.
Schwarz, Arthur H. (J'42) (MCJBKL), Devel. Engr., Carboloy Co., Inc., Box 237, Roosevelt Park Annex; *for mail,* Rm. 754, Hotel Webster Hall, Cass & Putnam, Detroit, Mich.
Schwarz, Emil Arthur (J'34) (CJLYMS), Factory Supt., Crunden Martin Mfg. Co., 730 S. 2nd St., St. Louis 2; *for mail,* 6443 Devonshire Ave., St. Louis 9, Mo.
Schwarz, Eugene A. ('22; '28), Mgr., Geo. L. Squier Mfg. Co., 490 Broadway, Buffalo; *for mail,* 264 Washington Highway, College Hill, Snyder, N.Y.
Schwarz, Franz Herman ('88; '18) (BHS), Retired; 165 Ferry St., Lawrence, Mass.
Schwarz, Frederick William (J'41) (SFKEOR), Dist. Mgr., Am. Engrg. Co., 2447 Aramingo Ave., Philadelphia 25, Pa.; *for mail,* 233 S. Clinton Ave., Oak Park, Ill.
Schwarz, Hans (J'44) (ABEKNS), Supercharger Engr., Elliott Co., Jeannette, Pa.
Schwarz, Michael ('15; '35) (CMYJ), V.P., Crunden Martin Mfg. Co., 760 S. 2nd St., St. Louis 2; *for mail,* 5858 Itaska, St. Louis 9, Mo.
Schwebel, Edwin Carl ('22; '35) (ACJ), Plant Engr., Wadsworth Watch Case Co., 5th & Clay Sts., Dayton, Ky.; *for mail,* 3939 Elsmere Ave., Norwood 12, Ohio.
Schweighofer, Horst Martin (J'42) (YBKFMO), Technician, 4th Grade, 987th Engr. Maint. Co., A.P.O. 75, c/o Postmaster, San Francisco, Calif.; *for mail,* 211 E. 2nd St., Corning, N.Y.
Schweikart, Herbert Carl ('38; '41) (FKS), Mech. Engr., Gilbert Associates, Inc., 412 Washington St., Reading, Pa.
Schweisthal, Fred G. ('21; '35), Engr., Charge of Research Div., Stewart-Warner Corp., 1826 Diversey Pkwy., Chicago; *for mail,* 8108 N. Kostner Ave., Skokie, Ill.
Schweitzer, Paul H. ('33) (EHSABN), Prof. Engrg. Research, Pa. State College, State College, Pa.
Schweitzer, Reuben Richard ('18; '27) (H), Pres., Layne Atlantic Co., 3322 Cromwell Rd., Norfolk, Va.
Schweizer, Paul Ernest ('30; '35) (ABCEOK), Design Engr., Mech. Research Dept., Corning Glass Works, Corning, N.Y.
Schwemler, Paul Richard (J'45) (EFS), Lt. (j.g.), U.S.N.R., U.S.S. *Haven,* AH-12, F.P.O., San Francisco, Calif.
Schwendener, Harry Garfield ('29; '35) (CGOS), Engr., Purchasing Agt., Pet Milk Co., 1401 Arcade Bldg., St. Louis 1; *for mail,* 245 Union Blvd., St. Louis 8, Mo.
Schwendler, Richard Henry ('44) (ABNO), Engr., Diamond Alkali Co., Fairport; *for mail,* Jackson St., Mentor, Ohio.
Schwendner, A. Frank ('40) (SEHNYO), Control Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 516 Swarthmore Ave., Ridley Park, Pa.
Schwenk, Heino Otto ('42) (ENOCFS), V.P., Charge of Engrg. & Sales, Nickles-Schwenk Corp., 600 S. 1st St.; *for mail,* 1004 E. Central, Ponca City, Okla.
Schwennesen, H. A. ('21) (CDE), 111—84th St., Niagara Falls, N.Y.
Schwerin, Charles Henry (J'41) (KNEBJH), Engr., George J. Hagan Co., 2400 E. Carson St., Pittsburgh 3; *for mail,* 333—4th St., Beaver, Pa.
Schwerin, Frank Henry ('29) (MNBRC), Mgr. of Engrg., Duff-Norton Mfg. Co., Box 1889, Pittsburgh 30; *for mail,* 4 Kingsford Dr., Ben Avon, Pittsburgh 2, Pa.
Schwertfeger, A. J. (J'33) (CJDKMS), Supvg. Maint. Engr., E. I. du Pont de Nemours & Co., Box 100, Richland, Wash.
Scipio, Lynn A. ('12) (RA), Indus. Rehabilitation Specialist, United Nations Relief & Rehabilitation Admin., 1735 De Sales St., Washington, D.C.; *for mail,* 6802 Dartmouth Ave., College Park, Md.
Scofield, J. Harry ('38) (NCBAEM), Assoc. Prof. Mech. Engrg., Mech. Engrg. Bldg., Colo. Agric. & Mech. College, Ft. Collins, Colo.
Scorah, Ralph L. ('26; '38) (KSNFBR), Prof. Mech. Engrg., Univ. of Mo.; *for mail,* 18 Sunset Lane, Columbia, Mo.
Scordelis, Solon Lincoln (J'46) (HS), 2295 Filbert St., San Francisco 23, Calif.
Scott, Alexander Hodge ('20) (CEFS), Pres., Gen. Mgr., New Britain Gas Light Co., 25 W. Main St.; *for mail,* 339 Hart St., New Britain, Conn.
Scott, Arthur Leslie ('38; '44) (CDKSHJ), Works Mgr., Gen. Engrg. Div., John Inglis Co., Ltd., 14 Strachan Ave., Toronto; *for mail,* 9 Island Vie Blvd., Mimico, Ont., Can.
Scott, Burton D. (J'37), 1st Lt., U.S.A., 423rd Coast Arty. (AA), Ft. Bliss, Tex.
Scott, Campbell ('20) (CJM), Chmn. of Bd., Graphite Metallizing Corp., 15 Park Row, New York 7, N.Y.; *for mail,* 57 Union St., Montclair, N.J.
Scott, Charles R., Jr. (J'44), 5941 Frontenac St., Philadelphia 24, Pa.
Scott, Donald C. (J'34) (CDLT), Asst. Mgr., Ware Coupling & Nipple Co., Ware; *for mail,* Main St., Wilbraham, Mass.
Scott, Edward William (J'44) (CMOKY), Ensign, U.S.N., YMS—446, F.P.O., San Francisco, Calif.; *for mail,* 98 Cedar St., Roxbury 19, Mass.
Scott, Edwin M. ('39) (ABK), Asst. to Elec. Engr., C. C. Moore & Co., 450 Mission St., San Francisco; *for mail,* 3440 Richmond Blvd., Oakland 8, Calif.
Scott, Frederick William (J'44) (NBSYZJ), Mech. Ord. Engr., Naval Ord. Lab., Naval Gun Factory, Washington 25; *for mail,* 2013—31st St., S.E., Washington 20, D.C.
Scott, Frederick William Elmer ('38; '46) (SK OG), Owner, Mgr., Earl F. Scott & Co., 140 Westminster Dr., Atlanta, Ga.
Scott, Henry Francis ('07) (G), Box 45, West Falmouth, Mass.
Scott, Howard Wright (J'41) (MLDG), 687 Averill Ave., Rochester 7, N.Y.
Scott, James Bride ('96; '00) (S), Retired; 503 S. 46th St., Philadelphia 43, Pa.
Scott, John M. (J'40) (BMP), Ensign, U.S.N.R., A.O.O. Sch. Bldg. 611, Naval Aviation Tech. Trng. Center, Jacksonville, Fla.
Scott, Oliver McFarlane (J'40) (A), Capt., Hdq., 80th Fighter Gp., A.P.O. 629, New York, N.Y.
Scott, Robert (J'37) (ABM), Jr. Mech. Engr., Bur. of Ord., Navy Dept., 17th & Constitution Ave., Washington, D.C.; *for mail,* 346 E. 78th St., New York 21, N.Y.
Scott, Robert John (J'43) (CAM), Boulevard Manor Hotel, 1115 E. Armour Blvd., Kansas City, Mo.
Scott, Roger Monroe ('32; '36) (NTBC), Ch. Engr., New England Butt Co., 304 Pearl St., Providence 7; *for mail,* 108 Charles Field St., Providence 6, R.I.
Scott, Rossiter Stockton ('15) (EJKOY), Dist. Engr., Dresser Mfg. Div., Dresser Industries, Inc., 122 E. 42nd St., New York 17, N.Y.
Scott, Walter B. (J'40), 59 Falls Ave., Manchester, N.H.
Scott, William Ernest ('32) (CSFDJZ), Plant Engr., Continental Can Co., Inc., 16th & Coles Sts., Jersey City 2; *for mail,* 283 Bergen Ave., Jersey City 5, N.J.
Scott, William Orville Craig ('36) (CEJMSZ), Asst. Supt., Dominion Bridge Co. Ltd., 275 W. 1st Ave.; *for mail,* 3308 Slocan St., Vancouver, B.C., Can.
Scott, Wray Moore (A'46), Contrg. Engr., 2811 Dodge St., Omaha 2, Neb.
Scottron, Victor Everett (J'45) (HBAZS), Assoc. in Mech. Engrg., Columbia Univ., Pupin Bldg., New York 27, N.Y.
Scoville, James Douglas ('37) (ABH), Asst. Ch. Engr., S. Morgan Smith Co.; *for mail,* Box 1082, York, Pa.
Scoville, Warren E., Jr. (J'34) (BDL), Devel. Dept., U.S. Rubber Co., 1 Market St., Passaic; *for mail,* 58 N. Spring Garden Ave., Nutley, N.J.
Scranton, George Jenks (J'37) (ACDEM), Inspc. Dept., Ford Motor Co., 3000 Schaefer Rd.; *for mail,* 22611 Morley Ave., Dearborn, Mich.
Scranton, William Maxwell (J'42) (CAM), Graduate Sch. of Business Admin., Harvard Univ., Soldier's Field; *for mail,* D-21, McCulloch Hall, Harvard Business Sch., Boston 63, Mass.
Scribner, Claude Lewis ('42) (MDJC), Ch. Engr., Elgin Softener Corp., 134 N. Grove Ave., Elgin, Ill.
Scrivener, Richard Harding ('36; '46) (DLOCM), Cons. Engr., Rm. 521, 57 Bloor St. W., Toronto, Ont., Can.
Scrivenor, Arthur ('01) (MLCSEO), Cons. Engr., Pat. Atty., Mutual Bldg., Richmond 19; *for mail,* 1002 West Ave., Richmond 20, Va.
Scruggs, Edwin Lyle ('45) (CSTHFM), Ch. Engr., Springs Cotton Mills; *for mail,* Box 710, Lancaster, S.C.
Scudder, Charles Morrison ('44) (BJL), Asst. Engr., A. O. Smith Corp., Milwaukee; *for mail,* 1433 N. 70th St., Wauwatosa, Wis.
Scutt, Edwin Dimon ('44) (ZSEGFO), Dr.; Field Engr., Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44; *for mail,* 5 Chelfield Rd., North Hills, Pa.
Seago, Robert Mallard ('42) (EFHKSZ), Effic. Engr., New Orleans Pub. Serv., Inc., 317 Market St., New Orleans 13, La.
Seaholm, Knut Einar ('45) (BDGHMN), Mech. Engr., Charge of Engrg., Peco Mfg. Corp., 3000 N. Hope St., Philadelphia 33, Pa.
Seale, William John (J'45) (ABS), Asst. Gen. Foreman, Testing Div., Gen. Elec. Co.; *for mail,* 849 Harrison Ave., Schenectady, N.Y.
Sealey, Philip Theodore (J'44) (ACMNLB), Mech. Engr., Equip. Lab., Air Tech. Serv. Comd., Army Air Forces, Wright Field, Dayton; *for mail,* 63 W. Wren Circle, Dayton 10, Ohio.
Seaman, Joseph ('28; '35) (EHJNW), Propr., Saw Shop, 808 Greenwood Ave., Jackson, Mich.
Seaquist, William H. ('40), Chief, Div. of Design & Constr., Natl. Bur. of Stands.; *for mail,* 219 Rittenhouse St., Washington, D.C.
Seargeant, Wm. A. (J'41), Goodyear Airplane Parts Co., Litchfield; *for mail,* Cashion, Ariz.
Searing, Hudson Roy ('44) (CS), Exec. V.P., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Searl, John (J'36) (SLAO), Engr., Tide Water Associated Oil Co., E. 22nd St., Bayonne, N.J.; *for mail,* 156 Bay Terrace, Staten Island 6, N.Y.
Searle, Russell M. ('18; '21; '35) (CMGB), Secy., Advertising Specialty National Association, 1426 G St., N.W., Washington 5, D.C.; *for mail,* 19 Winston Dr., Bethesda 14, Md.
Searle, Thomas Charles (J'41) (CMBN), Ensign, U.S.N.R.; *for mail,* Curtis Rd., Fountain City, Tenn.
Searle, Wilbur Chapin ('09; '21; '35) (CDNK FW), Mech. Engr., Leland-Gifford Co., 1001 Southbridge St., Worcester 1; *for mail,* 1299 Pleasant St., Worcester 2, Mass.
Searle, Wm. F., Jr. ('40), 3605 Kingston Pike, Knoxville, Tenn.
Searles, Elwood Fay ('24; '35; '35) (SCFLEO), N.Y. Dist. Sales Office Rep., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* Box 5, R.F.D. 1, Red Bank, N.J.
Searles, William Linton ('42) (CB), Asst. Ch. Engr., Rockefeller Center, Inc., 50 Rockefeller Plaza, New York 20, N.Y.; *for mail,* 1349 Watchung Ave., Plainfield, N.J.
Sears, Harold R. ('14; '21; '35) (SJB), Mech. Engr., Dept. of Water & Power, City of Los Angeles, 207 S. Broadway, Los Angeles 13; *for mail,* 941 S. Bonnie Brae St., Los Angeles 6, Calif.
Sears, Harold Thompson ('20; '33) (EFKS), Design Engr., Phillips Petroleum Co., Bartlesville, Okla.

Sears, Miles F. (J'38) (AKS), Mech. Engrg. Draftsman, Superheater Co., 151st St. & Railroad Ave., East Chicago; *for mail*, c/o O. L. Sears, Route 2, Valparaiso, Ind.

Sears, Walton Harvey ('37) (HKSB), Mech. Engr., Metro. Dist. Water Supply Comm., 20 Somerset St., Boston 8, Mass.

Seaton, Laurence Froyd ('40) (CDFMOS), Oper. Supt., Univ. of Neb., Lincoln 8, Neb.

Seaton, Roy Andrew ('13; '15; '16; F'45) (BCFE), Dean, Sch. of Engrg. & Arch., Kan. State College, Manhattan, Kan.

Seawell, Samuel Davis (J'43) (ABG), Officer-Candidate, Army Air Forces, Sqd. 3, Gp. A, Officer Candidate Sch., Miami Beach, Fla.; *for mail*, C-1, Kenmore Apts., Danville, Va.

Sebald, Leslie E. ('19; '25; '35) (KS), Ch. Engr., Griscom-Russell Co., 285 Madison Ave., New York, N.Y.

Seban, Ralph A. (J'39) (KFZLHF), 1611 Bancroft Way, Berkeley 3, Calif.

Sebo, E. George V. (J'44) (BEK), Liaison Engr., Consltd. Diesel Elec. Corp., 28 E. 135th St., New York; *for mail*, 2395 Palisade Ave., New York 63, N.Y.

Sebring, Frederick Archie (J'45), 135 Lasell St., West Roxbury 32, Mass.

Sebring, Robert Elwood, Jr. (J'46), Lt., U.S.A.; 135 Lasell St., West Roxbury 32, Mass.

Seck, Charles Ferdinand (J'43) (OMNCBJ), Owner, Firm of Chas. F. Seck, Engrg. Consultant, 846 Center St., Des Plaines, Ill.

Seckendorff, E. W. ('22; '35) (CL), Process Engr., Stone & Webster Engrg. Corp., Box 1751, Knoxville 10, Tenn.; *for mail*, 260 S. Orange Dr., Los Angeles 36, Calif.

Secor, Ambrose T. ('45) (FSHC), Mech. Engr., Power Prod. Div., Tenn. Valley Authority, Power Bldg.; *for mail*, Box 453, Chattanooga, Tenn.

Secor, William Leonard (J'41) (CJM), Box 597-K, Route 16, W. 79th St., Indianapolis, Ind.

Secoy, Lauren Daniel ('44), Closure Div., Anchor-Hocking Glass Corp., Connellsville, Pa.; *for mail*, 118 Lincoln Pl., Wadsworth, Ohio.

Sedat, Gustave William (J'42) (KLE), Project Engr., Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Calif.

Sedgwick, Earl H. ('99; '25), 168 Howard St., Passaic, N.J.

Sedgwick, Harry Abbey ('15) (NYHJ), Retired; R.F.D. 1, Nashotah, Wis.

See, George Langdon (J'43), 122 Bradley, Peoria, Ill.

Seeder, Carl A. ('29; '36) (BJA), Plant Engr., Wyman-Gordon Co., 146th & Wood St., Harvey, Ill.

Seekins, Addison Washburn ('30; '35) (SBDC), Steam Engr., Celanese Corp. of Am., 180 Madison Ave., New York; *for mail*, 40 Utterby Rd., Malverne, N.Y.

Seelaus, John J. (J'40) (ABS), Ensign, E-V(S), U.S.N.R.; *for mail*, Bancroft & Barry Aves., Bay Ridge, Annapolis, Md.

Seeley, Lauren E. ('25; '31), Dean, College of Tech., Univ. of N.H., Durham, N.H.

Seeley, Wirt D. ('41) (FKSC), Union Iron Works, Inc., 623 Graybar Bldg., 420 Lexington Ave., New York 17, N.Y.

Seelig, Charles B. (J'40) (SEFKC), Sales Engr., Turbine Specialist, Gen. Elec. Co., 920 Western Ave., West Lynn, Mass.

Seely, Fred B. ('27) (ABHJNY), Prof. Theoretical & Applied Mechanics, 214 Talbot Lab., Univ. of Ill., Urbana, Ill.

Seely, Ralph W. ('46) (BJL), Project Engr., Consltd. Steel Corp., Los Angeles; *for mail*, 820 S. Oakland Ave., Pasadena 5, Calif.

Seely, Warner (A'24) (M), Secy., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3, Ohio.

Seem, Charles Benjamin ('11; '19; '24) (HO), Box 2, Route 1, Zionsville, Pa.

Sefcik, Edward Frank (J'46), Jr. Designer, Black & Veatch, Cons. Engrs.; *for mail*, 215 W. Dartmouth Rd., Kansas City, Mo.

Sefing, Nicholas R. (J'39) (CMJD), Field Engr., Mgmt. Consultant, Albert Ramond & Associates, Inc., Tribune Tower, Chicago, Ill.; *for mail*, Box 535, Falls Creek, Pa.

Segall, David (J'42) (MCDAO), 2320 Kings Highway, Brooklyn 29, N.Y.

Segall, Karl B. ('29), Pres., Gen. Mgr., Automatic Htg. & Cooling Co., 14025 Hamilton St., Detroit; *for mail*, 1629 Virginia Pk., Detroit 6, Mich.

Segel, Joseph ('21; '35) (BEFS), Retired; 1008 Madison St., Wilmington 16, Del.

Segeler, John Calvin ('38) (COSF), Ch. Engr., Univ. of Chicago, 6101 Blackstone Ave., Chicago 37, Ill.

Segl, Walter E. ('23; '35) (CFS), Tech. Engr., Hercules Powder Co., 900 Market St., Wilmington 99; *for mail*, 603 Marsh Rd., Wilmington 281, Del.

Segre, Paul Mark (J'45) (EMS), 3rd Asst. Engr., U.S. Maritime Comm., 50 W. 106th St., New York 25, N.Y.

Segur, Asa Bertrand ('19; '35) (CM), Owner, A.B. Segur & Co., 1135 S. Ridgeland Ave., Oak Park, Ill.

Seibert, Charles Jay ('24), 6310—8th St., N.W., Washington, D.C.

Seibert, Rolf H. (J'40) (EJR), Am. Blower Corp., 50 W. 40th St., New York, N.Y.

Seid, Bernard (J'43), Pvt., Army Air Forces, U.S.A., 795th Tech. Sch. Sqd., Bks. 1237, Seymour Johnson Field, Goldsboro, N.C.; *for mail*, 111-03—103rd Ave., Richmond Hill, N.Y.

Seid, Robert Bernard (J'45) (BMZ), Mech. Engr., Hudson Am. Corp., 300 Pearl St., Brooklyn; *for mail*, 1926 Walton Ave., New York 53, N.Y.

Seiden, Norman Jack (J'45) (CMJD), Engr., Unique Metal Specialties Co., 70-02—34th Ave., Jackson Heights, L.I., N.Y.

Seidl, Julius C. G. ('37) (CDM), Prof. Mgmt., Sch. of Business, Fordham Univ., 302 Broadway, New York 7; *for mail*, 266 W. 23rd St., New York 11, N.Y.

Seidler, Mason F. (J'26) (CMJDYW), 412 Washington Ave., Cliffside Park 10, N.J.

Seifert, Robert Frederick (J'39) (BL), Mech. Engr., Monsanto Chem. Co., 1700 S. 2nd St.; *for mail*, 5819 Lindenwood Ave., St. Louis 9, Mo.

Seifried, Paul E. (J'41), Sperry Gyroscope Co., Inc.; *for mail*, 380 Washington St., Hempstead, N.Y.

Seigel, Arnold Elliott (J'45), 1540 Geranium St., N.W., Washington 12, D.C.

Seim, Orville Stanley (J'43) (EYSKH), Mech. Engr., Milwaukee Gas Specialty Co., 722 N. Jackson, Milwaukee 1; *for mail*, 3847 S. Kinnickinnic Ave., Milwaukee 7, Wis.

Seip, Norman Walter (J'40) (CRAW), Sales Engr., Gen. Elec. Co., 112 N. 4th St., St. Louis 2; *for mail*, 5315 Sunshine Dr., St. Louis 9, Mo.

Seiple, Morris Paul (J'43) (MB), Ord. Engr., Projectile Fuse Unit Head, Naval Ord. Lab., Naval Shipyard, Washington, D.C.; *for mail*, 4200 Kaywood Dr., Mt. Rainier, Md.

Seitz, Raymond J. (J'42), 163-24—89th Ave., Jamaica, N.Y.

Selden, Dudley ('42), V.P., Erie City Iron Works, Box 1678, Perry Sq. Sta., Erie, Pa.

Selig, Ernest T., Jr. ('40; '43) (SFL), c/o Rybolt Heater Co., Ashland, Ohio.

Selkirk, Richard Herbert (J'38), Area Engr., Charge of Projects, Ala. Ord. Plant, E. I. du Pont de Nemours & Co., Box 270; *for mail*, 5 Washington St., Sylacauga, Ala.

Selkirk, William Marshall ('17) (EFA), Retired; 725 Broad Ave., Belle Vernon, Pa.

Selleck, Everett Dale (J'44), Pvt., U.S. Army, 808 Cheyenne Ave., Alliance, Neb.

Selleck, Robert William (J'40) (NDBKL), Mech. Engr., Magnuson Engrs., 841 Alameda; *for mail*, Box 150A, Route 4, San Jose, Calif.

Sellers, Andrew August ('44) (YSCFKD), Asst. Gen. Supt., Butadiene Div., Koppers Co., Inc., Box 92, Monaca, Pa.

Sellers, Coleman, III ('28) (MJCNDO), Asst. to Supt. of Mch. Shops, Midvale Co., 4300 Wissahickon Ave., Philadelphia 40; *for mail*, 17 St. Pauls Rd., Ardmore, Pa.

Sellers, George Spencer ('37; '45) (LKSZFD), Plant Engr., Colgate-Palmolive-Peet Co., 6th & Carlton, Berkeley 2; *for mail*, 2408 Virginia St., Berkeley 4, Calif.

Sellers, William Noble (J'36) (LMTYHC), Asst. Ch. Engr., Am. Bemberg Corp.; *for mail*, 362 Pine Hill Rd., Elizabethton, Tenn.

Sellman, Nils Ture ('14; '23) (FESC), Exec. V.P., Westchester Lighting Co., 9 S. 1st Ave., Mt. Vernon, N.Y.

Selser, Thomas W. ('11; '14; '16) (EFP), Engr., Macco Constr. Co., 815 Paramount Blvd., Clearwater; *for mail*, 2341 Fargo St., Los Angeles, Calif.

Seltzer, Henri Schneider ('37; '37) (CMD), Treas., Bengue, Inc., 2023 Kerrigan Ave., Union City, N.J.

Selvey, Arthur M. ('30; '39) (CDEFHJ), Ch. Engr., Shell Chem. Corp., Box 431, Pittsburg, Calif.

Selvey, William M. ('35) (FKS), Cons. Engr., The Ridings, West Grove, Walton-on-Thames, Surrey, England.

Semble, Sam (J'42) (MCLJD), Asst. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny, N.J.; *for mail*, 462 Herzl St., Brooklyn 12, N.Y.

Semchuk, Peter (J'37) (DLMOCR), Pvt., 1st Class, Tabulating Mch. Oper., 108th Mch. Records Unit (Mobile), Ft. Bragg, N.C.; *for mail*, 324—64th St., West New York, N.J.

Semino, Angelo F. ('44) (KOCEFH), Cons. Engr., Box 69, Route 1, Mountain View, Calif.

Semonin, Emmet Victor (J'43) (JMNGYA), Design Engr., Rim Div., Goodyear Tire & Rubber Co., E. Market St., Akron; *for mail*, 404 Foursome Ave., Akron 3, Ohio.

Semple, David McHardy ('19; '28) (MCHDKN), Managing Dir., Mirrlees Watson Co., Ltd., 45 Scotland St., Glasgow, C5, Scotland.

Semple, George Phillip (J'40) (HBNLC), Sales Engr., Hyd. Press Mfg. Co.; *for mail*, 311 W. High, Mt. Gilead, Ohio.

Sencebaugh, Clarence K. ('22; '35) (BCEHLN), Plant Engr., Designer, Ryan Aero. Co., 2701 Harbor Dr., San Diego; *for mail*, 3612—31st St., San Diego 4, Calif.

Senft, John H. (J'40) (ACP), Lub. Engr., Shell Oil Co., Inc., 624 S. Michigan; *for mail*, 1343 Touhy Ave., Chicago, Ill.

Senger, Werner Irving ('40; '41) (BNAJCG), Mgr., Balancing Mch. Div., Gisholt Mch. Co., 1200 E. Washington Ave., Madison 3, Wis.

Sengstaken, Charles William ('43) (SZ), Partner, Simler & Sengstaken, 271 Madison Ave., New York 16, N.Y.

Sengstaken, Donald John (J'46) (EFS), Ensign, LST-84, F.P.O., San Francisco, Calif.

Sengstaken, John Henry ('15; '25; '35) (FSLE), East. Mgr., West. Precipitation Corp., 405 Lexington Ave., New York 17, N.Y.

Senk, Edmund P. (J'39) (EFK), 5655 Windsor Ave., Chicago 30, Ill.

Senner, Arthur Henry ('39) (SEKF), Mech. Engr., Gas Turbine Testing, Exper. Sta., Navy Dept., Annapolis; *for mail*, 3242 Abell Ave., Baltimore 18, Md.

Sennstrom, Harold R. (J'41) (BRE), Asst. Supt., Diesel Research Lab., Am. Loco. Co., Schenectady 5, N.Y.

Senseman, William Burgess ('27) (LSF), West. Mgr., Raymond Pulverizer Div., Combustion Engrg. Co., Inc., 406 S. Main St., Los Angeles 13, Calif.

Senser, Lisle Herbert, Jr. (J'44) (EAOCJN), Apt. 102, 1700 Abingdon Dr., Alexandria, Va.

Sensibar, Ezra ('43) (SJDHEF), Pres., Compania de Fomento Termoelectrico, S.A., 5 de Mayo 20, Mexico, D.F., Mex.

Sentance, Lawrence Crawley (J'42) (CMN), Dir., Mfg. Methods Dept., Canadian Westinghouse Co., Ltd., Hamilton, Ont., Can.

Sepic, Joseph Barth (J'45) (AO), Asst. Liaison Engr., Curtiss-Wright Corp., Caldwell; *for mail*, 39 Walnut St., Livingston, N.J.

Sepulveda, Rafael A. (J'39) (BJZO), Mech. Engr., Box 667, Mayaguez, P.R.

Serrell, John J. (J'41) (BMCAJ), Research Engr., Research Labs., Sharples Corp., 201 Spring Garden St., Philadelphia 23, Pa

Serrell, Morton Albert (J'46) (MK), Penn Indus. Instrument Corp., 3116 N. 17th St., Philadelphia; *for mail*, 611 Wynnewood Pk., Wynnewood, Pa.

Serrell, Peter Van Horne (J'36) (ABM), 999 N. Los Robles Ave., Pasadena 6, Calif.

Service, Robert Ross (J'43) (CDH), Gen. Mech. Engr., Engrg. Dept., Canadian Breweries Ltd., 16 Gould St., Toronto 2; *for mail*, Apt. 11, 549 Jarvis St., Toronto 5, Ont., Can.

Sessions, Frank Lord ('10) (OT), Partner, Sessions & Sessions, Rockefeller Bldg., Cleveland 13, Ohio.

Sessions, Robert Crighton ('41) (ABCOE), Asst. Mgr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 1020 Estill Dr., Lakewood 7, Ohio.

Sessler, Norman Arthur (J'45) (ABHKSM), Test Engr., River Works, Gen. Elec. Co., Lynn; *for mail*, 60 Lewis Ave., Winthrop, Mass.

Sessums, Thomas Redding (J'45), Box 662, Overton, Tex.

Setchell, John E. ('21; '32) (CKS), Asst. Prof. Mech. Engrg., Brooklyn Poly. Inst., 99 Livingston St., Brooklyn; *for mail*, 7923—13th Ave., Brooklyn 28, N.Y.

Setchell, John Stanford (J'35) (FEKRSG), Asst. Utilization Engr., American Gas Association, 420 Lexington Ave., New York 17; *for mail*, Box 62, Babylon, N.Y.

Sether, John A. ('27: '35), Ch. Engr., Hotel St. George, 51 Clark St.; *for mail*, 8202—10th Ave., Brooklyn, N.Y.

Seton, Bertram Wyndham ('31) (JMHKBS), Propr., Seton Engrg.-Inspc. Co., 660 St. Catherine St. W., Montreal 2, Que. Can.

Setterholm, Vernon Miles (J'45) (BHL), Design Engr., Wilmot Castle Co., 1255 Univ. Ave., Rochester 7, N.Y.

Setterlund, Gordon George (J'43) (ABJER), Asst. Mech. Engr., Elec. Dept., Mech. Engrg. Sec., Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.

Seutter, Louis ('18; '22; '27) (CLMAWB), Cons. Indus. Engr., 5604 W. Washington Blvd., Milwaukee 8, Wis.

Severding, Hunter William (J'44) (JSY), 1550 Barton St., Cincinnati 14, Ohio.

Severn, Harry Anthony (J'40) (CJR), Gen. Foreman, Mch. Shop, Rustless Iron & Steel Corp., 3400 E. Chase St.; *for mail*, 4621 Northwood Dr., Baltimore, Md.

Severs, Benjamin Clark (J'42) (SKF), Naval Inspr., Field Prod. Dept., N. Y. Naval Shipyard, Brooklyn, N.Y.; *for mail*, 107 Upland Terrace, Bala-Cynwyd, Pa.

Severs, Elmer Bonnell ('17) (DEFHKS), Power Engr., United Engrs. & Constructors Inc., 1401 Arch St., Philadelphia 5; *for mail*, 107 Upland Terrace, Bala-Cynwyd, Pa.

Severson, Asbjorn M. (J'40) (AJE), Ensign, 2427—23rd Ave. S., Minneapolis 4, Minn.

Severy, Lawrence Arthur ('45) (CGO), Devel. Engr., United Shoe Mchy. Corp., Beverly; *for mail*, 165 Atlantic Ave., Marblehead, Mass.

Sevier, Horace Alvin (J'42) (HBNMES), Project Engr., Gerotor May Corp., Maryland Ave. & Oliver St., Baltimore 3; *for mail*, 2627 Fait Ave., Baltimore 24, Md.

Seward, Herbert Lee ('12; '14; '18; F'43) (CMBSF), Prof. Mech. & Mar. Engrg., Yale Univ., 211 Strathcona Hall, New Haven 11, Conn.

Sewell, J. G. Clifton ('92; '25), Retired; 77 Dinsmore Ave., Crafton, Pittsburgh, Pa.

Sexton, George Stuart (J'43) (ABC), 127 Downing St., Buffalo, N.Y.

Sexton, Joseph Matthew (J'42) (LMJCSB), Matls. Inspc. Engr., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.; *for mail*, 160—2nd Ave., Newark 4, N.J.

Sexton, Reginald Mason (J'44) (MERSZC), Pvt., U.S.A., Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.
Sexton, Samuel Budd, III (J'34) (CHSAM), Sr. Admin. Asst., Safe Harbor Water Power Corp., Lexington Bldg., Baltimore 3, Md.
Seybold, Roscoe ('44), V.P., Controller, Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30; *for mail*, Churchill Rd., Pittsburgh 21, Pa.
Seybolt, Robert Strickland (J'42) (BKS), Lt., U.S.N.R.; 144 Rice Ave., Port Richmond, S.I. 2, N.Y.
Seyfarth, Francis (J'36) (ABJ), Assoc. in Mech. Engrg., Dept. of Mech. Engrg., 107 Trans. Bldg., Univ. of Ill., Urbana, Ill.
Seyfried, Warren Raymond, Jr. (J'45) (ACJM), Lt., U.S.N.R., Asst. to Prod. Supt., Assembly & Repair Dept., Naval Air Sta., Alameda, Calif.
Seymour, Edgar Duff ('45) (ABJNZO), Devel. Engr., Devel. Dept., Camera Works, Eastman Kodak Co., 333 State St., Rochester 4; *for mail*, 17 Ayer St., Rochester 13, N.Y.
Seymour, John Harold (J'46) (CHM), Instr. in Engrg. Drawing, Univ. of Toronto, Toronto; *for mail*, 29 Tudor St., Ajax, Ont., Can.
Shaal, Lester F. (J'29) (CDL), Coordinator of Supply & Trans., Atlantic Refining Co., Box 1354, Providence; *for mail*, 39 Norwood Ave., Providence 5, R.I.
Shade, Walter R. (J'40) (FJS), R.D. 1, Temple, Pa.
Shafer, Albert Hieber ('44) (ZLOYCK), Branch Mgr., Foxboro Co., 5151 Baum Blvd., Pittsburgh 24, Pa.
Shafer, Donald Vercil (J'45) (AMCKEH), Lt. (j.g.), A(L), U.S.N.R., Engrg. Officer, VB-98, F.P.O., San Francisco, Calif.
Shafer, Maurice (J'45) (SKFDNJ), Asst. to Pres., Kennedy Van Saun Mfg. & Engrg. Corp., 2 Park Ave., New York; *for mail*, 30 Clinton Pl., New Rochelle, N.Y.
Shaffer, Bernard William (J'44) (BNAJ), Research Equip. Design Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 998 Parkwood Dr., Cleveland 8, Ohio.
Shaffer, Milton Wilshire (J'28) (SLDFJM), Plant Engr., Am. Ice Co., 10th & Caldwell Sts., Philadelphia 23; *for mail*, 7629 Fayette St., Philadelphia 38, Pa.
Shaffer, Thomas Graham ('19; '23), Pres., Thomas G. Shaffer, Inc., 49 Pearl St., Hartford 3, Conn.
Shaffer, William George (J'42) (AFJ), Test Engr., Gen. Elec. Co., 69 Elmwood St., Philadelphia; *for mail*, West Point, Pa.
Shaffer, William Roth (J'41) (BNA), Research Engr., Mellon Inst. of Indus. Research, 4400—5th Ave., Pittsburgh 13; *for mail*, 4126 Franklin Rd., Pittsburgh 14, Pa.
Shaffner, Charles Richard ('16; '23; '30) (SFD EKM), Asst. Mgr. Raw Matl., Fuel & Power Div., Carnegie-Ill. Steel Corp., 208 S. La Salle St., Chicago 90, Ill.
Shahin, Edward S. (J'46) (ABE), Lt. (j.g.), U.S.N.R.; 219 E. 28th St., New York 16, N.Y.
Shailer, Harold Read (J'41) (ESFO), Asst. Fuel Engr., Chase Brass & Copper Co., Grand St., Waterbury, Conn.
Shairman, Alvin Henry (J'43) (NMYAC), Research Engr., Servomechanisms Lab., Mass. Inst. of Tech., Cambridge 19; *for mail*, 114 St. Paul St., Brookline 46, Mass.
Shakman, James G. ('21; '35) (CL), V.P., Pabst Brewing Co., 221 W. La Salle St., Chicago, Ill.
Shakun, Frank ('40) (CGM), 311 W. 66th St., New York, N.Y.
Shallenberger, George Glass ('19; '35) (CDJ YA), Pres., Lake Mining Co., 1424 Pioneer Bldg., St. Paul 1, Minn.
Shallenberger, William Hugus ('38; '44) (ABE), Prof. Mech. Engrg., Univ. of So. Calif., University Park; *for mail*, 421 S. Oxford Ave., Los Angeles, Calif.
Shames, Albert A. (J'37) (BHAKDM), Staff Mem., Radiation Lab., Mass. Inst. of Tech., Massachusetts Ave., Cambridge; *for mail*, 25 Allen Ave., Waban 68, Mass.
Shames, Sidney J. (J'41) (ANHBEJ), Project Engr., Stratos Corp., Babylon, L.I.; *for mail*, 1568 Macombs Rd., New York 52, N.Y.
Shanahan, John J. (J'41) (SKF), Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 6 Jefferson St., Port Washington, N.Y.
Shank, James L. (J'37) (FKS), Lt. Comdr., U.S.N., Staff, Serv. Sqd. 8, Serv. Force, Pacific Fleet, c/o Postmaster, San Francisco, Calif.
Shank, John Martin (J'36) (LDACZJ), Sales Mgr., Dings Magnetic Separator Co., Milwaukee; *for mail*, 4141 N. Murray Ave., Milwaukee 11, Wis.
Shank, Maurice Edwin (J'42) (BJM), 39 Jerome Ave., New Rochelle, N.Y.
Shank, Philip R. (J'41) (JMSNZ), Mech. Engr., Yarnall-Waring Co., Chestnut Hill, Philadelphia 18; *for mail*, 212 Airdale Rd., Rosemont, Pa.
Shannon, Baker Lee (J'45) (EKP), Box 31, Pierce, Tex.
Shannon, Francis Patrick ('32; '41) (BKNL), Asst. Ch. Engr., Refrig. Dept., Henry Vogt Mch. Co., 10th & Ormsby Sts., Louisville 10; *for mail*, 504 W. Ormsby St., Louisville 3, Ky.
Shannon, Lester Norman ('27; '35) (CJM), V.P., Factory Mgr., Stockham Pipe Fittings Co., Box 2592, Birmingham 2, Ala.
Shannon, Robert Howard (J'43) (SKE), Lt. (j.g.), E-V (S), U.S.N.R., Naval Landing Forces Equip. Depot, Norfolk, Va.; *for mail*, 1152 Wycombe Ave., Darby, Pa.
Shannon, Wilbur R. (J'39) (FRS), Asst. Fuel Supvr., No. Pacific Ry. Co., 176 E. 5th St., St. Paul 1; *for mail*, 50 W. Summit Ave., St. Paul 2, Minn.
Shannon, William Boyd ('37) (SBKFJN), Ch. Asst. Constr. Engr., London Power Co., Ltd., Ergon House, Horseferry Rd., London, S.W. 1; *for mail*, Upton, 52 Parkside Way, North Harrow, Middlesex, England.
Shanor, Edgar E. ('23) (BDJK), Plant Engr., Plant 2, Gen. Am. Trans. Corp.; *for mail*, 190 Cedar Ave., Sharon, Pa.
Shapiro, Ascher H. (J'42) (BAKHZL), Asst. Prof. Mech. Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge, Mass.
Shapiro, Daniel Harry (J'43) (ABEFJY), Layout Draftsman, Engrg. Dept., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford 5; *for mail*, 60 Gillett St., Hartford 5, Conn.
Shapiro, Nathan (J'43) (AEHBJS), Project Engr., Stand. Aircraft Prod. Corp., 345 Hudson St., New York; *for mail*, 57 W. 175th St., Bronx 53, N.Y.
Shapiro, Ralph Benjamin (J'45), 10 Montgomery St., New York, N.Y.
Sharko, Sam (J'37) (MAJS), Exec. Asst. to Ch. Tool Engr., Edo Aircraft Corp., College Point; *for mail*, 35-33—204th St., Bayside, L.I., N.Y.
Sharp, Campbell King ('42) (CMS), Pres., Gen. Mgr., Sharvania Oil & Grease Corp. 1185 Tulley St., Memphis 1, Tenn.
Sharp, J. Cecil (J'43) (CGK), 1255 Westminster Ave., Salt Lake City 5, Utah.
Sharp, R. E. Brunswick ('18) (HS), Cons. Engr., S. Morgan Smith Co., York, Pa.
Sharp, Robert Gordon (J'40) (KCBM), Sales Engr., Refrigeration Equip. Co., 1515—4th Ave., San Diego 1, Calif.
Sharp, Robert W. ('37) (EMS), 771 N.E. 82nd St., Miami 38, Fla.
Sharpe, Charles Evan (J'36) (CANYML), Mfg. Engr., Plant B-1, Lockheed Aircraft Corp.; *for mail*, 1141 N. Orchard Dr., Burbank, Calif.
Sharpe, Harry William ('43) (BCEJRS), Mech. Engr., Temiskaming & No. Ont. Ry., Regina St.; *for mail*, 170—1st Ave. E., North Bay, Ont., Can.
Sharpe, Henry Dexter (A'01) (NJC), Pres., Treas., Brown & Sharpe Mfg. Co., 234 Promenade St., Providence 1; *for mail*, 84 Prospect St., Providence 6, R.I.
Sharpe, Irwin Paul (J'45), Sec. Engr., Kellex Corp., 233 Broadway, New York; *for mail*, 8051—188th St., Jamaica 3, N.Y.
Shattuck, Charles Henry ('14; '20; '35) (CEM), Gen. Mgr., Stand. Steel Corp., 5001 S. Boyle Ave., Los Angeles 11; *for mail*, 1585 Lombardy Rd., Pasadena 5, Calif.
Shaver, Perry E. ('34) (BCEJMN), Sales Engr., Sun Shipbldg. & Dry Dock Co., Chester; *for mail*, 40 Linden Ave., Lansdowne, Pa.
Shaw, Alan David (J'44), 3847 Meadowbrook Blvd., University Heights, Ohio.
Shaw, Benjamin Franklin, II ('30; A'39; '45) (CS), Pres., Benjamin F. Shaw Co., 2nd & Lombard Sts., Wilmington 99, Del.
Shaw, Burton Edward ('29; '36) (ZNMKAC), Exec. Engr., Photoswitch, Inc., 77 Broadway, Cambridge 42; *for mail*, East Boxford, Mass.
Shaw, Harold Wilson (J'35) (CMO), Asst., Sales Dept., Monarch Marking System Co., 216 S. Torrence Ave., Dayton; *for mail*, 300 Spirea Dr., Dayton 9, Ohio.
Shaw, Herbert G. ('16; '25; '35) (JM), Designer, Sargent & Co., Water St., New Haven; *for mail*, 30 Atwater Ave., Derby, Conn.
Shaw, James Cicero ('09) (AEFHJO), Retired; 1261 Prospect St., La Jolla, Calif.
Shaw, John E., Jr. (J'46) (BCL), Freeport Sulphur Co., Port Sulphur, La.
Shaw, John Francis ('40; '46) (BH), Design Engr., West. Oil Tool & Engrg. Co., 4100 San Fernando Rd., Glendale; *for mail*, 1879 E. Mendocino St., Altadena, Calif.
Shaw, Joseph Henry ('14; '43) (JLOYS), Tech. Supvr., Mech. & Elec. Sec., Engrg. Dept., E. I. du Pont de Nemours & Co., 10th & Market Sts., Wilmington 98; *for mail*, 1415 Woodlawn Ave., Wilmington 73, Del.
Shaw, Louis Eaton ('13) (BAEGNY), 146 Lafayette St., Newark 5, N.J.
Shaw, Peter A. (J'39), Design Draftsman, Apparatus Devel., Bell Tel. Labs., Inc., Murray Hill; *for mail*, Box 211, Millington, N.J.
Shaw, Wilbur Augustus (J'41) (KHBSEL), Devel. Engr., Alco Products Div., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Shawcross, J. Gilbert ('45) (JPS), Supt., Pipe Fabricating Shop, McArdle & Cooney, Inc., 2320 E. Tioga St., Philadelphia 34; *for mail*, Apt. D-20, M & Bristol Sts., Philadelphia 24, Pa.
Shawk, Harry Abel ('29) (CEFHJ), Comdr. E-V(S), U.S.N.R., Comd. Officer, Sec. Base, Eureka, Calif.
Shayne, Alexander ('18; '19) (CKMO), Cons. Engr., Marcalus Mfg. Co., River Rd., East Paterson, N.J.; *for mail*, 1 W. 64th St., New York 23, N.Y.
Shea, James Andrew (J'42), Serv. Engr., Sperry Gyroscope Co., Inc., Manhattan Bridge Plaza, Brooklyn; *for mail*, Apt. 3-A, 14 Beach Rd., Great Neck, N.Y.
Shea, John Richard ('17; '35) (CMNTY), Works Mgr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore 24, Md.
Shea, Thomas Martin (J'45), Plant Engr., Sharp & Dohme, Inc., 640 N. Broad St., Philadelphia 1; *for mail*, 3311 Indian Queen Lane, Philadelphia 29, Pa.
Sheaffer, Ervin F. (J'41) (BLJKA), Mech. Engr., M. W. Kellogg Co., 225 Broadway, New York, N.Y.; *for mail*, 209 Chestnut St., Roselle Park, N.J.
Sheare, E. J. (J'30) (ZLYT), Ch. Engr., Taylor Instrument Cos. of Can., Ltd., 110 Church St.; *for mail*, 496 Glenholme Ave., Toronto, Ont., Can.
Shearer, David Robert ('17; '35) (OBHSWZ), Engr., Box 502, Johnson City, Tenn.
Shearer, Jesse Lowen (J'44) (ABH), Tool Engrg. Dept., Climax Engrg. Co.; *for mail*, 324—8th Ave. S., Clinton, Iowa.
Sheda, Raymond M. (J'41) (ESJLZ), Mech. Engr., Denver Fire Clay Co., 2301 Blake, Denver 17; *for mail*, 421 S. Emerson, Denver, Colo.
Shedd, Fred R. ('31; '35) (FKCSZJ), Supt. of Plants, Ohio Power Co., 606—2nd St., Canton 2, Ohio.
Shedd, Ward R. ('21) (CFS), Plant Engr., Barber-Colman Co.; *for mail*, 719 Ashland Ave., Rockford, Ill.
Sheehan, Daniel Joseph ('39) (RCSDJ), Supt. M.P., Chicago & East. Ill. R.R. Co., Danville, Ill.
Sheehan, Edward Frank ('42) (SBH), Capt., Corps of Engrs., 1662nd Engr. Constr. Det., Camp Claiborne, La.; *for mail*, 580 E. Rosalie St., Philadelphia 20, Pa.
Sheehan, Edward W. (J'33) (CMJRA), Plant Engr., Ry. Steel Spring Div., Am. Loco. Co., 23rd & Euclid; *for mail*, 1509 Euclid Ave., Chicago Heights, Ill.
Sheehan, Thomas Henry, Jr. (J'42) (ONMKHB), Ch. Engr., F. L. Jacobs Co., Spruce St., Detroit; *for mail*, 16205 Washburn Ave., Detroit 21, Mich.
Sheehan, Thomas Vincent ('40) (ABCELZ), Asst. Head Engr., Oil Design, Stand. Oil Co. of Ind., 119th & Front St., Whiting, Ind.; *for mail*, 138-33—224th St., Laurelton, L.I., N.Y.
Sheehan, William Mark ('20) (RJYC), V.P., Gen. Steel Castings Corp., Eddystone, Pa.
Sheeran, Lloyd Alan ('43) (MNOHZC), Pres., Dir., R. I. Task Engrg. Co., 77 Reservoir Ave., Providence 7; *for mail*, 111 Magnolia St., Cranston 10, R.I.
Sheets, Herman Ernest ('41; '45) (BEH), Dir. of Research, Appliance Corp. of Am., 4235 N. Port Washington Rd., Milwaukee 12; *for mail*, 4444 W. Medford Ave., Milwaukee 10, Wis.
Sheflin, Bob W. (J'41) (EFHAB), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 1340 W. 105th St., Cleveland 2, Ohio.
Sheketoff, Harry Mayer (J'46), 57 Deerfield Ave., Hartford 5, Conn.
Shelden, Roger F. (J'37) (BCDGHM), Lt., U.S.N.R., Safety Officer, Naval Shipyard, Mare Island, Calif.
Shelden, William Leslie ('36) (BHSCOM), Mech. Engr., Canadian & Gen. Finance Co. Ltd., 25 King St. W., Toronto, Ont., Can.
Sheldon, Arthur Noyes ('16) (TSFKBE), Propr., Mgr., F. P. Sheldon & Son, 1038 Hospital Trust Bldg., Providence 3, R.I.
Sheldon, Harold Horton ('43) (OZACM), Box 2618, Miami, Fla.
Sheldon, Lucian A. ('13) (SAFRBK), Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 809 Lakewood Ave., Schenectady 8, N.Y.
Sheldon, Marshall Barnitt (J'36) (CLAMDJ), Asst. Pur. Agt., Freeport Sulphur Co., 122 E. 42nd St., New York 17; *for mail*, Box 442, Thornwood, N.Y.
Sheldon, Otis C. ('32; '35) (FKS), Dist. Mgr., Riley Stoker Corp., 103 Park Ave., New York 17, N.Y.
Sheldon, Stanley Simon (J'39) (BAMNZJ), Prod. Engr., Tool Designer, DeJur Amsco Corp., 45-01 Northern Blvd., Long Island City; *for mail*, 1695 Grand Ave., New York 53, N.Y.
Sheldon, William Dunster, Jr. ('28; '40) (ABH KS), Ch. Engr., Sheldons Ltd., Galt, Ont., Can.
Shelley, Van Cleve (J'38) (ABR), Sr. Test Engr., Wright Aero. Corp., Paterson, N. J.; *for mail*, 3817 Review Pl., New York 63, N.Y.
Shelly, Llewellyn W. ('45) (BCHNY), Ch. Engr., Aldrich Pump Co., Allentown, Pa.
Shelson, William (J'46) (BGH), Lt., Royal Canadian Elec. & Mech. Engrs., 22 Beatrice St., Toronto 3, Ont., Can.
Shelton, Norman Thomas ('32; '43) (CMW), Indus. Engr., Hughes Aircraft Co., Culver City; *for mail*, Box 179, Chico, Calif.
Shenk, Donald Hugh ('43) (SKEF), Acting Head, Mech. Engrg., Univ. of Ala., Box 2034, University, Ala.
Shenk, Robert Henry ('41; '43) (BCHJMN), Admin. Engr., J. A. Zurn Mfg. Co.; *for mail*, 824 Cranberry St., Erie, Pa.
Shenton, F. George (J'40) (CLKHB), Designer in Chief, Gen. Labs., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail*, 449 E. Barber Ave., Woodbury, N.J.

Shepard, Berger M. (J'38) (CMS), Engr., Naval Ord. Lab., Naval Shipyard; *for mail*, 3649 Greenway Dr., S.E., Washington, D.C.
Shepard, Frank Edward ('89; '02) (BCD), Mech. Engr., Denver Equip. Co., 1400—17th St., Denver; *for mail*, Apt. 11, 1120 E. 10th Ave., Denver 3, Colo.
Shepard, Frederick Johnson, Jr. ('17; '29) (DMN), Treas., Lewis-Shepard Co., 125 Walnut St., Watertown 72, Mass.
Shepard, Ralph Hungerford ('32; '35) (NAMR), Design Engr., Nathan Mfg. Co., 416 E. 106th St., New York 23, N.Y.
Shepard, Simeon ('04; '13) (ACDLMO), Mgr., Washington Office, James Stewart & Co., Inc., 1038 Washington Bldg., Washington 5, D.C.
Sheperdson, John W. ('08; '11), V.P., Morgan Constr. Co., 15 Belmont St., Worcester, Mass.
Shepherd, Burchard Post, Jr. (J'37) (LKCW), Maj., Corps of Engrs.; *for mail*, 1924 N.E. Clackamas St., Portland 12, Ore.
Shepherd, George Woodthorpe, Jr. (J'40) (AHS), *Student Award*, '40; Asst. Prof. Mech. Engrg., Princeton Univ.; *for mail*, 31 Palmer Sq., Princeton, N.J.
Shepherd, James Livingston ('44), Research Engr., Ga. Coastal Plain Exper. Sta., Tifton, Ga.
Shepley, Raymond (J'37) (LMOC), Engr., Okeelanta Growers & Processors Cooperative, Okeelanta; *for mail*, 914 Hillcrest Blvd., West Palm Beach, Fla.
Sheppard, George William ('38) (MS), Box 23, Pomona, Calif.
Sheppard, Raymond ('41) (BHS), Control Engr., Gen. Elec. Co.; *for mail*, 1437 Clifton Park Rd., Schenectady, N.Y.
Sheppard, William Lund (J'36) (BARJY), Chief, Tech. Sec., Aircraft Eng. Div., Edw. G. Budd Mfg. Co., Philadelphia; *for mail*, 840 Montier Rd., Glenside, Pa.
Sher, David William (J'44), U.S. Merchant Mar.; 475 Poplar St., Verona, Pa.
Sherban, Daniel V. ('40) (FKZSBL), Application Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail*, R.D. 1, Keyport, N.J.
Sherfey, Miss Hester Franklin (J'42) (ODC), Shop Engr., Murray Corp. of Am., 7700 Russell, Detroit; *for mail*, 8862 Olivet, Detroit 9, Mich.
Sheridan, Hugo Grotius (J'43) (AYFE), Aerodynamicist, Glenn L. Martin Co., Baltimore; *for mail*, 5218—42nd Pl., Hyattsville, Md.
Sheridan, Loyal B. (J'38) (CMS), Lt. (S.B.) (E), c/o "B" Supt. Maint., F.M.O., Halifax, N.S., Can.
Sherilla, James M. (J'37) (DJR), Asst. Engr., U.S. Engr. Office, War Dept., 17 Battery Pl., New York; *for mail*, 49 Jackson Ave., West Hempstead, N.Y.
Sherman, Alexander (J'45), Seaman, 2nd Class, Ward 32 E, U.S. Naval Hospital, Great Lakes, Ill.; *for mail*, 1313 Eton Pl., Dayton, Ohio.
Sherman, Delmar Charles ('22; '27) (BDM), V.P., Sherman & Reilly, 1st & Broad Sts., Chattanooga 2, Tenn.
Sherman, Gordon Leslie (J'44), 7 Morningside Ave., Montclair, N.J.
Sherman, Paul Francis (J'38) (CHJLMS), Sales Engr., Union Steam Pump Co., 414 Empire Bldg., Pittsburgh 22, Pa.
Sherman, Ralph A. ('24; '30) (FRESKA), Supvr., Fuels Div., Battelle Memorial Inst., 505 King Ave., Columbus, Ohio.
Sherman, Robert Campbell (J'44) (MNKZDJ), Jr. Engr., Plant Engrg. Dept., Delco-Remy Div., Gen. Motors Corp., Anderson, Ind.
Sherman, Robert Samuel (J'43) (J), Matls. Engr., Boston Naval Shipyard, Chelsea St., Charlestown; *for mail*, 1 Atlantic St., Lynn, Mass.
Sherman, Roger J. (J'39) (ES), Lt. (j.g.), U.S.N., Ship Supt., Naval Drydocks, Terminal Island, San Pedro, Calif.
Sherman, Thomas J. F. (J'46) (AEO), Power Plant Engr., Northrop Aircraft Co., Hawthorne; *for mail*, 648 N. Hobart Blvd., Los Angeles 4, Calif.
Sherman, Victor L. ('35) (MBK), Editor, Engrg. & Applied Sci., Consltd. Book Publishers, 153 N. Michigan Ave., Chicago 1; *for mail*, 643 Hillside Ave., Glen Ellyn, Ill.
Sherman, Waldo L. ('16; '25), Secy., Treas., John Robertson Co., Inc., 133 Water St., Brooklyn, N.Y.
Sherman, Walter P. ('42), 3721—79th St., Jackson Heights, N.Y.
Sherman, Warren Severn ('22; '24) (HJCEM), Pres., Sherman Mch. & Iron Works, 18-36 E. Main St., Oklahoma City 2, Okla.
Sherman, Warren Severn, Jr. (J'43) (HJMN), Lt., U.S.N.R., Asst. Matl. Supt., U.S.N., 26 E. Main St., Oklahoma City, Okla.
Sherman, William T. ('32) (SFEKLH), Power Engr., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail*, 249 S. Warner St., Woodbury, N.J.
Shermet, Robert Martin (J'39) (ACDKOY), Gen. Mgr., Insul-Mastic Products Co., Inc., 6127-37 Cedar Ave., Philadelphia 43, Pa.
Sherrill, Bruce Elgin ('44) (SWY), Mgr., Indus. Piping Dept., Grinnell Co., Inc., 260 N. Highland Ave., N.E., Atlanta, Ga.
Sherrill, Roy L., Jr. (J'39), Student Engr., Tex. Co.; *for mail*, 2216—5th St., Port Arthur, Tex.
Sherritt, Rex Carlton (J'46) (HKL), Lt., U.S.N.R., Aircraft Maint. Officer, Assembly & Repair Dept., Naval Air Sta., Memphis 15, Tenn.
Sherrod, Charles Anthony (J'46), Engrg. Dept., Combustion Engrg. Co., Inc.; *for mail*, 202 N. Crest Rd., Chattanooga, Tenn.
Sherron, John (J'34) (FRS), Asst. Enginehouse Foreman, Morris Park, L.I., N.Y., L.I. R.R. (Pa. R.R.), Philadelphia; *for mail*, 105 Dudley Ave., Narberth, Pa.
Sherry, Leo Bernard, Jr. (J'45) (CDKMFT), Lt., Supply Corps, U.S.N.R., Asst. Supply Officer, Naval Air Sta., Navy 720, F.P.O., New York; *for mail*, 50 Parkside Court, Utica 3, N.Y.
Sherwood, Edward L. ('20; '35) (CDL), Col., U.S.A., Reserve; Cons. Engr., 24 W. 40th St., New York 18, N.Y.; *for mail*, 215 Prospect St., Ridgewood, N.J.
Sherwood, Mather Williams ('09) (NBEAM), Draftsman, Goodman Mfg. Co., 4832 S. Halsted St., Chicago 9; *for mail*, 1901 Farwell Ave., Chicago 26, Ill.
Sherwood, Noble Pierce (J'39) (EANM), Ch. Engr., Kiekhaefer Corp.; *for mail*, 59 Spring St., Cedarburg, Wis.
Shetland, Donald Victor (J'34) (SEHBKF), Technician, Oswego Steam Sta., Cent. N.Y. Power Corp.; *for mail*, 188 Conway Terrace, Oswego, N.Y.
Shetler, Alvin Elmer (J'36) (MCY), Prod. Engr., Eagle Rubber Co., 710 Orange St., Ashland, Ohio.
Shettel, W. Roy ('25) (ECSBJW), Constr. Engr., So. Calif. Gas Co., Box 3249, Terminal Annex, Los Angeles 54, Calif.
Shevlin, Arthur Robert ('44; '46) (CJ), A. R. Shevlin Co., Park Sq. Bldg., Boston, Mass.
Shideman, Edward G. (J'40) (EKS), Instr. in Mech. Engrg., Univ. of Mich., Ann Arbor; *for mail*, Box 27, Albion, Mich.
Shie, Clifford Horatio ('21; '25; '35) (SOLH), Works Engr., Diamond Alkali Co.; *for mail*, 69 Levan Dr., Painesville, Ohio.
Shields, Julian W. ('44) (BSJE), Secy., Boiler Code Comm., A.S.M.E., 29 W. 39th St., New York 18; *for mail*, 236 Ancon Ave., Pelham 65, N.Y.
Shields, Ralph I. ('32), Supt., Pacific Portland Cement Co., Redwood City; *for mail*, 1536 Cherry St., San Carlos, Calif.
Shields, William Henry ('31; '35) (CEH), Supt., So. Div., Tex.-Empire Pipe Line Co., 603 Philtower Bldg., Tulsa 2, Okla.
Shilliday, Lloyd A. (J'38) (AEM), Ensign, U.S.N.R., U.S.S. *Starlight*, F.P.O., San Francisco, Calif.
Shimazaki, Thomas Tamotsu (J'41) (KHBAN), Ellerbe & Co., St. Paul; *for mail*, 365 W. Central Ave., St. Paul 3, Minn.
Shimer, Abraham A. ('17) (CDFS), Dir. of Opera., Naval Stores Dept., Hercules Powder Co., Del. Trust Bldg.; *for mail*, 829 Harrison St., Wilmington, Del.
Shimer, John M. ('31) (HM), 4330 Emerson St., Dallas, Tex.
Shimer, John Morris, Jr. (J'38) (AE), 2nd Lt., Army Air Forces, Engrg. Officer, Matl. Comd., Dept. 60-7, Inspc. Div., Wright Field, Dayton, Ohio.
Shine, Robert Henry (J'44) (KHOG), 1st Lt., Corps of Engrs., U.S.A.; Box 389, Route 7, Birmingham 7, Ala.
Shinkle, Vincent G. ('27) (EFOS), Cons. Engr., 1701 H St., N.W., Washington 6, D.C.
Shinn, William Irvin (J'38) (CJOMN), Mech. Engr., Wm. Shinn & Co. Inc., Shallcross Ave. & B. & O. R.R., Wilmington; *for mail*, 707 W. 32nd St., Wilmington 276, Del.
Shipley, Grant Batchelder (A'07; F'45) (CMW RSL), Chmn. of Bd., Elliott Co., Jeannette, Pa.
Shipman, Lawrence A. ('38) (FKS), Fuel Engr., So. Coal & Coke Co., Inc., Hamilton Bank Bldg., Knoxville 2; *for mail*, 2118 White Ave., Knoxville 16, Tenn.
Shipman, William H. ('42) (CM), Factory Mgr., Bendix-Westinghouse Automotive Air Brake Co., Elyria; *for mail*, Box 1307, Akron, Ohio.
Shirer, Charles Curtis ('44) (SKFJBE), Steam Turbine Engr., Westinghouse Elec. Corp., 40 Wall St., New York 5; *for mail*, 51-01—39th Ave., Long Island City 4, N.Y.
Shires, Frank (J'40) (SEBNJH), Mech. Engr., Terry Steam Turbine Co., Terry Sq., Hartford 1, Conn.
Shirk, Ivan A. (J'39) (NKBJE), Prof. Mech. Engrg., Gonzaga Univ., E. 502 Boone; *for mail*, N. 1311 Lidgerwood, Spokane 11, Wash.
Shirley, Harvey J. ('18; '25; '30) (CLM), Maj., Dir. of Manpower, Hdq., Air Serv. Comd., Patterson Field, Fairfield; *for mail*, 3436 Cornell Pl., Cincinnati 20, Ohio.
Shirley, John G. ('14; '22; '27) (CMTD), Mech. Engr., Behr-Manning Corp.; *for mail*, 5 Broadview Terrace, Troy, N.Y.
Shirrell, Chas. P. ('25), Mch. Design, Rm. 131, Bldg. 2, Gen. Elec. Co.; *for mail*, 1649 Oneida St., Schenectady, N.Y.
Shirtley, Spencer Lodwick ('35; '44) (FSCK DE), Sales Mgr., Babcock & Wilcox Ltd., Park Rd., Regents Park, N.S.W., 1, Australia.
Shivek, Herbert Lewis (J'43) (CNBAZ), Engr., Modern Die & Mch. Co., 100 Union Park St., Boston 18; *for mail*, 64 McLellan St., Dorchester 21, Mass.
Shively, John Joel ('45) (BES), c/o H. R. Van Deventer, Inc., Rm. 1215, 342 Madison Ave., New York, N.Y.
Shlafmitz, Abe (J'43) (MSZACJ), 1566 Longfellow Ave., Bronx 60, N.Y.
Shneider, Harold Meyer (J'43) (MBZ), U.S.N.R.; 146 Fremont St., Harrison, N.Y.
Shoaf, Royal G., Jr. (J'39), 2237 Spring Garden St., Philadelphia 30, Pa.
Shodron, John George ('08; '21) (BEJKMZ), Asst. Prof. Mech. Engrg., Engrg. College, Marquette Univ., 1515 W. Wisconsin Ave.; *for mail*, Apt. 703, 1810 W. Wisconsin Ave., Milwaukee 3, Wis.
Shoemaker, Andrew Frederick (J'40) (EHS), Test Man, Steam Power Plant Work, Consltd. Gas, Elec. Light & Power Co. of Baltimore, Baltimore; *for mail*, 5213 Patrick Henry Dr., Baltimore 25, Md.
Shoemaker, Fred Raphael ('21; '31; '35), Assoc. Ord. Engr., Civilian Head, Specification Sec., Watervliet Arsenal; *for mail*, 512—16th St., Watervliet, N.Y.
Shoemaker, Guy Thurman ('26; 'F'45) (BCEF KS), *Manager*, '39-'42, *Vice-President*, '42-'44; V.P., Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City 10, Mo.
Shoemaker, Norris Edgar (J'43), 700 E. 10th Ave., Tarentum, Pa.
Shoemaker, Richard Woolsey ('36) (KSMNBJ), Cons. Engr., Chase Brass & Copper Co. Inc., 236 Grand St., Waterbury 91, Conn.
Shoemaker, Robert Edward (J'42) (LJCH), Maint. Engr., Carbide & Carbon Chemicals Corp., 437 MacCorkle Ave.; *for mail*, 334 Montrose Dr., South Charleston 3, W.Va.
Shoemaker, Robert Jay ('46), Ch. Engr., Magnus Metal Corp., 721 Ry. Exchange Bldg., 80 E. Jackson Blvd., Chicago 4, Ill.
Shoemaker, William H. (J'42) (JYMCH), Pvt., Army Air Forces, 1377th Base Unit, Del.-Emb. 206, Westover Field, Mass.
Shoemaker, William MacClean ('43) (CM), Staff Engr., Downingtown Mfg. Co., Downingtown; *for mail*, Sunny Acres, R.D. 2, Kennett Square, Pa.
Shoffner, John Mark (J'43) (ACDM), Lt. (j.g.), Ord., U.S.N.; *for mail*, 318 Allegheny Ave., Kittanning, Pa.
Shonnard, Harold Waldron ('19) (CHO), Cons. Engr., Edge Moor Iron Works, Inc., Edge Moor, Del.; *for mail*, 131 Clarewill Ave., Montclair, N.J.
Shook, William Merriel (J'41) (EN), Draftsman, Lima Loco. Works, Inc., Lima; *for mail*, 618 S. Main St., Ada, Ohio.
Shoop, Charles Franklin ('14) (SZFKO), Prof. Steam Engrg., Univ. of Minn., Minneapolis 8; *for mail*, 1001 E. River Rd., Minneapolis 14, Minn.
Shoop, John William (J'42) (ACD), 2nd Lt., Ground Officer, Air Corps Ferrying Comd., 12th Ferrying Sqd., New Castle County Airport, Wilmington, Del.
Shoor, Samuel (J'35) (EFK), 1820 E. 13th St., Brooklyn 29, N.Y.
Shopp, Charles John (J'43) (ECBS), Lt., U.S.N.R., Rep., Bur. of Ord. Navy Dept., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail*, 103 Lipton Ave., Trenton 8, N.J.
Shopp, William H. (J'40) (SZCFE), Plant Engr., Los Angeles County Gen. Hospital, 1200 N. State St., Los Angeles 33; *for mail*, 2229½ Johnston St., Los Angeles 31, Calif.
Shor, George G., Jr. (J'44) (FH), Lt. (j.g.), U.S.S. *Sarita*, AKA-39, F.P.O., San Francisco, Calif.
Shore, Justin Ormiston ('43) (WCMBJE), Assoc. Matls. Engr., Field Prod. Div., N.Y. Naval Shipyard, Brooklyn; *for mail*, 322 W. 72nd St., New York 23, N.Y.
Short, Byron Elliott ('27; '30; '35) (KSLEFH), Prof. Mech. Engrg., Univ. of Tex., Box 1659, Univ. Sta., Austin 12, Tex.
Short, Merle K. (J'36) (CDJML), Works Mgr., Sioux Steel Co., Sioux Falls, S.D.
Short, Robert Justin ('43) Ch. Engr., Procter & Gamble Co., Ivorydale; *for mail*, 121 Elm Ave., Wyoming, Ohio.
Shoudy, Charles Allen (J'38) (CMW), Paper Mill Supt., W.Va. Pulp & Paper Co., Box 832, Charleston 1, S.C.
Shoudy, Paul A. (J'45) (CJMYNZ), Research & Prod. Engr., Aero-Kool Pipe Co., 506 Natl. Bldg., Seattle 4; *for mail*, 1930—34th Ave. W., Seattle 99, Wash.
Shoudy, William Allen ('03; '17; F'38) (SFOC), *Vice-President*, '35-'37; Chas. H. Tompkins Co., 907—16th St., N.W., Washington 6; *for mail*, 3000—39th St., N.W., Washington 16, D.C.
Shoulders, Walter L. (J'43) (NM), Lt. (j.g.), U.S.N.R., Ship Repair, 525 Riverside Dr., Portsmouth, Va.
Shouvlin, Patrick J. ('09), Pres., Natl.-Superior Co., Springfield, Ohio.
Shovar, Clarence B. ('33) (BDELNO), Sr. Naval Arch., Water Div., San Francisco Port of Embarkation, Air Trans. Comd., Ft. Mason, San Francisco; *for mail*, 64 Nelson Ave., Mill Valley, Calif.
Showalter, Harold Jesse (J'41) (LCS), Engr., Tex. Co., Box 712; *for mail*, 731—3rd Ave., Port Arthur, Tex.
Showler, Alfred J. (J'43) (CMJD), Asst. Supt. Inspc. & Test, Gen. Elec. Co., 2901 E. Lake Rd., Erie, Pa.
Shrader, Donald Eugene (J'46) (AHS), Ch. Engr., Electroline Co., 4121 S. La Salle St., Chicago 9; *for mail*, 5314 Hyde Park Blvd., Chicago 15, Ill.

Shreeve, Charles Alfred, Jr. (J'36) (ONSEKL), Assoc. Prof. Mech. Engrg., Univ. of Md.; *for mail,* 4612 Drexel Rd., College Park, Md.
Shreve, Earl Owen ('16) (ACS), V.P., Customer Relations, Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Shriner, Charles R. (J'40), 216 S. Walnut St., Orrville, Ohio.
Shriver, William Brent (J'41) (OMECSA), Engr., Charge of Spec. Devel., I. C. Div., Naval Research Lab., Washington 20, D.C.; *for mail,* 4327 Beverly Dr., Dallas 5, Tex.
Shuart, Arthur Clinton ('31; '35) (CMYN), Asst. Ch. Engr., Servel, Inc., 119 Morton, Evansville 20; *for mail,* 3318 E. Blackford St., Evansville 14, Ind.
Shuckhart, Josiah B. ('33) (EF), 5209—3rd Ave., Brooklyn, N.Y.
Shuff, Evans Layton ('21; '27) (FSKHZ), Sales Engr., Evans L. Shuff & Associates, Rm. 303, 5 Ivy Bldg., Atlanta 3, Ga.
Shuff, William Edward (J'41) (CKM), Tech. Sgt., Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.; *for mail,* 2621 Stuart Ave., Richmond 20, Va.
Shufflebarger, Frederick N., Jr. ('41) (MRS), Officer, U.S.N.R., Bur. of Ships, Navy Dept.; *for mail,* 323 Florida Ave., Portsmouth, Va.
Shuh, Lewis Matthew (J'42) (NBCGDO), 1021—6th Ave., De Witt, Iowa.
Shull, Dean Ellsworth, Jr. (J'45) (FJS), Ensign, U.S.N., U.S.S. *Bole,* DD-755, F.P.O., San Francisco, Calif.; *for mail,* 714 Blythe Ave., Drexel Hill, Pa.
Shull, Donald Way ('37; '45) (CM), Project Engr., Chefford Master Mfg. Co. Inc.; *for mail,* 1103 Chef-field Dr., Fairfield, Ill.
Shulman, Martin (J'45) (MCJDB), Time Study Engr., Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10; *for mail,* 11327 Bellflower Rd., Cleveland 6, Ohio.
Shulters, Earl Stanley ('40) (CDGSKO), Tech. Asst. to Dir. of Constr., U.S. Maritime Comm., 14th & Franklin, Oakland 12; *for mail,* Box 734, Oakland 4, Calif.
Shultz, Emmett O. ('29; '35) (OSEFH), Pfeifer & Shultz, 702 Wesley Temple Bldg., Minneapolis 4, Minn.
Shumaker, Clifford H. ('38; '41) (CMSG), Assoc. Prof. Mech. Engrg., So. Methodist Univ., Dallas 5, Tex.
Shuman, Edward Steven (J'39) (CMLK), Plant Mgr., Union Asbestos & Rubber Co., 88 E. 11th St., Paterson; *for mail,* 406 Prospect Ave., Hackensack, N.J.
Shunfenthal, Benjamin (J'46), 1802 Crotona Ave., Bronx 57, N.Y.
Shure, William Henry N. (J'41) (E), 217 Hawthorne Rd., Baltimore 10, Md.
Sibbitt, Wilmer Lawrence (J'42) (EHKLSZ), Sr. Chem. Engr., Phillips Petroleum Co.; *for mail,* 422 Bartlesville, Okla.
Sibert, William B. (J'44) (CFS), U.S. Merchant Mar., 202 S. Randolph Ave., Elkins, W.Va.
Sibler, Charles J. ('44) (FLSW), Supvr. of Power, W.Va. Pulp & Paper Co., 230 Park Ave., New York 17, N.Y.
Sibley, Edward W. ('18; '21), Mech. Engr., United Fruit Co., 1 Federal St., Boston; *for mail,* 18 Eliot Rd., Lexington, Mass.
Sibley, Robert ('12; F'36) (FHS), *Vice-President,* '21-'23; Exec. Mgr., Univ. of Calif. Alumni Assn., 301 Stephens Union, Berkeley 4, Calif.
Sibson, Horace E. ('04; '19) (CKL), V.P., Gen. Sales Mgr., Cochrane Corp., 17th St., below Allegheny Ave., Philadelphia 32; *for mail,* 403 Mayfair House, Johnson St. & Lincoln Dr., Philadelphia 44, Pa.
Sibson, William Abbott ('44) (JSFOKZ), Mech. Engr., Head, Inspc. Div., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail,* 38 N. Childs St., Woodbury, N.J.
Sich, Edward Joseph (J'46) (BHO), Engrg. Technician, Oliver Corp., Cleveland; *for mail,* 3214 E. 132nd St., Cleveland 20, Ohio.
Sicka, Louis T. ('12), Gen. Mgr., St. Joseph Lead Co.; *for mail,* Box 231, Bonne Terre, Mo.
Sickles, Eugene Charles ('96; '04) (AHRY), Pres., Phillips Cooling Tower Co., 114 Liberty St., New York 6, N.Y.; *for mail,* 256 N. 11th St., Newark 7, N.J.
Sidler, Edward H. ('26) (CKL), c/o F. Kuhni, Caixa Postal 88, Recife (Pernambuco), Brazil, S.A.
Sidler, Paul Robert ('33; '35) (SERKHF), Pres., Brown, Boveri Corp. (affiliate of Brown, Boveri & Co., Ltd., Baden, Switzerland), 19 Rector St., New York 6, N.Y.
Sidney, William Edwin ('28) (BDHN), Prin. Engr., Chief, Constr. Div., Pittsburgh Dist., U.S. Engr. Office, U.S.A., New Fed. Bldg., Pittsburgh; *for mail,* 31 Oregon Ave., Pittsburgh 5, Pa.
Sieben, Clarence Mathias ('41; '43) (SEFLK), Ch. Engr., Sieben's Brewery Co., 1470 Larrabee St., Chicago 10, Ill.
Sieber, Janez Jack ('42), Research Engr., Broderick & Bascom Rope Co., St. Louis; *for mail,* 7278 Maryland Ave., University City, Mo.
Sieber, William John ('40) (MS), Engr., M. W. Kellogg Co., 225 Broadway, New York 7, N.Y.
Siebert, Virgil William (J'35) (CDZH), Plant Engr., Metal & Thermit Corp., 415 E. 151st St., East Chicago; *for mail,* Box 201, Ogden Dunes, Gary, Ind.
Siebold, Harrison Niles (J'40) (ACM), Capt., Co. D, 22nd QM. Regt., A.P.O. 782, c/o Postmaster, New York, N.Y.; *for mail,* 192 Park St., Carbondale, Pa.
Siecke, Kurt H. ('39) (FKS), 5224 S.W. Dosch Rd., Portland 1, Ore.
Siefert, Charles George ('45) (CHYJB), Mgr., Die Casting Div., Cleveland Hardware & Forging Co., 4518 Lakeside Ave., Cleveland 5; *for mail,* 2045 Staunton Rd., Cleveland Heights 18, Ohio.
Siefert, George Christian ('45) (FSROEZ), Mgr., Mexico City Office, Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* Sierra Nevada 425, Mexico, D.F., Mex.
Siefert, Raymond J. (J'40) (SLJB), 1088 Keith Ave., Berkeley 8, Calif.
Sieffert, Joseph Edward (J'43) (NOBJMS), Ensign, U.S.N.R., Ord. Specialist, Office of Port Dir., U.S.N., c/o Sun Shipbldg. & Dry Dock Co., Chester, Pa.
Siegel, Carl L. ('37; '40) (BCKLSO), Ch. Engr., Sharples Chemicals, Inc., Wyandotte, Mich.
Siegel, Geoffrey Judah (J'43), Mech. Engr., Mech. Equip. Design, Austin Co., 16112 Euclid Ave., East Cleveland; *for mail,* 3361 Euclid Heights Blvd., Cleveland Heights, Ohio.
Siegel, Herman Jonas (J'38) (NMZJHB), Ord. Engr., P-4, Ord. Design Sec., N.Y. Naval Shipyard, Brooklyn 1; *for mail,* 945 West End Ave., New York 25, N.Y.
Siegel, Leonard (J'42) (EK), Engr. Officer, U.S.N.; *for mail,* 321—67th St., Arverne, N.Y.
Siegerist, Walter ('19; '26) (MJNCB), Pres., Treas., Ch. Engr., Dir., Medart Co., 100 Potomac St., St. Louis 18; *for mail,* 7355 Westmoreland Dr., St. Louis 5, Mo.
Siegerist, Walter Laurenz (J'46) (CMZ), Ensign, Navy 115, Box 44, F.P.O., New York, N.Y.; *for mail,* 7355 Westmoreland Dr., St. Louis 5, Mo.
Siegesmund, John C. ('24; '30) (KDLHBJ), Dir., Engrg. Div., Eli Lilly & Co., 740 S. Alabama St., Indianapolis 6; *for mail,* 61 Campbell Ave., Indianapolis 1, Ind.
Siegler, Newman (J'42), 315 W. 19th St., New York, N.Y.
Siemers, Ralph Albert John (J'42) (ZNBAOK), Engrg. Lab. Technician, Cadillac Motor Car Div., Gen. Motors Corp., 2860 Clark St., Detroit 32; *for mail,* 2051 W. Grand Blvd., Detroit 8, Mich.
Siemon, Karl Otto (J'38) (CL), Engr., Bakelite Corp., River Rd., Bound Brook; *for mail,* 423 Lenox Ave., Westfield, N.J.
Siess, Edward O. ('31; '35) (GACMHJ), Engr., Duplex Ptg. Press Co., 500—5th Ave., New York; *for mail,* 88-17 Union Turnpike, Glendale, N.Y.
Siess, Leo E. (J'40), 1st Lt., Army Air Forces, Weight & Balance Officer, Hdq., O.C.A.S.C., Maint., Tinker Field, Oklahoma City, Okla.
Sietsma, Stuart John (J'35) (SEAD), Capt., Ord. Dept., Army Air Forces, Air Serv. Comd., Patterson Field, Ohio; *for mail,* R.R. 3, Lafayette, Ind.
Sievers, Ernest J. J. ('10; '23; '35) (BCDS), Draftsman, Federal Shipbldg. & Dry Dock Co., Lincoln Highway, Kearny; *for mail,* 11 Abbett St., Jersey City 7, N.J.
Sieweck, Carl Albert ('31) (CMBZN), 4915 Broadway, New York 34, N.Y.
Sigmon, Arvil Chester ('25; '36) (CDFMSY), Ch. Plant Engr., Inland Rubber Corp., Wentworth Ave. & 27th St.; *for mail,* 1035 W. North Shore Ave., Chicago, Ill.
Sigmund, Herbert Allen (J'44) (CLS), R.F.D. 1, Newark, Del.
Sigmund, Miroslav ('45), Sigmund Pumps Ltd., Kingsway Team Valley Trading Estate, Gateshead 11, Durham, England.
Signarowitz, Ferdinand J. ('15) (ABH), Hyd. Engr., Pelton Water Wheel Co., 2929—19th St., San Francisco; *for mail,* 715 San Mateo Dr., San Mateo, Calif.
Signorelli, Alfred A. (J'42) (CNML), Propr., Design Serv. Co., 3 William St., Newark 2, N.J.
Signoret, Albert John ('40) (BMT), Engr., Munitions Div., Am. Safety Razor Corp., 315 Jay St., Brooklyn 1, N.Y.
Sigworth, Robert Y. ('20; '26) (CFS), Supvr. of Utilities, Pa. State College; *for mail,* 225 S. Atherton St., State College, Pa.
Sihler, John H. (J'31) (BCMHZR), Prin. Ord. Engr., Bur. of Ord., Navy Dept., Washington 25; *for mail,* 4429 Fessenden St., N.W., Washington 16, D.C.
Sikes, John M. ('37; '41) (RSB), Supt. M.P., Office of Pres., Savannah & Atlanta Ry. Co., Savannah, Ga.
Sikorsky, Igor Ivan (F'41), *Worcester Reed Warner Medallist,* '43; Engrg. Mgr., Sikorsky Aircraft Div., United Aircraft Corp., South Ave., Bridgeport 1, Conn.
Sikosek, Ferdinand John (J'33) (CML), 2409—6th St., Coytesville, N.J.
Silberger, Marvin E. ('32) (DCBNJH), Ch. Engr., Am. Chain & Cable Co. Inc., Charles & Princess Sts.; *for mail,* 1531—3rd Ave., York, Pa.
Silberman, Morris (J'38) (BJS), 1449 Bella Vista Dr., Dallas 18, Tex.
Silberstein, Morton Stanley (J'44) (BZAN), 417 Graduate House, Mass. Inst. of Tech., Cambridge 39, Mass.
Siler, William Augustus ('45) (NBLMYJ), Engr., Delco-Remy Div., Gen. Motors Corp., Columbus Ave., Anderson, Ind.
Silha, Henry William (J'46) (JLM), Asst. Prof. Mech. Engrg., Univ. of Idaho; *for mail,* 908 W. 6th St., Moscow, Idaho.
Sill, Francis J. ('40) (SFOWE), Partner, Sill & Warrington, 79 Milk St., Boston 9, Mass.
Sillcox, Lewis Ketcham ('16; '18; F'43) (FJRS), *A.S.M.E. Medallist,* '43; 1st V.P., N.Y. Air Brake Co., Starbuck Ave., Watertown, N.Y.
Sills, Titus Odell ('36; '45) (MST), Engr., Cannon Mills Co.; *for mail,* 467 Harris St., Concord, N.C.
Silver, Francis, 5th (J'42) (DKL), Cpl., Ord. Dept., U.S.A., Camp Santa Anita, Arcadia, Calif.; *for mail,* 501 S. Queen St., Martinsburg, W.Va.
Silver, Max ('41) (BM), Southampton, N.Y.
Silverman, Harry Thomas ('41; '45) (CMLJDO), Asst. to Supt. of Shops, Youngstown Sheet & Tube Co., East Chicago; *for mail,* 207 Taft St., Gary, Ind.
Silverman, Leslie (J'40) (BDHKLZ), Asst. Prof. Indus. Hygiene, Sch. of Pub. Health, Harvard Univ., 55 Shattuck St., Boston 15, Mass.
Silverman, Sol (J'45) (AYJMCO), Project Engr., Robinson Aviation, Inc., Teterboro Air Terminal, Teterboro, N.J.; *for mail,* 960 Grand Concourse, Bronx 56, N.Y.
Simeon, Charles John ('13) (MCO), Univ. Club, Providence 3, R.I.
Simeone, Vincent (J'40) (EYKMSH), Assoc. Engr., Naval Shipyard, Boston; *for mail,* 37 Prescott St., West Medford 55, Mass.
Simington, Robert Evans (J'42) (ZMNH), Product Engr., Sperry Gyroscope Co., Manhattan Bridge Plaza, Brooklyn; *for mail,* 54 Grant Ave., Brooklyn 8, N.Y.
Simins, Anicod (J'41) (EKS), Steam Design Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia, Pa.; *for mail,* 132 Irving Ave., Brooklyn 27, N.Y.
Simmang, Clifford M. (J'41) (SEKBNA), Asst. Prof., Mech. Engrg. Dept., Agric. & Mech. College of Tex., College Sta., Tex.
Simmons, Cecil Earl (J'43) (BCMNOW), Dynamometer Engr., Gen. Motors Corp. of Can.; *for mail,* Box 96, Oshawa, Ont., Can.
Simmons, Charles H. (J'40) (MCZ), Pur. Agt., Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill.
Simmons, Edward E., Jr. (J'40) (ZBJKN), Research Engr., Rheem Mfg. Co., Richmond; *for mail,* 455 S. Oakland Ave., Pasadena 5, Calif.
Simon, Arthur ('12) (OMDB), Engr., Pat. Consultant, 735 N. Water St., Milwaukee 2, Wis.
Simon, Harrison J. (J'43) (CSDMO), Maint. Foreman, J. D. Adams Mfg. Co., 217 S. Belmont Ave., Indianapolis 6; *for mail,* Apt. 3, 2063 N. Meridan St., Indianapolis 2, Ind.
Simon, Samuel S. (J'39) (FHKS), Mech. Engr., Dept. of Pub. Works, City of New York, Rm. 816, 125 Worth St., New York, N.Y.
Simon, Walter Richard (J'42) (SEFKHB), Asst. Engr., Ballistic Research Lab., Aberdeen Proving Ground; *for mail,* 320 Lyndhurst St., Baltimore 29, Md.
Simons, Claude Wicks (J'44) (JKNR), Lt. (j.g.), U.S.N.R., Asst. Communications Officer, U.S.S. *Cabot,* Naval Base, Philadelphia, Pa.; *for mail,* 466—8th St., Manistee, Mich.
Simons, Eugene Maurice (J'41) (NBJZ), Research Engr., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Simons, Morris ('41) (JZ), Pres., Union Wire Die Corp., 71 W. 45th St., New York, N.Y.
Simons, Radcliffe Mills ('41) (ABC), Asst. to Sales Mgr., Adam Scheidt Brewing Co., Norristown; *for mail,* 4642 Pilling St., Philadelphia 24, Pa.
Simonson, Howard Elmer (J'36) (EKZH), Design Engr., Phillips Petroleum Co.; *for mail,* 1520 Jennings St., Bartlesville, Okla.
Simonson, J. Ward (J'39) (LHDCMZ), Asst. to Plant Supt., Lucidol Corp., 1740 Military Rd., Buffalo 5, N.Y.
Simpers, Robert Sanders (J'43) (CFKS), Engr., Works Engrg. Div., E. I. du Pont de Nemours & Co.; *for mail,* 4006 Forest Hill Ave., Richmond, Va.
Simpson, Arthur M. ('43), Factory Mgr., Kawneer Co., Niles, Mich.
Simpson, Colin Campbell ('12; '21) (CSDEFK), Asst. V.P., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Simpson, George Buchan ('22) (DCBS), Mgr., B.C. Conveying Mchy., Ltd., 123 W. Pender St., Vancouver, B.C., Can.
Simpson, Graham E. (J'37) (BKLCDS), Div. Supt., Linde Air Products Co., 114 Sansome St., San Francisco 4, Calif.
Simpson, Henry A. ('37) (BDMNT), Cons. Engr., Secy., Treas., Mch. & Tool Designing Co., 1011 Chestnut St., Philadelphia 7, Pa.
Simpson, James Henry ('45) (FLST), Plant Engr., Fair Lawn Finishing Co., Fair Lawn; *for mail,* 170 Lee Ave., Haledon 2, N.J.
Simpson, James Henry, Jr. (J'44) (NHJSYF), Mech. Engr., Goodyear Synthetic Rubber Corp., Houston; *for mail,* 3209 Charleston, Houston 4, Tex.
Simpson, James W. (J'45) (ASNBK), Mech. Engr., Fredric Flader, Inc., 775 Main St., Buffalo 3, N.Y.
Simpson, Jesse Ward, Jr. (J'42), Capt., Army Air Forces, 322nd Supply Sqd., 93rd ADG, A.P.O. 942, c/o Postmaster, Seattle, Wash.
Simpson, Maurice A. (J'38) (ACH), 3116 Darrell, Ft. Worth, Tex.

Simpson, Richard W. (J'37) (BLM), Inspr., Scovill Mfg. Co.; *for mail,* 9 Sands St., Waterbury, Conn.
Simpson, Robert Henry (J'43) (KFEDCM), Lt. (j.g.), U.S.N.R., Asst. Ship Supt., Naval Shipyard, Philadelphia; *for mail,* 4017 Bonsall Ave., Drexel Hill, Pa.
Simpson, Robert J. O. ('06), Cons. Engr., L. S. Starrett Co.; *for mail,* 129 Union St., Athol, Mass.
Simpson, Robert Tate (J'43) (SEKJC), Comdr., U.S.N.R., Research Sub-Sec., Turbine & Gear Sec., Bur. of Ships, Navy Dept., Washington 25, D.C.
Simpson, Roswell H. ('15; '27) (NMHBJY), Ch. Designer, Joseph Dixon Crucible Co., Wayne & Monmouth Sts., Jersey City 3, N.J.; *for mail,* 63 Burbank Ave., New Dorp, Staten Island 6, N.Y.
Simpson, William Klein ('09; '25) (CJMLDZ), 21 Sands St., Waterbury 30, Conn.
Simpson, Wm. M. (J'38) (FSW), Am. Creosoting Co., Inc., Louisville, Ky.
Sims, Ellis M. ('38; '42) (ESFAZ), Prof. Mech. Engrg., Univ. of Okla., Norman, Okla.
Simson, Jerome ('38) (SEHJLR), Project Engrg. Dept., Kellex Corp., 233 Broadway, New York, N.Y.
Sinclair, Edward Leslie ('38; '44) (KBSLZC), Design Engr., Socony-Vacuum Oil Co., Inc., Rm. 1407, 154 Nassau St., New York 7, N.Y.
Sinclair, Harold ('39) (HRSCKA), Managing Dir., Hyd. Coupling & Engrg. Co. Ltd., Fluidrive Works, Worton Rd., Isleworth, Middlesex, England.
Sinclair, Leo P., Jr. ('34; '44) (JMCDLN), Lt. Comdr., U.S.N.R., Prod. Div., Bur. of Ord., Navy Dept., Washington 25, D.C.; *for mail,* 4628—36th St. S., Arlington, Va.
Singer, Ferdinand L. ('28; '39) (BNHAJY), Asst. Prof. Engrg. Mechanics, N.Y. Univ., Box 187, Univ. Heights, New York 53, N.Y.
Singer, Joseph Harold (J'42), Sales Engr., Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.
Singer, Maurice Abraham ('24; '35) (MJCWYZ), Pres., Gen. Mgr., Doall Philadelphia Co., 3107 N. Broad St., Philadelphia 32; *for mail,* 504 W. Clapier St., Philadelphia 44, Pa.
Singer, Sidney Charles ('09; '16; '21) (CESF), No. Div. Mgr., So. Calif. Gas Co., 126 N. Maryland Ave., Glendale 6, Calif.
Singh, Jagir (J'34) (CFS), Grewal Industries, Chheharta, Amritsar, India.
Singh, Nand ('16; '18; '23), Cons. Indus. Engr., Phillaur, Jullundur, Punjab, India.
Singleton, John C., Jr. (J'39) (FKS), Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* R.D. 2, Plainfield, N.J.
Singleton, Philip Arthur (J'44) (CYTLS), Asst. to Pres., Monsanto Chem. Co., Shoreham Bldg., Washington 5, D.C.
Singleton, Walter Turner, Jr. (J'46), 1001 Prospect, El Paso, Tex.
Sinha, J. M. ('28; '32; '35), 16 Hill View Rd., Jamshedpur, India.
Sinica, John, Jr. (J'41) (FKS), Ensign, U.S.N.R., A.T.B., Camp Bradford, Norfolk, Va.
Sink, Russell S. ('45) (M), Product Engr., Auburn Mfg. Co., Auburn, Ind.
Sinton, John Joseph (J'40) (ACM), Supvr., No. 2 Time Study Dept., Plant 2, Fleetwings Div., Kaiser Cargo, Inc.; *for mail,* 20 Schumacher Dr., Bristol, Pa.
Sinton, Walter (J'42) (ASHJRB), Design Engr., Steam Div., Westinghouse Elec. Corp., Lester; *for mail,* Apt. 112, 1700 S. 60th St., Philadelphia 42, Pa.
Sintz, Claude ('17) (AEM), Pres., Claude Sintz, Inc., 1940 Stanley Ave., Detroit, Mich.
Sipin, Anatole Joshua (J'43) (ACMHZ), Aviation Electronics Technician's Mate, 2nd Class, U.S.N.R., Box 50-F, Naval Air Sta., Grosse Ile, Mich.; *for mail,* 135 Eastern Pkwy., Brooklyn, N.Y.
Sipko, Michael (J'45) (ACJ), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 9710 Anderson Ave., Cleveland 5, Ohio.
Sipp, Edmund F. (J'42) (E), 3755 N. Nora Ave., Chicago 34, Ill.
Sir, Walter W. ('30) (SKECJ), Asst. Ch. Engr., Commonwealth Edison Co., 3501 S. Pulaski Rd., Chicago 23, Ill.
Sirc, Joseph Howard (J'42) (ABC), Aerodynamicist, Goodyear Aircraft Corp., 1210 Massillon Rd., Akron 15; *for mail,* Y.M.C.A., 80 W. Center St., Akron 8, Ohio.
Sirotkin, George Von Bimman (J'41) (MJC), Ensign, U.S.N., LST-275, F.P.O., San Francisco, Calif.; *for mail,* 7325 W. Lloyd St., Wauwatosa 13, Wis.
Sirrine, Joseph Emory ('21) (HMOST), Sr. Partner, Exec. Head., J. E. Sirrine & Co., 215 S. Main St., Greenville, S.C.
Sisson, Edmond Francis ('43) (CDKL), V.P., Engrg., Newport Industries, Inc., Box 911; *for mail,* 700 W. Moreno St., Pensacola, Fla.
Sitarski, Alfred Walter (J'43) (CES), Lt. (j.g.), U.S.N.R., Naval Shipyard Annex, Maspeth, L.I., N.Y.; *for mail,* 521 Maple Ave., Linden, N.J.
Sitek, Alexander Stanislaw (J'43) (BJSCKN), 6 Oak St., Peabody, Mass.
Sites, Benj. L. ('38), 421 S. Sunnyside Ave., Elmhurst, Ill.

Sizer, Harold S. ('34; '39) (BJM), Design Analyst, Brown & Sharpe Mfg. Co., Providence; *for mail,* 77 Don Ave., Rumford, R.I.
Sizer, William D. ('22; '35) (CHGKN), Exec. Engr., Worthington Pump & Mchy. Corp., Harrison, N.J.
Skabo, Hans H. ('27; '30) (BCS), Maj., Corps of Engrs., Office of Base Utilities Officer, Langley Field, Va.
Skaggs, Henry C., Jr. (J'36), 711—1st Ave., Montgomery, W.Va.
Skaistis, Stanley Joseph (J'42), Ensign, U.S.N.R.; 4824 Addison St., Chicago 41, Ill.
Skalwold, Robert Norman (J'42), Inter-Haven Ave., Plainfield, N.J.
Skarbek, Henry F. J. ('38; '41) (ACMBJN), Admin. Engr., Ch. Draftsman, Breeze Corporations, Inc., 41 S. 6th St., Newark 7; *for mail,* 2038 Pleasant Pkwy., Union, N.J.
Skaredoff, Nikolai N. (J'39) (EHMS), Ch. Engr., Fabric Laundry & Dry Cleaning Mch. Corp., 192 Banker St., Brooklyn 22, N.Y.
Skawden, Odd Johannessen ('44) (NJCM), Ch. Engr., Sutton Engrg. Co.; *for mail,* 242 W. High St., Bellefonte, Pa.
Skees, Leonard Elisha ('45) (LNDYBC), Mech. Design Engr., Armstrong Tire & Rubber Co., Natchez, Miss.
Skene, E. Matthew (J'41) (ABGNR), Project Engr., Sperry Gyroscope Co., Inc., Great Neck; *for mail,* 18 Webb Ave., Hempstead, L.I., N.Y.
Sketchley, John Alfred (J'42) (JWL), Safety Engr., State Compensation Ins. Fund, 450 McAllister St., San Francisco 1; *for mail,* 3418 Kempton Ave., Oakland 11, Calif.
Skidmore, Benjamin, Jr. ('24) (HCMDJZ), Gen. Mgr., Skidmore Corp., Box 809, St. Joseph, Mich.
Skidmore, George Bernard (J'43) (NGCL), Engr. Machinist, Skidmore & Molitor, 75 Cliff St., New York 7, N.Y.
Skillman, Edward ('43) (BCMNWZ), Engr., J. B. Ehrsam & Sons Mfg. Co., Enterprise; *for mail,* Tribune, Kan.
Skillman, Willett Runyon (J'42) (AMDC), Lt. (j.g.), U.S.N.R.; 189 Hollister Dr., East Hartford 8, Conn.
Skilton, Harry Ingersoll ('25) (EJS), Pres., Treas., Cia. Importadora Skilton, S. A., Tejadillo 7, Havana, Cuba.
Skinner, Charles K. ('43) (CJM), Asst. Mgr., Pur. Div., Gen. Elec. Co., 1285 Boston St., Bridgeport, Conn.
Skinner, Halcyon N. ('13; '26) (SFTCLO), Plant Engr., Alex. Smith & Sons Carpet Co., Saw Mill River Rd., Yonkers 1, N.Y.
Skinner, James Chase ('33; '45) (ACJMTZ), V.P., Treas., Thomas & Skinner Steel Products Co., 1120 E. 23rd St., Indianapolis, Ind.
Skinner, James Dudley ('15) (SHB), Pres., Fuller & Co., Inc., 177 State St., Bridgeport 3, Conn.
Skinner, Sherrod Emerson ('21; '31) (ACDM), V.P., Gen. Motors Corp., Gen. Mgr., Oldsmobile Div., Lansing, Mich.
Skinner, William James (J'42) (ABMCO), Instr., Mech. Engrg. Dept., 100 Mines Bldg., Ore. State College, Corvallis, Ore.
Sklovsky, Max ('22) (BEM), Ch. Engr., Deere & Co., Moline, Ill.
Skog, Ludwig ('35) (SKEFJC), Partner, Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Skog, Ludwig, Jr. (J'39) (SOKF), Mech. Engr., Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Skoglund, Victor John ('36; '45) (KOALHB), 4939 Canterbury Dr., San Diego 4, Calif.
Skonberg, Elmer Alexander ('45) (WMCDSB), Ch. Engr., Furniture Div., Mengel Co., Louisville 1; *for mail,* 1705 Clayton Rd., Louisville 5, Ky.
Skoog, R. W. (J'37) (DEYZ), Ch. Engr., United Carbon Co., Borger, Tex.
Skortz, A. C. (J'34) (BCH), Lt., U.S. Navy; Scheller, Ill.
Skrotzki, Bernhardt G. A. ('43) (SFKHEB), Asst. Editor, McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18; *for mail,* 99 Connecticut Ave., Freeport, N.Y.
Skubik, Emil Bernard (J'42) (KBSMA), Asst. Maint. Engr., Owens-Corning Fiberglas Corp., Case Ave.; *for mail,* 364 Eddy St., Newark, Ohio.
Skultety, Stephen Ladislaus ('42) (BRS), Supvr., Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport; *for mail,* 317 Riverside Dr., Fairfield, Conn.
Skutt, Harold Asher (J'43) (AC), Morton, N.Y.
Skwarek, Frank John (J'43) (NZMCBA), Ch. Engr., Doelcam, Inc., 56 Elmwood St., Newton 58; *for mail,* R.F.D., South Bedford St., Burlington, Mass.
Slabaugh, James Evans (J'46), Ensign, U.S.N.; 504 Front St., Syracuse, Ind.
Slade, Frank L. ('23; '24; '35) (CJM), Mgr., Spec. Apparatus Div., Century Elec. Co., 1806 Pine St., St. Louis; *for mail,* 476 W. Lockwood Ave., Webster Groves 19, Mo.
Slajer, John, Jr. (J'42), 2nd Asst. Engr., U.S. Merchant Mar.; Apt. 11, 1528 Broadway, Galveston, Tex.
Slater, Charles A., Jr. (J'41) (MAC), Design Engr., Mch. Gp., Glenn L. Martin Co., Baltimore 3; *for mail,* 14 Fir Dr., Stansbury Estates, Baltimore 20, Md.

Slater, Frederick ('42) (BCM), Organizer, Natl. Assn. of Mar. Engrs., 319 Pender St. W.; *for mail,* 1332 Victoria Dr., Vancouver, B.C., Can.
Slater, Raymond E. O. ('44) (COMD), Asst. Mgr., Ernst & Ernst, 231 S. La Salle St., Chicago 4, Ill.
Slaughter, Herbert Herman, Jr. (J'45), Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 1220 Forest St., Inglewood, Calif.
Slauson, Harold Whiting ('08; '23) (FY), Mech. Engr., Box 29, Scarsdale, N.Y.
Slavenak, Adam Joseph (J'41) (EFHMNS), Asst. Ch. Draftsman, De Laval Steam Turbine Co., Nottingham Way, Trenton 2; *for mail,* 21 Samdin Blvd., Trenton 10, N.J.
Slavin, Ernest Francis (J'42) (LBESFN), Lub. Engr., Ludlow Mfg. & Sales Co., Ludlow, Mass.
Slaymaker, Philip Kuhns ('28) (NMR), Prof. Mch. Design, Engrg. College, Univ. of Neb.; *for mail,* 425 S. 26th St., Lincoln 8, Neb.
Slaymaker, Robert Ridgley ('26; '33; '35) (BMR), Dean, Jr. Div., Prof. Mch. Design, Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6, Ohio.
Slayton, Clarence Harvey, Jr. (J'44) (YCLZOD), Mech. Engr., Chem. Works, Plastics Div., Gen. Elec. Co., 1 Plastics Ave., Pittsfield; *for mail,* Schermerhorn Pkwy., Lenox, Mass.
Sleeman, Earl Carlton ('18; '30) (CJONMB), Ch. Engr., Detroit Seamless Steel Tubes Co., 6837 Wyoming Ave.; *for mail,* Terrace 2, 4616 Firestone, Dearborn, Mich.
Slegel, Louis (J'45) (KNWZYA), Research Prof. Mech. Engrg., Oregon State College, Corvallis, Ore.
Slichter, Walter I. ('02; '12; F'41) (CHS), Prof. Emeritus Elec. Engrg., Columbia Univ., New York, N.Y.
Slindee, Edward A. (J'42) (ACD), Test & Devel. Engr., Internatl. Harvester Co., 2600 W. 31st St.; *for mail,* 501 N. Central Ave., Chicago, Ill.
Slingman, Theodore Daniel, Jr. ('28; '35) (BCKLY), Mgr. Indus. Sales, Baldwin Hill Co., 501 Klagg Ave., Trenton 2, N.J.
Sloan, Robert Allan (J'41) (SJEMBK), Mech. Engr., Test Lab., Bldg. 221, Naval Shipyard, Philadelphia 12, Pa.
Sloan, Robert Sidney, Jr. ('44) (YTMCJ), Field Engr., Plastics Div., Continental Can Co., Inc., Cambridge, Ohio; *for mail,* 427 Kirk Rd., Decatur, Ga.
Sloan, Walker MacLaughlin (J'37) (CMLD), E. I. du Pont de Nemours & Co., Box 100; *for mail,* 202 Falley St., Richland, Wash.
Sloan, William Allan ('21; '30) (SEFK), Prof. Mech. Engrg., Univ. of Pa., Philadelphia 4, Pa.
Sloane, Alvin ('37) (NB), Assoc. Prof., Mass. Inst. of Tech., Cambridge 39, Mass.
Sloane, Reginald Gordon ('30; '35) (ESFAYL), Technologist, Pub. Relations, Stand. Oil Co. of N.J., 30 Rockefeller Plaza, New York 20, N.Y.; *for mail,* 19 Pingry Pl., Elizabeth 3, N.J.
Sloat, Benjamin Crosby ('18; '23) (CSMG), Retired; 107 E. 48th St., New York 17, N.Y.
Slobey, Robert Joseph (J'42) (CDG), Prod. Exec., Am. Greeting Publishers, 1300 W. 78th St., Cleveland, Ohio.
Slocomb, Charles Aubert (J'40) (CMODS), Matl. Controller, West. Pipe & Steel Co., San Pedro; *for mail,* 9011 S. Hoover St., Los Angeles 44, Calif.
Sloma, Leonard Vincent (J'42) (BANG), Pvt., Army Air Forces, Project Engr. (Tech. Aide), Air Tech. Serv. Comd., Sqd. S, 4000th Base Unit, Wright Field, Dayton, Ohio; *for mail,* 5811 Ainslie St., Chicago 30, Ill.
Slonneger, John Lee (J'43), Devel. Engr., Indus. Control, Gen. Elec. Co., Schenectady; *for mail,* Alplaus Ave., Alplaus, N.Y.
Sloss, Peter Penrose (J'42) (MRS), Ensign O-V(S); *for mail,* 105 E. 53rd St., New York, N.Y.
Slote, Irwin (J'42), 68 Garden St., Poughkeepsie, N.Y.
Smack, John Clinton ('30; '35) (JZC), Sales Engr., Sperry Products, Inc., 1505 Willow Ave., Hoboken, N.J.; *for mail,* 59 Rockinchair Rd., White Plains, N.Y.
Smalenberger, Edward Arnold, Jr. (J'45) (MCNDLB), Ch. Engr., Tanner Mfg. Co., Lawrence Park, Erie, Pa.; *for mail,* West Lake Rd., Westfield, N.Y.
Smalis, Isadore Ambrose (J'44) (CDM), Lt., U.S.N.R., 61 N. Evans, Pottstown, Pa.
Small, Malcolm Harold ('45) (CELKHJ), Ch. Engr., East Chicago Refinery, Socony-Vacuum Oil Co., Inc., East Chicago, Ind.; *for mail,* 14425 S. Michigan Ave., Chicago 27, Ill.
Small, Ramond E. (J'40) (ACK), Application Engr., Turbosuperchargers, Gen. Elec. Co., 920 Western Ave.; *for mail,* 24 Baker St., Lynn, Mass.
Small, Samuel G. (J'41) (CEL), Jerome Carpenter & Cabinetmaker Co., 1884 Jerome Ave., Bronx 53; *for mail,* 1337 Wilkins Ave., Bronx 59, N.Y.
Smallberg, Merle (J'42) (ZABHJY), Pvt., 1st Class, 7th Marines, 1st Mar. Div., Hdq. & Serv. Co., F.P.O., San Francisco, Calif.
Smart, Douglas Levering ('37) (CBS), Charge of Serv. Div., Research Dept., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 935 Westchester Rd., Grosse Pointe Park 30, Mich.
Smart, Frank ('43) (SECF), Ch. Engr., Beth Israel Hospital, Stuyvesant Pk. E., New York 3; *for mail,* 87 Ardell Rd., Bronxville 8, N.Y.

Smart, Richard Addison ('94; '00; '06) (LKZ), Field Engr., No. Am. Mfg. Co., 2910 E. 75th St., Cleveland 4, Ohio; *for mail,* 201 Slater Bldg., Worcester 8, Mass.
Smellie, Donald Gavin ('28) (CAYM), Ch. Engr., Hoover Co., North Canton; *for mail,* 357—23rd St., N.W., Canton 3, Ohio.
Smetters, Samuel Tupper ('15), Civil Engr., Rm. 700, The Sanitary Dist. of Chicago, 910 S. Michigan Ave., Chicago; *for mail,* 1550 E. 61st St., Chicago 37, Ill.
Smiley, Charles B. (J'26) (FHJKSZ), Foreman, Utilities & Tests, Irvin Works, Carnegie-Ill. Steel Corp., Dravosburg; *for mail,* 5204 Sherwood Dr., Pittsburgh 27, Pa.
Smith, Abram E. ('16), Box 221, R.F. 3, Dixon, Ill.
Smith, Addison Taylor (J'37) (MND), Partner, Compressed Air Products, 1060 Broad St., Newark 2; *for mail,* 17 Hoffman St., Maplewood, N.J.
Smith, Albert Elias (J'43) (AHM), 128 Newnan Ave., East Point, Ga.
Smith, Albert K. ('17; '23; '27) (WNYG), Mem. Tech. Staff, Bell Tel. Labs., Inc., Murray Hill; *for mail,* 3 Oxford Terrace, West Orange, N.J.
Smith, Alfred Richmond ('45) (CLS), Devel. Engr., Universal Atlas Cement Co., 135 E. 42nd St., New York 17; *for mail,* Apt. G-1, The Inn, Forest Hills, L.I., N.Y.
Smith, Allen C. ('02), 78 Thompson St., Ft. Trumbull Beach, Milford, Conn.
Smith, Alton Lincoln ('04), Prof. Emeritus Drawing & Mch. Design, Worcester Poly. Inst., Boynton, Worcester 2; *for mail,* 67 Barnard Rd., Worcester 5, Mass.
Smith, Andrew Nelson (J'43) (ASR), Test Engr., Head, Steam Turbine Devel. Test, Gen. Elec. Co., 1 River Rd.; *for mail,* 13 State St., Schenectady 5, N.Y.
Smith, Arthur D., Jr. ('29; '35) (DLS), 414 Chichester Lane, Wynnewood, Pa.
Smith, Arthur Dale (J'39) (SKFZ), Engr., Charge of Proposals, Foster Wheeler Ltd., Eastchester St.; *for mail,* 40 Ontario St. S., St. Catharines, Ont., Can.
Smith, Arthur R. ('22; F'41), Managing Engr., Turbine Generator Div., Bldg. 56, Gen. Elec. Co., Schenectady 5, N.Y.
Smith, Beauchamp E. ('24; '44) (CHNS), Pres., S. Morgan Smith Co., 500 Lincoln St., York, Pa.
Smith, Ben D. ('46), Mech. Engr., Internatl. Harvester Co., 5565 Brookville Rd., Indianapolis 1, Ind.
Smith, Bernard (J'38) (MCNBL), Ord. Engr., Supvr. Ord. Design, Naval Shipyard, San Francisco 24; *for mail,* 629 Euclid Ave., Berkeley 8, Calif.
Smith, Bill K. (J'43) (SBRMF), Field Engr., Gen. Elec. Co., 187 Spring St., Box 4659, Atlanta, Ga.
Smith, Blanchard Drake ('43) (NYMJB), Dir. of Research, Scripto Mfg. Co., 421 Houston St.; *for mail,* 18 Peachtree Circle, N.E., Atlanta, Ga.
Smith, Briton Oliver ('24) (BEFHKS), Engr., Mchy. Scien. Div., Gibbs & Cox, Inc., 21 West St., New York 6, N.Y.; *for mail,* 820—3rd Pl., Plainfield, N.J.
Smith, Bruce Mills ('37; '44) (DCMW), Indus. Engr., Rich's Inc., Atlanta, Ga.
Smith, Bruce More (J'43), Asst. Welding Engr., Am. Loco. Co., Schenectady; *for mail,* 614 Park Pl., Mechanicville, N.Y.
Smith, Carl David ('09; '21) (CDFM), Plant Engr., Fed. Glass Co., 515 E. Innis St., Columbus 7; *for mail,* R.D. 1, Groveport, Ohio.
Smith, Charles Adelbert ('26; '30; '35) (FSC BDL), Ch. Engr., Plant Mgr., Mex. Refractories Co., Box 267, Mexico, Mo.
Smith, Charles Herbert ('24; '28; '35) (CKL), *Student Award,* '24; Dist. Mgr., C. F. Braun & Co., 1000 S. Fremont St., Alhambra; *for mail,* 29 Hacienda Dr., Arcadia, Calif.
Smith, Charles O. (J'42) Lt. (j.g.), A.R.U. Navy 3133, F.P.O., San Francisco, Calif.
Smith, Charles Oliver (J'38) (CLS), Maj., Flying Control Officer, 25th Sta. Complement Sqd., A.P.O. 558, c/o Postmaster, New York, N.Y.; *for mail,* 2720 Walnut St., Alton, Ill.
Smith, Charles Richard (A'18) (BDHNS), Supt., Am. Appraisal Co., 525 E. Michigan St.; *for mail,* 2147 N. 52nd St., Milwaukee 8, Wis.
Smith, Chetwood ('21) (KYRL), 112 Dartmouth St., Boston 16, Mass.
Smith, Daniel Thomas (J'42), Lt., U.S.N., Supvr., Shipbldg., Chicago, Ill.; *for mail,* 694 Laurel Ave., Bridgeport 4, Conn.
Smith, Donald Armstrong (J'33) (GLDC), Mgr., Brooklyn Factory, F. H. Levey Co., 222—44th St., Brooklyn 32, N.Y.
Smith, Donald Campbell (J'42), 554 Church St., Toronto, Ont., Can.
Smith, Donald Irving (J'44) (AER), 104-51—125th St., Richmond Hill, L.I., N.Y.
Smith, E. Lovell ('38) (NBJCM), V.P., Charge of Engrg., Package Mchy. Co., 132 Birnie Ave., Springfield 7; *for mail,* 148 Colony Rd., Longmeadow 6, Mass.
Smith, Earl B. ('32) (SCAK), Prof. Mech. Engrg., College of the City of N.Y., 140th St. & Amsterdam Ave., New York 31, N.Y.
Smith, Earle E. ('24; '30; '35) (JBHA), Mgr. of Sales, Natl. Tube Co., Philtower Bldg., Tulsa 3, Okla.
Smith, Eastman ('22; '27; '35) (BWNEZA), Assoc. Prof. Mech. Engrg., Univ. of Mo.; *for mail,* 1400 Rosemary Lane, Columbia, Mo.
Smith, Ed Sinclair ('23; '31) (ZBKHAS), *Junior Award,* '30; Pat. Liaison Engr., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 375 Warwick Ave., West Englewood, N.J.
Smith, Edric Brooks ('13; '35), Business Mgr., Rockefeller Inst. of Medical Research, 66th St. & York Ave., New York, N.Y.
Smith, Edward Arthur ('45) (ONDBSC), Equip. Engr., Raymond Concrete Pile Co., 140 Cedar St., New York 6, N.Y.; *for mail,* 30 Edgewood Rd., Chatham, N.J.
Smith, Edward Sherman (J'45) (YLTNBG), Ch. Engr., Plastic Coating Corp., Holyoke; *for mail,* Woodbridge Terrace, South Hadley, Mass.
Smith, Edwin Dwight ('14) (DAMS), Plant Engr., Natl. Cash Register Co., Main & K Sts., Dayton 9, Ohio.
Smith, Edwin Lewis ('42) (SKFE), Ch. Engr., Harry R. Byers, Inc., U.S. Natl. Bank Bldg., Denver; *for mail,* 723—19th St., Boulder, Colo.
Smith, Edwin R. ('21; '35) (BCM), V.P., Gen. Mgr., Seneca Falls Mch. Co., 314-40 Fall St., Seneca Falls, N.Y.
Smith, Elbridge Milton (J'44) (ABMNO), 125th Naval Constr. Bn., F.P.O., San Francisco, Calif.; *for mail,* 86 Bolton Ave., White Plains, N.Y.
Smith, Elliott Dunlap ('40) (CM), Provost, Carnegie Inst. of Tech., Schenley Pk., Pittsburgh 13, Pa.
Smith, Elliott Earl ('25) (SFRHKZ), Ch. Engr., Scranton Elec. Co., 507 Linden St., Scranton 3; *for mail,* 1516 N. Webster, Scranton 9, Pa.
Smith, Elmer ('13) (SEC), Steam Turbine Specialist, Gen. Elec. Co., 140 Federal St., Boston 1; *for mail,* 70 Pinckney St., Boston 14, Mass.
Smith, Elmer ('44) (CEFKSZ), Gen. Supt., Owensboro Mun. Utilities, Box 780; *for mail,* 116 W. 17th St., Owensboro, Ky.
Smith, Elwyn L. ('19; '25; '34), Mem. of Firm, L. C. Smith & Corona Typewriters, Inc., 701 E. Washington St., Syracuse, N.Y.
Smith, Emerson Warfield (J'44) (ANHKC), Aviation Electronics Technician's Mate, 2nd Class, U.S.N.R., Naval Auxiliary Air Sta., Chincoteague, Va.; *for mail,* 226 S. Elm St., Henderson, Ky.
Smith, Eric Hooper ('32; '38) (FSJNKC), Ch. Engr., Riley Stoker Corp., 9 Neponset St., Worcester; *for mail,* 22 Moreland St., Worcester 2, Mass.
Smith, Francis Carroway ('37; '41) (ELS), Editor-in-Chief, *Southern Power and Industry,* Grant Bldg., Atlanta 3, Ga.
Smith, Frank E. ('31) (SEJRLB), Devel. Engr., E. I. du Pont de Nemours & Co.; *for mail,* 2907 Lewiston Rd., Niagara Falls, N.Y.
Smith, Frank Fenton (J'44) (NHMCAB), Engr., Dowty Equip. (Canada) Ltd., 999 Aqueduct St., Montreal 3, Que., Can.
Smith, Frank J. ('45) (OSZFLM), Cons. Engr., Girard Trust Bldg., Philadelphia 2, Pa.
Smith, Frank Wesley ('44) (JMBN), New England Mgr., D. A. Stuart Oil Co., Ltd., 803 Park Sq. Bldg., 31 St. James Ave., Boston 16, Mass.
Smith, Fred Bradshaw (J'36) (NBCM), Asst. Engrg. Officer, Ord. Dept., U.S.A., Watervliet Arsenal, Watervliet; *for mail,* R.D. 1, Cohoes, N.Y.
Smith, Frederick Tucker (J'43) (ACBE), Lt. (j.g.), U.S.N.R., Asst. Engrg. Officer, U.S.N., Air Trans. Sqd. 1, VR-1, Naval Air Sta., Patuxent River, Md.
Smith, George Glenwood (J'38) (CBM), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14, N.Y.
Smith, George Hillier ('43) (DBKLN), Provincial Engrg. Rep., William & J. G. Greey Co., Ltd., 56 Esplanade St., Toronto 1, Ont.; *for mail,* 6184 Notre Dame DeGrace Ave., Montreal 28, Que., Can.
Smith, George W. (J'38) (CDM), 446½ Albert St., Lima, Ohio.
Smith, George Wetherall ('29; '35) (BZS), Gen. Mgr., Hagan Corp., Hagan Bldg., Pittsburgh 30; *for mail,* 1121 Wightman St., Pittsburgh, Pa.
Smith, Gerald Edward (J'37), Maj., Royal Canadian Engrs., Can. Army, C.M.F., 4th Canadian Field Park Sqd.; *for mail,* 52 Parkway Ave., Toronto 3, Ont., Can.
Smith, Gerald Leonard ('36; '45) (CMDJ), Indus. Engr., York Ice Mchy. Corp., Roosevelt Ave.; *for mail,* 1434—1st Ave., York, Pa.
Smith, Glenn Kitterman (J'42), Mech. Engr., Natl. Starch Products, Inc., 1515 S. Drover St., Indianapolis 6; *for mail,* 603 W. North St., Greenfield, Ind.
Smith, Grant F. (J'43) (EAJF), Box 131, Ririe, Idaho.
Smith, Guy P. (J'43) (LMN), Ch. Draftsman, Stupakoff Ceramic & Mfg. Co.; *for mail,* R.D. 1, Latrobe, Pa.
Smith, H. Pearson ('35; '35) (BLS), Indus. Salesman, 27 S. Main St., Windsor Locks, Conn.
Smith, H. Raymond ('16), Mech. Engr., Raymond Concrete Pile Co., 140 Cedar St., New York 6, N.Y.
Smith, Harold A. (J'41), Lt., U.S.A., Retired; 2208 Oakland Ave., Covington, Ky.
Smith, Harold Ansorge ('32; '41) (ESABFC), Asst. M.M., Stand. Oil Co. of Ind., Sugar Creek, Mo.
Smith, Harold L. ('29) (EFS), Mech. Engr., Buffalo, Niagara Elec. Corp., Elec. Bldg., Buffalo; *for mail,* Freeman Rd., Orchard Park, N.Y.
Smith, Harold Thomas (J'41) (EFJZ), Mech. Engr., Naval Engrg. Exper. Sta., Annapolis, Md.
Smith, Harry A. ('29; '35) (BKEA), Pres., Engrg. Specialties Co., Inc., 39 Cortlandt St., New York 7, N.Y.
Smith, Harry Gray ('44), Ch. Field Engr., Smith Meter Co., 5743 Smithway St., Los Angeles; *for mail,* 445 N. Oakland, Pasadena, Calif.
Smith, Harry James ('21) (BEKOS), Retired; 201 Wawona St., San Francisco 16, Calif.
Smith, Harry L. ('42) (BMT), 562 Main St., Saco, Me.
Smith, Harry White, Jr. ('44) (DKL), V.P., Gen. Mgr., Tech. Information, Tech. Adv., John Mather Lupton Co., Inc., 420 Lexington Ave., New York 17; *for mail,* 66 Milton Rd. (H-51), Rye, N.Y.
Smith, Hartley LeH. ('28) (BFK), Bd. of Trans., Rm. 510, 250 Hudson St., New York, N.Y.
Smith, Hartley Perry ('44) (CDM), Assembly Supvr., Underwood Corp., 581 Capitol Ave., Hartford 6; *for mail,* 32 Alton St., Manchester, Conn.
Smith, Henry Eugene (J'44) (NKLOHT), Engrg. Dept., Tenn. Eastman Corp., Box 511; *for mail,* 308 W. Ravine Rd., Kingsport, Tenn.
Smith, Henry R. ('30), Mar. Supt., Colonial Beacon Oil Co., Everett; *for mail,* 9 Staples St., Melrose, Mass.
Smith, Henry S. ('19) (FJ), Cons. Engr., Linde Air Products Co., 30 E. 42nd St., New York, N.Y.
Smith, Herbert James ('13; '26) (CM), Pres., Threadwell Tap & Die Co.; *for mail,* 675 Bernardston Rd., Greenfield, Mass.
Smith, Herbert Stanley (J'45) (NMOCE), Draftsman, Canadian Sumner Iron Works Ltd., 3550 E. Broadway; *for mail,* 2449 Point Grey Rd., Vancouver, B.C., Can.
Smith, Herman Elmo, Jr. (J'44) (ACEHMS), Maj., Engrg. Officer, Army Air Forces, 2719 N.W. 18th St., Oklahoma City 7, Okla.
Smith, Hiram Gordon ('30) (SKZFB), Mech. Engr., E. I. du Pont de Nemours & Co., Sta. B, Buffalo; *for mail,* 294 Brantwood Rd., Snyder, N.Y.
Smith, Howard Thomas (J'43) (ASBHKE), 3520 Central Pl., Normandy 20, Mo.
Smith, Howard Wells ('92; '18) (JBOS), Cons. Engr., Box 349, Ellwood City, Pa.
Smith, Howard William (J'41) (BCS), 2208 Oakland Ave., Covington, Ky.
Smith, Hugh Taylor, Jr. (J'37) (RMDNCZ), Process Engr., Gen. Mchy. Ord. Corp., South Charleston; *for mail,* 603 E St., South Charleston 3, W.Va.
Smith, Miss Irma M. (J'38) (MCA), Prod. Supvr., R. G. Smith Tool & Mfg. Co., 245 South St., Newark 5, N.J.
Smith, Irvine Wray ('38; '44) (NBJMKZ), Asst. Prof. Mech. Engrg., Univ. of Toronto; *for mail,* Apt. 11, 40 Hazelton Ave., Toronto 5, Ont., Can.
Smith, J. Darrell ('30) (FRS), Asst. Engr., Philadelphia & Reading Coal & Iron Co.; *for mail,* 1803 Howard Ave., Pottsville, Pa.
Smith, J. F. Downie ('24; '33; '35) (NYKHBC), Head, Engrg. Dept., Research Div., United Shoe Mchy. Corp., Balch St., Beverly, Mass.
Smith, J. Jos. (J'36) (SFEZCJ), Ch. Petty Officer, U.S.N., U.S.S. *Leon,* F.P.O., New York; *for mail,* 25-10—30th Rd., Astoria, L.I., N.Y.
Smith, Jack Theodore (J'45) (EFS), Serv. Engr., Peabody Engrg. Corp., 580—5th Ave., New York; *for mail,* 52 Mitchell Ave., Yonkers, N.Y.
Smith, James K. (J'44) (CLP), 1216 S. Corning St., Los Angeles 35, Calif.
Smith, James Uriel ('28) (BN), Retired; 2477 Virginia St., Berkeley 4, Calif.
Smith, Jesse Lloyd (J'46) (BKS), U.S. Navy; 233 Juniper St., Quakertown, Pa.
Smith, John Edward (J'46), Lt., U.S.N.R., 4216 Adams St., Kansas City, Kan.
Smith, John Frederick ('39) (B), Pat. Counsel, Compo Shoe Mchy. Corp., 150 Causeway St., Boston 14, Mass.
Smith, John Hays ('22) (OFSKLE), Cons. Engr., 527 Bakewell Bldg., Pittsburgh; *for mail,* 6521 Darlington Rd., Pittsburgh 17, Pa.
Smith, John William (J'40) (LOB), Mech. Engr., Giffels & Vallet, Inc., 1000 Marquette Bldg., Detroit; *for mail,* 16697 Lauder Ave., Detroit 27, Mich.
Smith, Joseph Emmons (J'42) (ABM), 451 W. Kirby Ave., Detroit, Mich.
Smith, Kenneth Morris (J'39) (MLDCYN), Fdy. Engr., Caterpillar Tractor Co., East Peoria; *for mail,* 203 Flora, Peoria 5, Ill.
Smith, LaRue ('39) (HLS), Pres., Smith-Monroe Co., 1910-12 S. Main St., South Bend, Ind.
Smith, Lawrence Abner (J'44) (ABHKSN), Aviation Boatswain's Mate, U.S.S. *Midway,* CVB-41, F.P.O., New York, N.Y.; *for mail,* Yeso, New Mex.
Smith, Le Roy Franklin ('44) (FKS), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.; *for mail,* 79 Lilac St., Bergenfield, N.J.
Smith, Lester Courtland ('40) (BDJN), Ch. Engr., Spencer Turbine Co., Hartford 6, Conn.

Smith, Lewis Cheyney, Jr. (J'39) (A), Serv. Carburetor Engr., Wright Aero. Corp., Paterson 8; *for mail,* 259 E. 33rd St., Paterson 4, N.J.
Smith, Lewis F. (J'41) (ABE), Lt., U.S.N.R., 113 E. Genesee St., Auburn, N.Y.
Smith, Lloyd Lyman ('20) (DEFJKS), Mgr., Lloyd L. Smith Co., 605 Plymouth Bldg., Minneapolis 2, Minn.
Smith, Louis Golden ('23; '27; '35) (CFS), Asst. to Gen. Supt., Elec. Operas., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington & Liberty Sts., Baltimore 3; *for mail,* 3508 Newland Rd., Baltimore 18, Md.
Smith, Louis H. V. (J'40), 596 Monroe Ave., Rochester, N.Y.
Smith, Malcolm H. ('18; '35) (CJL), 62 Green Valley Rd., Wallingford, Pa.
Smith, Mark E. ('34), Ch. Engr., Union Iron Works; *for mail,* 227 W. 18th St., Erie, Pa.
Smith, Marshall M. ('29; '35) (CJ), Asst. Gen. Sales Mgr., E. W. Bliss Co., 450 Amsterdam, Detroit 2, Mich.
Smith, Marvin Wadsworth ('43) (SRECJL), V.P., Charge of Engrg., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30; *for mail,* 800 Valley View Rd., Pittsburgh 16, Pa.
Smith, Morgan Garsed (J'40) (CEFKMO), Cons. Mech. Engr., 2708 San Pedro St., Austin 21, Tex.
Smith, Morris S. (J'25) (SRCMN), Mech. & Metal. Engr., J. R. Johnson & Co. Inc., 2400 Maury St., Richmond 24; *for mail,* Box 345, R.F.D. 9, Richmond, Va.
Smith, Norman Leslie ('27; '35) (DCMO), Engr., Charge of Fdy. Equip. Div., Link-Belt Co., 2045 W. Hunting Park Ave., Philadelphia 40, Pa.; *for mail,* 105 Ogden Ave., Collingswood, N.J.
Smith, Oliver Hobson ('44) (ECFS), Mgr. Gas Prod., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Smith, Paul Burton (J'40) (KHCB), 256 Burwell, Bremerton, Wash.
Smith, Paul Eugene (J'45), 1919 Glenn Ave., Columbus, Ohio.
Smith, Philip Herbert (J'44), 609 S. Los Robles, Pasadena 5, Calif.
Smith, Philip M. (J'40) (CDL), Capt., 330 Felix, Muskogee, Okla.
Smith, Prescott A. (J'35) (CJM), R.F.D. 1, Monument St., Concord, Mass.
Smith, Ralph Edward ('44) (DJBN), Design Engr., Manning, Maxwell & Moore, Muskegon; *for mail,* 553 W. Broadway, Muskegon Heights, Mich.
Smith, Ralph R. ('20; '26; '30) (YMTBNZ), Head, New Product Devel., Natl. Carbon Co., 401 Gage St.; *for mail,* 113 Imperial Ave., Bennington, Vt.
Smith, Randolph Monroe (J'35) (COZKEA), Col., Ord. Dept., U.S.A., Chief, Tool & Equip. Sub-Office, Office of Ch. of Ord., Union Guardian Bldg., Detroit 32; *for mail,* 8100 E. Jefferson Ave., Detroit 14, Mich.
Smith, Ray M. ('44) (RKJ), Ch. Mech. Engr., Union Tank Car Co., 228 N. La Salle St., Chicago 1, Ill.
Smith, Reginald C. ('26; '35) (CLMDOS), Project Engr., H. K. Ferguson Co., Hanna Bldg., Cleveland, Ohio; *for mail,* 240 Pine Lake Ave., La Porte, Ind.
Smith, Reuel Lhamon ('32; '35) (NEBH), Assoc. Prof. Mech. Engrg., Univ. of Cincinnati, Cincinnati 21, Ohio.
Smith, Richard Bayly ('44) (JMY), Rm. 802, 40 W. 40th St., New York 18, N.Y.
Smith, Richard Frazier (J'46) (JLM), Engrg. Dept., Bucyrus-Erie Co.; *for mail,* Box 147-E, S. Chicago Ave., South Milwaukee, Wis.
Smith, Richard W. (J'40) (BMJ), Pvt., U.S.A., A.P.O. 502, c/o Postmaster, San Francisco; *for mail,* 2215 Rose St., Berkeley 7, Calif.
Smith, Robert A., Jr. (J'40) (CJM), Asst. Supt. of Welding, Fairbanks, Morse & Co., Lawton Ave.; *for mail,* 1320 Emerson St., Beloit, Wis.
Smith, Robert Douglas (J'45) (EAKB), Graduate Mech. Engrg. Student, Mass. Inst. of Tech., Cambridge; *for mail,* 9 Staples St., Melrose 76, Mass.
Smith, Robert Irving (J'40) (SAK), Apprentice Seaman, U.S.N.R., Co. 4445, Bks. 412U, Naval Trng. Center, Bainbridge, Md.; *for mail,* 932 Woodlawn St., Scranton 9, Pa.
Smith, Robert Warren (J'35) (NBMO), Ch. Engr., Motor Div., Thomas B. Gibbs Div., George W. Borg Corp., Delavan, Wis.
Smith, Robert William ('14), Plant Engr., Wyandotte Chem. Corp.; *for mail,* 80 Vinewood Ave., Wyandotte, Mich.
Smith, Roger Ramsdell ('45) (WCJ), Treas., L. B. Ramsdell Co., 170 Mill St.; *for mail,* 75 Elm St., Gardner, Mass.
Smith, Ronald B. ('30; '39) (BEKSA), V.P., Charge of Engrg., Elliott Co., Jeannette; *for mail,* 617 Ridge Way, Greensburg, Pa.
Smith, Ronald Chauvin (J'42), Engr., West. Elec. Co., Inc., Central Ave., Kearny; *for mail,* 745 Clifton Ave., Clifton, N.J.
Smith, Ronald R. ('28; '35; '38) (NBLMWJ), Mech. Engr., Beloit Iron Works; *for mail,* 1510 White Ave., Beloit, Wis.
Smith, Ronaldson (J'39) (BCJGRS), Mech. Engr., Designer, Sylvania Elec. Products, Inc., 75 Sylvan St., Danvers; *for mail,* 78 Londonderry Rd., Marblehead, Mass.
Smith, Roy E. (J'36) (NBM), Mch. Designer, Armour Research Foundation, 35 W. 33rd St., Chicago 16; *for mail,* 18121 Ridgewood Ave., Lansing, Ill.
Smith, Roy Harmon ('06; '17) (CJM), Pres., Gen. Mgr., Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, Ohio.
Smith, Roy Weare (J'37) (SBHER), Asst. to Outfitting Supt., Fed. Shipbldg. & Dry Dock Co., Port Newark; *for mail,* 44 Washington Ave., West Caldwell, N.J.
Smith, Russell John ('38; '42) (ABE), Asst. Prof., Head, Dept. of Mechanics, Marquette Univ., 1515 W. Wisconsin Ave., Milwaukee 3, Wis.
Smith, S. Arnold ('44) KJLCM), Ch. Engr., Whitlock Mfg. Co., Hartford; *for mail,* 1377 Main St., Newington 11, Conn.
Smith, Samuel Harold (J'32) (BANK), Pres., Treas., Smith Tool & Engrg. Co., 900 N. Sandusky Ave., Bucyrus, Ohio.
Smith, Scott Vernon ('44) (NLEGMC), Research & Devel. Engr., Sta-Rite Corp., 2010 W. Main St., Shelbyville, Ill.
Smith, Sidney Theodore (J'41) (FKL), Htg. Engr., Sid Smith & Co., 411 W. 5th St.; *for mail,* 609 Moir, Waterloo, Iowa.
Smith, Spencer Gray (J'45) (BCY), Mech. Engr., Flintkote Co., Galvez & Poland St., New Orleans; *for mail,* Apt. E, 3658 Gentilly Blvd., New Orleans 17, La.
Smith, Stanley Allen (J'43) (ABS), Lt., U.S.A., Co. A, 132nd Engr. Combat Bn., A.P.O. 9718, c/o Postmaster, San Francisco, Calif.
Smith, Stuart Brainard ('39) (SBH), V.P., Henry Pratt Co., 2222 S. Halsted St., Chicago 8, Ill.
Smith, Sydney E. (J'37) (BJS), Mech. Engr., Boiler Div., Foster Wheeler Corp., 6 Church St., New York 6, N.Y.
Smith, Talbert E. ('24; '38) (FHS), Mem. Firm, Burford, Hall & Smith, 140 Edgewood Ave., Atlanta 3, Ga.
Smith, Theodore Hunter ('29; '37) (SFKHZB), Div. Engr., Consltd. Edison Co. of N.Y., Inc., E. 134th St. & Locust Ave., New York 54; *for mail,* 459 Siwanoy Pl., Pelham Manor 65, N.Y.
Smith, Thomas James ('30; '38) (LZEF), Field Engr., Powers Regulator Co., 2720 Greenview Ave., Chicago 14, Ill.
Smith, Thomas Jennings ('45) (NMJWBY), Dir., Mfg. & Engrg., Pearl Wick Corp., 27-50—1st St., Long Island City 2, N.Y.; *for mail,* 180 Walnut St., Montclair, N.J.
Smith, Thomas Walbert (J'41) (EKS), Watch Engr., East. State Hospital, Williamsburg, Va.
Smith, Traver James (J'40) (NJALDZ), Mech. Engr., Magnuson Engrs., 841 The Alameda, San Jose 11; *for mail,* 714 Willis Ave., San Jose, Calif.
Smith, Truman Aldrich ('39) (DBOHSM), Ch. Engr., Shore Facilities, Waterman Steamship Corp., Merchants Bank Bldg., Mobile 13; *for mail,* 52 N. Monterey St., Mobile 17, Ala.
Smith, V. Weaver ('27) (CEKLS), Exec. in Charge, Contracts & Labor Relations, Lummus Co., 420 Lexington Ave., New York 17, N.Y.
Smith, Vernon ('28) (BFKZSM), Managing Dir., John Thompson (Australia) Pty. Ltd., 312 Flinders St., Melbourne, Australia.
Smith, Victor Jacob ('40) (JMNY), Mech. Engr., Natl. Tool Co., 11200 Madison Ave., Cleveland 2; *for mail,* 301 Groveland Club Dr., Cleveland 10, Ohio.
Smith, W. Manning ('36) (FS), Lt. Comdr., U.S.N.R., Design Officer, Hull Inspc. Officer, Office of Supvr. of Shipbldg., Bethlehem-Hingham Shipyard, Inc.; *for mail,* 21 Simmons Rd., Hingham, Mass.
Smith, Walter ('30) (RNFBJS), West. Mgr., Pilliod Co., Rm. 453, 332 S. Michigan Ave., Chicago 4, Ill.
Smith, Walter Gold (J'41) (BAHZ), X-Ray Div., Westinghouse Elec. Corp., 2519 Wilkens Ave., Baltimore 3, Md.
Smith, Walter H. ('40) (CSO), Ch. Engr., T. Eaton Co. Ltd., 190 Yonge St., Toronto 2, Ont., Can.
Smith, Warren H. ('28; '35; '35), 52 Piquette Ave., Detroit 2, Mich.
Smith, Wayne Everett ('28) (EF), 109 Kings Highway W., Haddonfield, N.J.
Smith, William Earhart ('08; '18) (SERF), Ch. Boiler Inspr., Exper. Sta., Hawaiian Sugar Planters Assn., Honolulu 4, T.H.
Smith, William Henry (J'37) (ECSR), Sales Engr., Caterpillar Tractor Co., Peoria; *for mail,* 1019 Elmhurst Ave., Peoria 4, Ill.
Smith, William James (J'44) (FSC), Mch. Devel. Engr., E. I. du Pont de Nemours & Co., Wilmington; *for mail,* Box 212, Delaware City, Del.
Smith, Wm. P. ('18; '35), Mgr., Indus Dept., Kelso Wagner Co., 134 W. 2nd St., Dayton 2; *for mail,* 1230 Kemper Ave., Dayton 10, Ohio.
Smith, Young Chalmers ('35) (SLHK), Propr., Young C. Smith & Co., 7310 Woodward Ave., Detroit 2, Mich.
Smithberg, Eugene Herbert (J'43), 1908 Prospect Ave., New York 57, N.Y.
Smithline, Sidney (J'40), 315 W. 25th St.. New York 1, N.Y.
Smoak, Ralph Andrew (J'42) (DNWKJ), Designer, Applied Engrg. Co., 16 S. Middleton St.; *for mail,* 16 Doyle St., Orangeburg, S.C.
Smolderen, Ferdinand Victor (J'24) (CLM), Asst. to Pres., Refined Syrups & Sugars, Inc., Foot of Vark St., Yonkers 1; *for mail,* 275 E. Sidney Ave., Mt. Vernon, N.Y.
Smolensky, Stanley Michael (J'42) (HCSAMN), Maj., Ord. Dept., U.S.A.; *for mail,* 202 Bay State Rd., Boston 15, Mass.
Smoot, Chas. H. (J'41) (SZLA), Research Engr., Republic Flow Meters Co., 2240 Diversey Ave., Chicago 47, Ill.
Smoot, Lewis E. ('30), Chmn. of Bd., Smoot Sand & Gravel Corp., 3020 K St., N.W., Washington, D.C.
Smothers, William C. (J'41) (AFE), Devel. Engr., Continental Motors Corp., 12801 E. Jefferson Ave., Detroit 14; *for mail,* 1352 Grayton Rd., Grosse Pointe 30, Mich.
Smuts, Edward Ernest (J'45) (SRCNED), Jr. Engr., Engrg. Div., Chrysler Corp., Oakland Ave.; *for mail,* 257 Colorado Ave., Detroit 3, Mich.
Smyth, Sigurd (J'45), Lt., Engrg. Officer, U.S. Navy, U.S.S. *Pocomo,* AGC-16, F.P.O., New York, N.Y.; *for mail,* 389 Columbia Ave., Cliffside Park, N.J.
Smyth, Stanley Leroy (J'44) (NJKBHM), Mech. Engr., Friden Calculating Mch. Co., 3500 Washington, San Leandro; *for mail,* 1020—106th Ave., Oakland 3, Calif.
Snashall, Newton W. (J'36) (C), 698 Duane Ave., Glen Ellyn, Ill.
Snavely, A. Bowman ('16; '21; '26; F'42) (CFS), Ch. Engr., Hershey Chocolate Corp., Hershey, Pa.
Sneath, Wm. H. ('21), Ch. Engr., Elec. Furnace Products Co., Ltd., 30 E. 42nd St., New York, N.Y.
Snedden, William Tait ('32) (JMN), Cons. Engr., 206 S. Spring St., Los Angeles 12, Calif.
Sneddon, Wayne O. (J'41) (ACH), 1028 Justin Ave., Glendale, Calif.
Sneegas, Eldon Carl (J'43) (AYBCJD), Kan. Univ.; *for mail,* R.F.D. 4, Lawrence, Kan.
Snell, Jacob Kirkwood (J'46) (JMS), Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 220 Steinmetz Home, Schenectady 4, N.Y.
Snellgrove, William Arthur, Jr. (J'39) (SLOJ), Sales Engr., Jenkins Bros., 376 Spring St., N.W., Atlanta, Ga.
Snelling, Henry Hornor ('18; '19) (MCAR), *Vice-President,* '38-'39; Sr. Mem., Snelling & Hendricks, 900 F St., N.W., Washington 4, D.C.; *for mail,* 6708—45th St., Chevy Chase 15, Md.
Sniader, David Herman (J'43) (AJK), 91 Kenview Ave., Kenmore 17, N.Y.
Snider, Arthur Melville ('19; '25; '35) (CMLJGZ), Gen. Mgr., Sunshine Waterloo Co., Ltd., Sunshine Ave.; *for mail,* 115 Allen St. W., Waterloo, Ont., Can.
Snider, Mike (J'38) (AECM), Project Engr., Eng. Devel., Wright Aero. Corp., Paterson 3; *for mail,* 331 Plaza Rd., N., Fairlawn (Radburn), N.J.
Sniffen, William Henry ('20; '35) (BEJKMS), Maint. Engr., Norwalk Co. Inc., South Norwalk; *for mail,* 16 Pershing St., Norwalk, Conn.
Snovel, Ellis Raymond, Jr. (J'41) (EFHJKS), Lafayette College; *for mail,* 24 N. 3rd St., Easton, Pa.
Snow, Raymond Muller (J'38) (AJFBNY), Aero. Engr., Naval Air Sta., Lakehurst; *for mail,* R.D. 2, 61 Syor Rd., Toms River, N.J.
Snow, Warren Sargent ('34; '43) (BCDHMN), 11 Wawecus Rd., Worcester 5, Mass.
Snowden, H. J. ('20), Stand. Steel Works Co., 627 Ry. Exch., Chicago, Ill.
Snyder, Clinton Creveling (J'41), 1708 Madison Ave., Scranton 9, Pa.
Snyder, Earl Vincent ('43) (YCM), 91 Berkeley St., Rochester 7, N.Y.
Snyder, Edwin Henry ('43) (SFAHJE), Asst. Ch. Engr., Elec. Engrg. Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Snyder, George Lawrence ('45) (NKOCMJ), Asst. Mgr. Devel. Engrg., Lukens Steel Co., Coatesville; *for mail,* 15 Uwchlan Ave., Downingtown, Pa.
Snyder, Gerald P. (J'39), Box 151, R.F.D. 1, Winters, Calif.
Snyder, Harry Eugene ('22) (CMKDYW), 206 S. 13th St., Philadelphia 7, Pa.
Snyder, Harry William (J'32) (CDJNMB), Pres., H. P. Snyder Mfg. Co., Inc., Main St., Little Falls, N. Y.
Snyder, Herbert W. ('17), 324 S. Rosedale Ave., Lima, Ohio.
Snyder, James H. ('21; '34) (NBLMK), Ch. Engr., Pfaudler Co., 89 East Ave., Rochester 4; *for mail,* 312 Roslyn St., Rochester 11, N.Y.
Snyder, Joe Dudley, Jr. (J'39) (BHNS), Tech. Draftsman, Newport News Shipbldg. & Dry Dock Co.; *for mail,* Box 72, Hilton Village Branch, 317 Palen Ave., Newport News, Va.
Snyder, John Henry (J'40) (CBMSJ), Mfg. Engr., River Works, Gen. Elec. Co., Lynn; *for mail,* 26 Beckford St., Salem, Mass.
Snyder, Leonard Leroy ('44) (NMCBHJ), Ch. Engr., Charge of Prod., Serv. Valves Co., 1449 S. San Pedro, Los Angeles 15; *for mail,* 149 N. 19th St., Montebello, Calif.
Snyder, Lewis L. ('23; '35) (BH), Ch. Engr., Snyder Engrg. Corp., 2444 E. 24th St., Los Angeles; *for mail,* 4234—4th Ave., Los Angeles 43, Calif.
Snyder, Nate Hart, Jr. (J'46), 310 E. Laurel St., San Antonio, Tex.

Snyder, Nathan William (J'41) (KBLOAH), Instr. in Mech. Engrg., Univ. of Calif.; *for mail,* 1856 Spruce St., Berkeley 4, Calif.
Snyder, Norman Stanley ('39) (BDHJS), Dist. Mgr., Engr., Link-Belt Co., 688 Ellicott Sq., Buffalo 3, N.Y.
Snyder, Richard Paul (J'38) (CAS), Mar. Engr., Design Sec., Naval Shipyard, S. Broad St., Philadelphia 12; *for mail,* 213 Tatnall Ave., Norwood, Pa.
Snyder, Seth M., Jr. ('36; '44) (FKS), 4403 Marble Hall Rd., Baltimore 18, Md.
Snyder, William E. (J'41) (EKS), 2420 Park Ave., Parkwood, Easton, Pa.
Sobel, Charles (J'44) (AJM), Asst. Engr., U.S. Engrs., 120 Wall St.; *for mail,* 264 E. 2nd St., New York, N.Y.
Sobol, Richard Philip (J'39) (BMR), Tool Designer, Electro-Motive Corp., La Grange; *for mail,* 1846 S. 57th Court, Cicero, Ill.
Soderberg, C. Richard ('24; '30) (EBSAKN), Prof., Mass. Inst. of Tech., Cambridge, Mass.
Soderberg, E. W. (J'35), Asst. Factory Mgr., Watertown Mfg. Co., Watertown, Conn.
Soderberg, O. Albert ('42) (FS), Dist. Mgr., Combustion Engrg. Co., Inc., 1537 Book Bldg., Detroit 26; *for mail,* 515 University Pl., Grosse Pointe 30, Mich.
Soderberg, Sten ('45) (JSZ), Ch. Engr., Leslie Co., Lyndhurst; *for mail,* 490 Passaic Ave., Nutley, N.J.
Soderquist, Leslie Edward ('43) (BMNY), Ch. Engr., McNeil Mch. & Engrg. Co., 96 E. Crosier St., Akron 11, Ohio.
Sodré, Lauro (Neto) (J'45) (CZONM), Design Engr., Companhia Brasileira de Instrumentos de Controle, Ave. do Estado 5081, São Paulo, Brazil, S.A.
Soiya, George William (J'45) (HKS), Designer, Clearing Mch. Corp., 6499 W. 65th St.; *for mail,* 5741 W. 64th Pl., Chicago, Ill.
Sokol, Sylvester Theodor ('42) (COB), Exec. Mech. Engr., Thatcher Mfg. Co.; *for mail,* 742 W. Church, Elmira, N.Y.
Sokol, Theodore Joseph ('45) (CMNOLJ), Indus. Engr., Natl. City Bank, E. 6th St. & Euclid Ave., Cleveland, Ohio.
Sola, Samuel Leonard (J'41) (ABEZ), Radio Technician, 2nd Class, S 105-R/S, Puget Sound Naval Shipyard, Bremerton, Wash.; *for mail,* 3333 Washington, Chicago, Ill.
Solberg, Harry Leland ('21; '28; '32) (SEFK), Head, Sch. of Mech. Engrg., Purdue Univ., Lafayette, Ind.
Soldan, Henry M. ('35), Asst. Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark; *for mail,* 195 Hillside Ave., Leonia, N.J.
Soler, Antonio (J'39) (LHSCM), Box 604, Caguas, P.R.
Soling, Sam Pinover ('31; '43) (KHZBLS), Sr. Application Engr., York Corp.; *for mail,* 425 W. Jackson St., York, Pa.
Solla, Alexander Philip (J'44), 34 Los Pinos St., Santurce, P.R.
Sollmann, Carl Henry (J'43) (B), Devel. Engr., Specialties Mfg. Co., Inc., 35 Farrand St., Bloomfield; *for mail,* 214 N. Mountain Ave., Montclair, N.J.
Solomon, John W. ('45) (ABE), Sr. Mech. Engr., Buick, Oldsmobile, Pontiac Div., Gen. Motors Corp., Linden, N.J.; *for mail,* Woodward Hotel, 55th St. & Broadway, New York 19, N.Y.
Solov, Abraham ('32; '44) (SKBCEN), Mar. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 7302—180th St., Flushing, L.I., N.Y.
Somers, Dwight LeRoy (J'34) (CKS), 136 Baldwin Ave., Waterbury 23, Conn.
Somers, George William (J'43), 1041 Marquette Bldg., 140 S. Dearborn St., Chicago 3, Ill.
Somers, John Cameron ('42) (CDLRY), Pres., Sales Mgr., Indus. Products Engrg. Co., 1 Hunter St., Long Island City 1; *for mail,* 21-48—78th St., Jackson Heights, N.Y.
Somers, William Eugene (J'33) (SFKR), Engrg. Asst., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Somervell, Philip A. (J'36) (MNJOEL), 601 Riverbank, Riverton, N.J.
Somervell, Somervell Bruce ('44) (HAYREM), Engr. Mgr., Application Co., 205 E. 42nd St., New York 17, N.Y.
Somerville, Geo. N. ('17; '35) (SLZBC), Mgr., Mchy. Dept., Atkins Kroll & Co., 230 California St., San Francisco 11; *for mail,* 1240 Carlotta Ave., Berkeley 6, Calif.
Sommer, William Bernard ('43) (ESMABR), Designing Engr., Joshua Hendy Iron Works, Hendy Ave., Sunnyvale; *for mail,* 360 Park Blvd., Palo Alto, Calif.
Sommers, Richard John (J'42) (BHJNFS), Engr., Oilgear Co., 1403 W. Bruce St., Milwaukee 4; *for mail,* 515 Lincoln Ave., Waukesha, Wis.
Somogyi, Charles E. ('24; '35) (CMNDJH), Prod. Mgr., Cincinnati Milling Mch. Co., Cincinnati 9, Ohio.
Sonderegger, John Robert (J'42) (NWDHTB), Asst. Engr., Long Reach Mch. Works, Box 2621, Houston 1, Tex.
Sonderman, Gerhard E. ('35; '36) (OSDL), Ch. Engr., Singmaster & Breyer, 420 Lexington Ave., New York 17, N.Y.; *for mail,* W. Brother Dr., Greenwich, Conn.
Sonn, Fred Wm. ('27; '35) (BCM), Asst. to Pres., Automatic Elec. Heater Co., Cross & Keim Sts.; *for mail,* 1204 Queen St., Pottstown, Pa.
Sonn, George Paul ('14; '16; '35) (CDMOSY), Cons. Engr., 117 Fenwood Ave., Trenton 9, N.J.
Sonnino, Mario (J'46) (HLT), Mech. Engr., Am. Cyanamid Co., 1937 W. Main St.; *for mail,* 60 Lanark Rd., Stamford, Conn.
Sonntag, Alfred ('41) (BCMNOZ), Pres., Treas., Sonntag Scien. Corp., 1 Seneca Pl., Greenwich, Conn.
Sontag, Herbert Paul ('25; '29; '35) (TDCNMS), Ch. Engr., Firth Carpet Co., 62 Columbus St., Auburn, N.Y.
Sontheimer, John Allen (J'42) (LKZCDS), Engr., Golden State Co., Ltd., 425 Battery St., San Francisco 11; *for mail,* 927 Clintonia Ave., San Jose 10, Calif.
Sookasian, George H. (J'46) (AJM), 3518 Shelmire Ave., Philadelphia 36, Pa.
Sooy, Walter Ernest ('27) (CGSM), V.P., Gardner-Richardson Co.; *for mail,* 3205 Fleming Rd., Middletown, Ohio.
Sopenoff, Bernard Hamilton (J'45) (BMNSZL), Seaman, 1st Class, Electronics Technician's Mate, U.S.N., Navy Pier, Chicago, Ill.; *for mail,* 1800 E. 18th St., Brooklyn 29, N.Y.
Sopenoff, Leon Paul (J'44) (CJM), Technician, 4th Grade, U.S.A., 416 Ord., M.M. Co., Ft. Bragg, N.C.; *for mail,* 8894—19th Ave., Brooklyn 14, N.Y.
Soper, Cleveland Conner ('45) (JMY), Engr., Ferry Mch. Co., West Main St.; *for mail,* 513 Earl Ave., Kent, Ohio.
Soper, Henry F. ('42) (ZHN), V.P., Neptune Meter Co., 50 W. 50th St., New York 20, N.Y.
Sopris, Robley Freeman ('45) (CMRBOE), Pres., Natl. Mch. Engrs., 213 Nelson Bldg., 4th & Broadway, Los Angeles 13, Calif.
Sorensen, H. P. (J'41) (ABJ), Test Engr., Field of Vibration, Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St.; *for mail,* 35 Hanmer St., East Hartford 8, Conn.
Sorensen, Harry Arnold ('31; '43) (SKEFH), Analytical Engrg., Gas Turbine Devel., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford 8; *for mail,* 20 Barker St., Hartford 6, Conn.
Sorensen, Kresten Theodore ('45) (CBMNJR), V.P., Asst. Gen. Mgr., Wm. Sellers & Co., 1600 Hamilton St., Philadelphia; *for mail,* R.D. 2, Middletown Rd., Media, Pa.
Sorenson, Alfred Edward ('27; '37) (EHSA), Assoc. Prof. Mech. Engrg., Princeton Univ.; *for mail,* 150 Patton Ave., Princeton, N.J.
Soria, Guido ('40) (OCM), Pres., European Am. Trade Devel. Corp., 37 Wall St., New York; *for mail,* 120 Cabrini Blvd., New York 33, N.Y.
Sossner, Theodore Townsend (J'40) (CJMN), Gen. Mgr., Sossner Tap & Tool Corp., 27 Broadway, Lynbrook; *for mail,* 600 W. 111th St., New York 25, N.Y.
Soulen, Peter James ('27; '35) (O), Cons. Engr., 111 E. Wisconsin Ave., Milwaukee 2, Wis.
Soulis, Wilbur Taggard ('25) (MCJDBG), Dixie Cup Co., 401-31 N. Western Ave., Chicago 12, Ill.
Souter, John Clyde (J'45) (ACO), Prod. Engr., West. Elec. Co., Inc., Locust St.; *for mail,* 495 Main St., Haverhill, Mass.
Southack, Tilden Ward (J'33) (HNRAZB), Ch. Exper. Engr., Acrotorque Plant, Curtiss-Wright Corp., 880 Main St., Stamford; *for mail,* Cherry Hill Rd., Greenwich, Conn.
Souther, George Luke, Jr. (J'44) (KRS), 837 Hillyer Ave., Macon, Ga.
Southerland, Thomas Chapman ('37) (YSEKHG), Sales Engr., Topflight Tool Co., 126 Maiden Lane, New York 7; *for mail,* 200 Clinton St., Brooklyn 2, N.Y.
Southern, Herbert ('19; '23; '35) (FKEBSN), Managing Dir., G. P. Wincott Ltd., 180 Attercliffe Rd., Sheffield 4; *for mail,* 76 Knowle Lane, Sheffield 11, England.
Southmayd, Charles Goodrich ('38; '44) (HCMJSN), Engr., Hyd. Dept., Canadian Allis-Chalmers, Ltd., 212 King St. W., Toronto; *for mail,* 8 Sunnybrook Rd., Toronto 3, Ont., Can.
Southwell, Richard V. (Non-Member), *Worcester Reed Warner Medallist,* '41; Rector, Imperial College of Sci. & Tech., Univ. of London, Prince Consort Rd., London, S.W. 7, England.
Southwick, Bertram Holden ('24; '25; '35) (SOFH), Engr., Elec. Supt's Office, Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 37 Elvir St., East Lynn, Mass.
Southworth, Fred Meier (J'42) (MZLC), Trainee, Gen. Elec. X-Ray Corp., 2212 Jackson Blvd., Chicago; *for mail,* 8 N. Lancaster, Aurora, Ill.
Sowden, Parkin Thomas ('08; '15; '20) (CDJLMR), Gen. Mgr., Torsion Balance Co., 92 Reade St., New York 13, N.Y.
Sowers, David Wood ('15) (C), Retired; 83 Bryant, Buffalo 9, N.Y.
Spahr, John Carl (J'42) (SBK), Turbine Design Engr., South Philadelphia Works, Westinghouse Elec. Corp., Lester; *for mail,* 220 E. Leamy Ave., Springfield, Pa.
Spainhour, Carroll Davis (J'44), 23 Parkway, Mountainside, N.J.
Spalding, Albert Ruff (J'42) (SKMEFW), Instr. in Mech. Engrg., Purdue Univ., West Lafayette; *for mail,* 919 Pontiac Ave., Lafayette, Ind.
Spalding, Francis Wheeler (J'24) (KSFBCL), Mech. Engr., Procter & Gamble Co., M. A. & R. Bldg., Ivorydale 17; *for mail,* 235 Greendale Ave., Cincinnati 20, Ohio.
Spalding, Luther P. ('45) (AJYBZL), Ch. Research Engr., N. Am. Aviation, Inc., 5701 Imperial Highway; *for mail,* 510 S. Flower St., Inglewood, Calif.
Spamer, Adolph Milton (J'40) (BMN), Student, Johns Hopkins Univ., Baltimore; *for mail,* 1606 N. Port St., Baltimore 13, Md.
Spang, Ferdinand J. ('43) (CMJNB), Pres., Spang & Co., Box 751, Butler, Pa.
Spang, J. Bennett (J'42), Ensign, U.S.N.R., 133 Rogers St., Aberdeen, Md.
Spangler, Samuel F. ('20; '25; '35) (LO), V.P., Chem. Constr. Corp., 350—5th Ave., New York 1; *for mail,* 71 Brewster Rd., Scarsdale, N.Y.
Spangler, Theodore Allen (J'43) (ACBO), Engr., Boeing Aircraft Co.; *for mail,* 4026 Brooklyn Ave., Seattle 5, Wash.
Spanjer, Clarence Earle (J'42) (WCAEMR), Mech. Engr., Spanjer Bros., Inc., 1160 Howe St., Chicago 10, Ill.
Sparkes, Harry Peckham ('44) (ABEHJK), V.P. Spar-Holl Mfg. Co., 20 Park Ave.; *for mail,* 24 Belgrade Terrace, West Orange, N.J.
Sparks, A. C. ('26), 25 Eccleston Pl., London, S.W. 1, England.
Sparks, Cedric Harold ('29) (FSKDJ), Dir., Ch. Engr., Babcock & Wilcox Ltd., Farringdon St., London, E.C. 4; *for mail,* Cairnsmuir, Burwood Park, Walton-on-Thames, Surrey, England.
Sparling, Philip W. (J'38) (BHM), Machinist, Vickers Inc., 1400 Oakman Blvd., Detroit; *for mail,* 955 W. Kirby Ave., Detroit 2, Mich.
Sparrow, Stanwood W. ('18; '19; '23) (BEN), Dir. of Research, Studebaker Corp., South Bend 27; *for mail,* The Morningside Club, South Bend 24, Ind.
Spatz, Donald (J'43) (BNKAFJ), Research Equip. Design Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 17635 Parkmount Ave., Cleveland 11, Ohio.
Spaulding, David Chase, Jr. ('45) (ONJBAL), Product Engr., Bunting Brass & Bronze Co., 715 Spencer St.; *for mail,* 3622 Beechway Blvd., Toledo 9, Ohio.
Spaulding, Ellis Russell (J'32) (ABHKZ), Supvr., Flight Test Planning, Chance Vought Aircraft Div., United Aircraft Corp., S. Main St.; *for mail,* 29 Minor Ave., Stratford, Conn.
Spaulding, Harold Sheerer ('29; '40) (LKHCJD), Process Engr., C. F. Braun & Co., 1000 S. Fremont, Alhambra, Calif.
Spaulding, John Downs ('40) (CAMRDJ), Works Mgr., V.P., Nathan Mfg. Co. & Nathan Aircraft Devices, Inc., 416 E. 106th St., New York; *for mail,* 5 Barnard Pl., Manhasset, L.I., N.Y.
Spaunburg, Harvey Lewis ('21; '26) (CMNT), V.P., Mgr., Veeder Root Inc., Hartford 2, Conn.
Spear, Allan I. (J'38), Apt. 2F, Cyllune Court, Eutaw Pl., Baltimore, Md.
Spear, Lawrence York ('15) (CEM), Pres., Elec. Boat Co., Groton, Conn.
Spears, Earl C. ('44) (ESDYAR), Sales Engr., Westinghouse Elec. Corp., 303 E. Brady, Tulsa 3, Okla.
Specht, Edward James (J'46) (AES), Aviation Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 400 Puritan Rd., Swampscott, Mass.
Speciall, Jos. V. (J'37), 2836—56th Pl., Woodside, L.I., N.Y.
Spector, Baruch ('42) (NOM), Spector Engrg. Serv., 139-06 Pershing Crescent, Jamaica 2, L.I., N.Y.
Speed, Wm. Shallcross ('00; '04), Chmn. of Bd., Louisville Cement Co., 315 Guthrie St., Louisville, Ky.
Speh, Herman A. (J'38) (BNZA), Project Engr., Sperry Gyroscope Co., Inc., Lakeville Rd., Great Neck; *for mail,* 101 Horace Ave., Roosevelt, N.Y.
Speich, Carl Johan ('38) (DLBMCF), Asst. Supt., Lone Star Cement Corp., Hudson, N.Y.
Speicher, Paul John ('43) (CMNJD); Consultant, Mech. Engr., Reconstr. Finance Corp., 505 N. 7th St., St. Louis; *for mail,* 1510-A De Soto Ave., St. Louis 7, Mo.
Speier, Richard N. (J'40), Spec. Assignment, Plymouth Motor Co., Detroit; *for mail,* 117 Rhode Island Ave., Highland Park, Mich.
Speight, Herbert ('23) (BELMSY), Indus. Engr., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.
Speirs, George W. ('37) (LMCDYZ), Indus. Engr., Hamilton Radio Corp., 510—6th Ave., New York 11; *for mail,* 115-91—223rd St., St. Albans 11, L.I., N.Y.
Spellman, Charles B. ('18; '27) (HS), Asst. Supvg. Engr., I. P. Morris Dept., Baldwin Loco. Works, Paschall Sta., Philadelphia 42; *for mail,* 43 N. Norwinden Dr., Media R.D. 3, Springfield, Pa.
Spellman, Reuben (J'38) (ACM), Lt., U.S.N.R., 5809 N. Four Mile Run Dr., Arlington, Va.
Spence, Clyde Theodore (J'43) (MJCDOZ), Seaman, 1st Class, U.S.N.R. Armory, Toledo 1, Ohio; *for mail,* 612 Thompson Ave., Clairton, Pa.
Spence, Hubert de Laserre ('30) (RBJ), Asst. Engr., Natl. Malleable & Steel Castings Co., 10600 Quincy Ave., Cleveland 6, Ohio.
Spence, Richard Slater (J'39) (BNMA), Instr. in Mech. Engrg., Rensselaer Poly. Inst., Troy, N.Y.
Spence, Robert Andrew ('35; '45) (COLSM), Mech. Engr., Dept. of Bldgs. & Grounds, Lehman Hall, Harvard Univ., Cambridge; *for mail,* 83 Barnard Rd., Belmont 78, Mass.

Spence, Stanley Francis ('45) (CMO), Asst. Regional Mgr., Loss Prevention Dept., Liberty Mutual Ins. Co., Rm. 333, 10 Rockefeller Plaza, New York, N.Y.
Spencer, Alexander Charles ('21), Pres., Highcrest Securities Ltd., R.R. 3, London, Ont., Can.
Spencer, Alfred Charles, Jr. (J'28) (REF), Engr., Stand. Oil Co. of N.J., 26 Broadway, New York 4, N.Y.; *for mail*, 11 Harvard Ave., Maplewood, N.J.
Spencer, Benjamin H. ('27) (OSEGK), Mech. Engr., Sanderson & Porter, 52 William St., New York 5, N.Y.
Spencer, Chas. W. (J'34) (BCM), Engr., Bell Tel. Labs., Inc., 463 West St., New York, N.Y.
Spencer, Clarence G. ('11) (FOS), Pres., Baker & Spencer, Inc., 27 William St., New York 5, N.Y.
Spencer, Ervin Russell ('38) (EKSFH), Installation Engr., Cooper-Bessemer Corp.; *for mail*, 519 E. Chestnut St., Mt. Vernon, Ohio.
Spencer, Frank C., Jr. (J'34), Asst. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail*, 2157 Raritan Rd., Westfield, N.J.
Spencer, Frank Calvin ('08; '12) (MOBCYJ), 10 English Village, Cranford, N.J.
Spencer, Frank Walton (J'42) (BKS), 4572 W. 220, Fairview Village, Ohio.
Spencer, Fred'k A. ('17; '26) (BDJ), Plant Engr., Gen. Cable Corp., 26 Washington St., Perth Amboy; *for mail*, 86 Grove Ave., Woodbridge, N.J.
Spencer, Harvey Gerber (J'43) (YLCMZ), 216 W. Genesee St., Flint 5, Mich.
Spencer, John Donald (J'44) (HKCB), Engr., Canadian Blower & Forge Co. Ltd.; *for mail*, 136 Water St. S., Kitchener, Ont., Can.
Spencer, John Hazelton (J'44), 6 University Rd., Cambridge, Mass.
Spencer, Julian M. (J'39), Test Foreman, Bethlehem Steel Co., Sparrows Point; *for mail*, 6812 Dunhill Rd., Dundalk 22, Md.
Spencer, Truman Edward (J'45) (SOJCMH), Field Engr., Allis-Chalmers Mfg. Co., 621 Southern Bldg., Washington 5, D.C.
Spenner, Fred H. ('42) (RSNJBM), Asst. V.P., Charge of Engrg., Scullin Steel Co., 6700 Manchester Ave., St. Louis 10, Mo.
Sperling, Erich ('45) (JMNCAY), Methods Engr., Columbia Mch. Works, Inc., 255 Chestnut St., Brooklyn; *for mail*, Box 388, Beach St., Islip, N.Y.
Sperry, Albert F. ('26; '35), Hubbard Engrg. Co., Consulting Engrs., 225 N. Michigan Ave., Chicago 1, Ill.
Sperry, Clarence Edgar ('25; '31; '33) (RFC), Lt. Col., Ord. Dept., U.S.A., Exec. Officer, Jefferson Proving Ground, Madison, Ind.
Sperry, Elmer A., Jr. ('29), V.P., Sperry Products, Inc., 1505 Willow Ave., Hoboken, N.J.; *for mail*, Wilton, Conn.
Sperry, Roger Sherman ('33) (N), Dir. Mech. Research, Scovill Mfg. Co., 99 Mill St.; *for mail*, 133 Pine St., Waterbury 33, Conn.
Sperry, S. M. ('28) (FSKEZC), Ch. Chemist, Gilbert Associates, Inc., 412 Washington St.; *for mail*, 1804 Olive St., Reading, Pa.
Sperry, Samuel E., Jr. ('17; '24; '35) (BES), Designer, Electro Chem. Dept., E. I. du Pont de Nemours & Co.; *for mail*, 116 West 19th St., Wilmington, Del.
Sperzel, Joseph Mahlon (J'30) (LKBTSY), Engr., Buensod-Stacey, Inc., 60 E. 42nd St., New York 17, N.Y.
Spicacci, A. R. ('28; '34) (BJM), Mech. Engr., Torrington Co., South Bend 21; *for mail*, 1403 N. O'Brien St., South Bend 16, Ind.
Spicknell, Charles Eugene ('40) (CSK), Ch. Oper. Engr., East Peoria Sta., Cent. Ill. Light Co., 1126 W. Washington St., Peoria 8, Ill.
Spiehler, Clarence H. ('24), Rm. 1002, Chamber of Commerce Bldg., Cincinnati 2, Ohio.
Spielman, Bernard Alan (J'42) (ACP), 1104 S. Durbin St., Casper, Wyo.
Spielman, John Robert (J'43) (ZACER), Asst. Ship Supt., Naval Shipyard, Philadelphia 12, Pa.; *for mail*, 1517 D St., S.E., Washington 3, D.C.
Spievak, Joseph (J'34) (NMJGCB), Owner, Transkill Engrg. Co., 614½ N. San Vicente Blvd., Los Angeles 46; *for mail*, 2826 Selby Ave., Los Angeles 34, Calif.
Spiller, William Raymond ('39) (GNBCJ), Ch. Engr., Seybold Div., Harris-Seybold-Potter Co., 819 Washington St., Dayton 7, Ohio.
Spilman, Robert Bruce (J'41) (KSFB), Engr., Compressor Engrg. Dept., Ingersoll-Rand Co.; *for mail*, 496 Kneedler Ave., Phillipsburg, N.J.
Spink, Leland Kenneth ('42) (HZA), Engr., Charge of Flow Measurement, Foxboro Co., Neponset; *for mail*, 4 Gray Rd., Foxboro, Mass.
Spiro, Irving John (J'36) (BNMKCO), Ch. Research Engr., Mission Appliance Corp., 7101 McKinley, Los Angeles 1; *for mail*, 4956 N. Ledge St., North Hollywood, Calif.
Spisak, Alois Vincent (J'41) (E), Ensign, U.S.N.R., Convoy Escort Repair Unit, Navy 131, F.P.O., San Francisco, Calif.
Spitler, Theodore Milton (J'41) (KBMNO), Engr., Gen. Elec. Co., 1 River Rd., Schenectady; *for mail*, 821 Hampton Ave., Schenectady 8, N.Y.
Spitzley, Lester (J'44), 11542 Rochester, West Los Angeles, Calif.
Spivak, Benjamin Louis ('24; '30; '35) (SFEKDO), Cons. Engr., 595 Madison Ave., New York; *for mail*, Box 905, Grand Cent. Annex, New York 17, N.Y.
Spoerer, Edward John, Jr. (J'40) (SEFHKC), U.S.N., N.Y. Naval Shipyard, Brooklyn, N.Y.; *for mail*, 84 Orchard Rd., Maplewood, N.J.
Spofford, Harry H. R. ('15; '22) (CMS), Trust Officer, Boston Safe Deposit & Trust Co., 100 Franklin St., Boston; *for mail*, 11 Windsor Rd., Wellesley Hills, Mass.
Spofford, Warren Ames ('45) (KBHC), Gen. Elec. Co., 5 Lawrence St., Bloomfield; *for mail*, 17 Hawthorne Ave., Glen Ridge, N.J.
Spohr, Joseph Conrad, Jr. (J'42) (AFZBC), Test Engr., Wright Aero. Corp., Lockland; *for mail*, 2368 Victory Pkwy., Cincinnati 6, Ohio.
Spooner, Charles Willett, Jr. ('34; '44) (SENFK), Asst. Prof. Mar. & Mech. Engrg., 322 W. Engrg. Bldg., Univ. of Mich., Ann Arbor, Mich.
Spooner, Franklin Maine (J'43) (KHZY), Chem. Engr., Process Control Dept., Bayway Refinery, Stand. Oil Co. of N.J., Linden; *for mail*, 120 North Ave., Elizabeth 3, N.J.
Spooner, John Clark (J'41), Lt. (j.g.), U.S.S. *Charles S. Sperry*, DD-697, F.P.O., San Francisco, Calif.
Sporleder, Edmund O. ('32; '44) (SNMCO), Ch. Engr., Laclede Stoker Co., 4438 Hunt Ave., St. Louis 10, Mo.
Sporn, Philip ('33) (SFKHE), Exec. V.P., Ch. Engr., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 8, N.Y.
Spotton, Arthur Knowlson ('06; '21) (EFHKS), Advis. Engr., Babcock-Wilcox & Goldie-McCulloch Ltd., 64 Grand Ave. S.; *for mail*, 92 Wentworth Ave., Galt, Ont., Can.
Spotts, Merhyle Franklin ('40) (BNJRM), Assoc. Prof. Mch. Design, Tech. Inst., Northwest. Univ., Evanston, Ill.
Spotts, William Max (J'46), 117 S. Stephens, Hobart, Okla.
Sprafke, David William (J'40) (MCS), Mech. Engr., Miller Co., 99 Center St., Meriden, Conn.
Sprague, E. Russell (J'36) (CDM), Prod. Mgr., Tampax Inc., Palmer, Mass.
Sprague, Lucian C. ('40) (CR), Pres., Minn. & St. Louis R.R. Co., 731 N.W. Bank Bldg., Minneapolis 2, Minn.
Sprague, Oscar Vosburgh ('21; '35) (SCFK), Asst. Supt., Power Dept., Kodak Park Works, Eastman Kodak Co., Rochester 13, N.Y.
Sprague, Philip Tripp ('26) (ZFSCBM), Pres., Hays Corp., Box 299, Michigan City, Ind.
Sprague, Robert H. (J'41) (CDM), Lt. (j.g.), 1768 Dean St., Brooklyn, N.Y.
Sprague, Theodore Smith (J'36) (HKS), Contract Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Sprague, William P. (J'45) (NCKJTB), Design Engr., Rayon Dept., E. I. du Pont de Nemours & Co., River Rd., Buffalo 7; *for mail*, 74 W. Cleveland Dr., Buffalo 21, N.Y.
Sprague, William Wallace ('28), Exec. V.P., Savannah Sugar Refining Corp., Savannah, Ga.
Sprake, Tyler Watson ('43), Cons. Engr., Hill, White & Sprake, Polson Bldg., Columbia & Western, Seattle 4, Wash.
Spraragen, William ('45) (BJ), Dir., Welding Research Council, 29 W. 39th St., New York 18; *for mail*, 119-47—177th Pl., St. Albans, N.Y.
Spratt, Hereward Philip (A'33), Asst. Keeper, Sci. Museum, London, S.W. 7, England.
Sprau, Benjamin W. ('38) (SKFC), Asst. Sales Mgr., Erie City Iron Works, Erie, Pa.
Spray, Claude Leroy ('43) (SFCOEK), Cons. Engr., N.Y. Life Ins. Co., 51 Madison Ave., New York 10; *for mail*, 300 W. 12th St., New York 14, N.Y.
Spray, Ellis L. ('44) (CMKO), V.P., Westinghouse Elec. Corp., 150 Pacific Ave., Jersey City 4; *for mail*, 225 Waiku Rd., Ridgewood, N.J.
Sprenkle, R. E. ('43) (AHZS), Hyd. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10; *for mail*, 2149 Reyburn Rd., Cleveland, Ohio.
Spriggs, Cecil T. ('45) (BNC), Sr. Engr., Otis Elev. Co., 260—11th Ave., New York 1; *for mail*, 60 Drake St., Malverne, N.Y.
Spring, Harry M. ('36; '39) (RSJFCH), Asst. Ch. Inspr., Mutual Boiler Ins. Co., 60 Batterymarch St., Boston 10; *for mail*, 280 Pleasant St., Canton, Mass.
Springe, Walter (A'23) (SFK), Mgr., Combustion Engrg. Co., Inc., 461 Bd. of Trade Bldg., Kansas City 6, Mo.
Springer, Edwin Kent ('36; '45) (NBCM), Asst. Prof. Mech. Engrg., College of Engrg., Univ. of So. Calif., University Park, Los Angeles 7, Calif.
Springer, Eugene Roy (J'43) (AB), Engrg. Designer, N. Am. Aviation, Imperial Highway, Inglewood; 4135—10th Ave., Los Angeles 43, Calif.
Springer, Russell Severance ('13), Retired; 1075 California St., San Francisco 8, Calif.
Spruitenburg, M. J. (J'45) (BKO), 5323 Macomb St., Washington 16, D.C.
Spurgeon, Joseph H. ('27) (SDLJEF), Sales Engr., Spurgeon Co., 5050 Joy Rd., Detroit 4, Mich.
Spurling, Oliver C. ('07) (AEFKOS), Retired; 315 Highland Ave., Upper Montclair, N.J.
Spurlock, Benjamin Hill, Jr. (J'43) (KHAE), Asst. Prof. Mech. Engrg., Univ. of Colo., Boulder, Colo.
Squeo, Joseph Richard (J'41), Cpl., Hdq. Co., 166th Inf., 37th Div., Camp Shelby, Hattiesburg, Miss.
Squire, Harris Alfred (J'41) (CMOBAJ), Lt. (j.g.), U.S.N.R., Bur. of Ord., Navy Dept., 18th St. & Constitution Ave., Washington, D.C.; *for mail*, 49 Franklin, Chagrin Falls, Ohio.
Squires, Richard (J'45) (GYDZM), Maint. & Devel. Engr., Natl. Publ. Co., 239 S. American St., Philadelphia 5; *for mail*, Gladwyne, Pa.
Staber, George Ingraham ('29; '46) (SLEFKO), Ch. Engr., Schware Labs., Inc., 202 E. 44th St., New York 17; *for mail*, R.F.D. 1, Huntington, L.I., N.Y.
Stabile, Vincent Albert (J'41) (CY), Lt. (j.g.), U.S.N.R., Div. 54-I, Naval Repair Base, San Diego 36, Calif.
Stackhouse, Howard Leigh (J'22) (SCFDBN), Plant Engr., Post Products Div., Gen. Foods Corp., 275 Cliff St., Battle Creek, Mich.
Stacy, Luke E. (J'40) (KSOBHE), Field Engr., Allis-Chalmers Mfg. Co., 2037 Koppers Bldg., Pittsburgh 19, Pa.
Stacy, Stanley C. ('21; '25; '35) (SKFCEZ), Mech. Engr., Bd. of Education, 13 S. Fitzhugh St., Rochester 4, N.Y.
Stacy, Thos. F. ('38) (CHM), Mgr., Hyd. Press Div., French Oil Mill Mchy. Co., Piqua, Ohio.
Stad, Andrew Nelson (J'38) (BMALC), Sr. Tool Designer, Ford Instrument Co., Rawson St. & Nelson Ave., Long Island City; *for mail*, 920—77th St., Brooklyn 28, N.Y.
Stadler, John ('20) (WSHOFD), Cons. Engr., 1117 St. Catherine St. W., Montreal, Que., Can.
Staege, Stephen Albert ('17; '18) (LYHKO), Chmn., Research & Devel., Black-Clawson Co., 2nd & Vine Sts.; *for mail*, 702 Main St., Hamilton, Ohio.
Stadler, Neumann Marshall (J'37) (JNHMBC), Engr., Rome Div., Revere Copper & Brass, Inc.; *for mail*, 1008 Beecham Blvd., Rome, N.Y.
Stafford, Carlos E. (J'40) (HKSW), Res. Mill & Constr. Engr., Plant Engr., B.F.D. Co., Ogdensburg, N.Y.
Stafford, John Wesley ('23) (KL), Pres., Turl Iron & Car Co., Inc., 82 Beaver St., New York 5, N.Y.
Stafford, Paris Howard (J'36) (HAMSN), Sec. 13, Pilotless Aircraft Unit, Naval Air Sta., Mojave, Calif.
Stahl, Edward C. M. ('26) (CS), Asst. Personnel Dir., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail*, Box 43, Chappaqua, N.Y.
Stahl, Eugene R. (J'37), 2722—21st St., Wyandotte, Mich.
Stahl, John Forrest (J'28) (CBJ), V.P., Stahl Gear & Mch. Co., 3901 Hamilton Ave., Cleveland 14; *for mail*, 135 E. 209th St., Euclid 19, Ohio.
Stahl, William Franklin (J'45) (KBSHC), Jr. Engr., Steam Div., Westinghouse Elec. Corp., Lester; *for mail*, 4317 Spruce St., Philadelphia 4, Pa.
Staiger, William M. ('23) (EFKLSZ), Mech. Engr., Sheffield Farms Co., Inc., 524 W. 57th St., New York; *for mail*, 8715—63rd Dr., Forest Hills West, N.Y.
Stainton, Robert S. ('43) (CMEZ), Assoc. Prof. Indus. Engrg., Bradley Poly. Inst., Peoria 5, Ill.
Stalder, Jackson R. (J'42) (AKN), Mech. Engr., Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field; *for mail*, Box 486, Route 1, Los Altos, Calif.
Staley, Duward Charles (A'42) (HYJLC), Pat. Atty., Toulmin & Toulmin, 308 W. 1st St., Dayton 2; *for mail*, 300 Shroyer Rd., Dayton 9, Ohio.
Stalhuth, William Edward (J'46), 2386 Grand Ave., Bronx, N.Y.
Stalker, Dwight Flack ('45) (CSKZF), Tech. Engr., Pacific Gas & Elec. Co., 51 Jefferson, Oakland; *for mail*, 438—49th St., Oakland 9, Calif.
Staller, Jack Joseph (J'42) (JAB), 83 Shore Dr., Winthrop 52, Mass.
Stamer, Frank R. ('18; '23; '29) (CMBDL), V.P., Charge of Prod., Sapolin Co. Inc., 229 E. 42nd St., New York 17; *for mail*, 32 Beech St., Floral Park, N.Y.
Stamets, William Kerr, Jr. (J'43) (BCJMN), Lt., U.S.N.R., Asst. Damage Control Officer, U.S.S. *Intrepid*, CV-11, F.P.O., San Francisco, Calif.; *for mail*, Wm. K. Stamets Co., Jenkins Arcade Bldg., Pittsburgh 22, Pa.
Stamm, John Duncan (J'32) (CJMO), 327 W. Pike St., Canonsburg, Pa.
Stampfl, Edward G. (J'41) (CJS), 1169 W. Front St., Plainfield, N.J.
Stampfle, Robert Buford (J'44) (RBJ), Engr., Devel. Dept., Bethlehem Steel Co., 701 E. 3rd St., Bethlehem, Pa.
Stampil, Leon Alvin (J'41) (R), Sgt., Co. A, 753rd Ry. Shop Bn., A.P.O. 512, c/o Postmaster, New York, N.Y.; *for mail*, 3531 E. New York St., Indianapolis 1, Ind.
Stanbrook, Reginald C. ('26) (SFCDBK), Mar. Engr., Pittsburgh Steamship Co., Rockefeller Bldg., Cleveland 13, Ohio.
Stancliff, Arthur Dale ('12; '19; '35) (BCDFS), Gen. Supt., Lone Star Cement Corp., 1120 Hibernia Bldg., New Orleans 12, La.

Standers, Leonard (J'43) (ABH), 2nd Lt., Army Air Forces, Weight & Balance Engr., Sec. F, Casper, Wyo.
Standerwick, Reginald G. ('45), Engr., Aircraft Gas Turbine Div., Gen. Elec. Co., 920 Western Ave., Lynn, Mass.
Standley, Bryan Hal (J'46) (DJL), Engr., Designer, So. Coach Mfg. Co.; *for mail,* Box 2, Evergreen, Ala.
Stanek, Jerome H. (J'36) (NMYCJB), Ch. Engr., Stanek Tool & Mfg. Co., 2900 W. Vliet St., Milwaukee 8, Wis.
Stanfield, Max L. (J'39), Mech. Engr., Ill. Pipe Line Co.; *for mail,* Martinsville, Ill.
Stanfield, Rowland S. ('42) (LKEFJC), Ch. Engr., Tide Water Associated Oil Co., E. 22nd St., Bayonne, N.J.
Stang, John Herbert (J'44) (FRS), Research Engr., Battelle Memorial Inst., 505 King Ave.; *for mail,* 503 W. 7th Ave., Columbus, Ohio.
Stangeland, Ole Ingvaldson (J'36) (BNHKMA), Design Engr., Foote Bros. Gear & Mch. Corp., 5225 S. Western Blvd., Chicago 9; *for mail,* 3729 N. Newcastle Ave., Chicago 34, Ill.
Stanger, Walter Eugene (J'42) (ESAKNM), Mech. Engr., Gen. Elec. Co., 920 Western Ave.; *for mail,* 82 Myrtle St., Lynn, Mass.
Stangland, Robert S. ('16) (ODLSEF), Ch. Engr., By-Products Recoveries, Inc., 90 West St., New York 6, N.Y.
Staniar, William ('14; '16) (BDLS), Mech. Power Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.
Stanier, Sir William Arthur (H'45) (RNMCAD), Sci. Adviser, British Govt., 2 Central Bldgs., Westminster, London, S.W. 1, England.
Stanitz, John Daniel (J'45) (BA), Sec. Head, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 1085 Wilbert Rd., Lakewood 7, Ohio.
Stanley, A. W. ('93; '03), Box 74, New Britain, Conn.
Stanley, C. Maxwell ('33; '35) (ESHO), Partner, Stanley Engrg. Co., Cent. State Bank Bldg., Muscatine, Iowa.
Stanley, Carroll Mark ('21; '23) (CNJLDB), Sales Engr., Pulverizing Mchy. Co., Chatham Rd.; *for mail,* 29 Edgewood Rd., Summit, N.J.
Stanley, Clyde Earl (J'42) (CSL), Lt., U.S.N.R., Box 15F, Armory Annex 3, NavPhib Base, Little Creek, Va.
Stanley, John Gilbert (J'45) (LYSHDB), Engr., Mech. Engrg. Dept., Am. Hard Rubber Co., 11 Mercer St., New York 13; *for mail,* 386 Wilson Ave., Brooklyn 21, N.Y.
Stansfield, Frank H. ('16; '35), Community Chest & Council, Inc., 881 Lafayette St.; *for mail,* 20 W. Liberty St., Bridgeport, Conn.
Stansfield, William A. (J'31) (RYMJSF), Fla. Power & Light Co., Lake Monroe; *for mail,* Box 712, Sanford, Fla.
Stanton, Curtis Henderson (J'42) (CS), Supvr. Sales Trng., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 2032 Coolidge Pl., Schenectady 8, N.Y.
Stanton, John James ('45) (OY), 70 Winthrop St., Framingham, Mass.
Stanton, Robert Eugene ('45) (LOP), Cons. Engr., Box 1261, Terminal Sta.; *for mail,* 171 Downing St., Denver, Colo.
Stanwick, Chas. A. ('20; '35), 131 Rynda Rd., South Orange, N.J.
Stanyan, S. W. ('22; '35) (CDM), Factory Mgr., Hoague-Sprague Plant, United Shoe Mchy. Corp., 130 Eastern Ave., Lynn; *for mail,* 4 Mystic Bank, Arlington, Mass.
Stapfer, Rudolf D. ('38; '42) (EHS), Lt., U.S.N.R., Bur. of Ships, Navy Dept.; *for mail,* 5434 Westford Rd., Philadelphia, Pa.
Staples, Arthur Justin ('44) (EKS), Asst. Prof. Mech. Engrg., Worcester Poly. Inst., Boynton St., Worcester 2, Mass.
Staples, Charles Albert ('38) (LENSCZ), Asst. Edible Prod. Supvr., Procter & Gamble Mfg. Co., 1226 Loomis St., Dallas; *for mail,* 4806 Gaston Ave., Dallas 4, Tex.
Staples, Charles W. ('24; '32; '35) (AB), Research Engr., Socony-Vacuum Oil Co., Inc., Paulsboro; *for mail,* 116 Cyrus Ave., Pitman, N.J.
Staples, Frank C. ('36; '39) (DMCK), V.P., Am. Molasses Co., 280 Richards St., Brooklyn 31; *for mail,* 168 Aspen St., Floral Park, N.Y.
Staples, William J. (J'42) (CJM), Mech. Engr., Demerara Bauxite Co., Mackenzie, British Guiana, S.A.
Stapleton, Lynwood A. (J'38) (JMS), Ensign, U.S.N.R.; *for mail,* 319 Park Ave., Rock Hill, S.C.
Starbird, Clinton Virgil (J'41) (WCMKSE), Factory Mgr., Estate of C. V. Starbird, Main St., Strong, Me.
Starbuck, George Franklin ('01; '23) (RBJYW), Calculator, Mech. Engrg. Office, Boston & Me. R.R., High St., North Billerica; *for mail,* 141 Weston St., Waltham 54, Mass.
Starbuck, Robert A. (J'33) (DSWFL), Maint. Engr., Imperial Paper & Color Corp.; *for mail,* 17 Flandreaux Ave., Glens Falls, N.Y.
Stark, Albert Clark (J'41) (SZFCOH), Stoker Engr., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail,* 547 Holmes Rd., Morton, Pa.
Stark, Julian Emil ('17; '20) (CJ), Mgr., Engrg. & Research Div., Crane Co., 836 S. Michigan Ave., Chicago 5, Ill.
Stark, LaRue Hollister ('25) (CDFMST), Plant Engr., Phœnix Hosiery Co., 320 E. Buffalo, Milwaukee 1, Wis.
Stark, Martin (J'42) (AEF), Carburetion Test Engr., Wright Aero. Corp., Paterson; *for mail,* 4-36 Grunauer Pl., Warren Point, N.J.
Stark, Michael J. (J'39) (CJM), 957 Glendon, Route 6, St. Paul 9, Minn.
Starke, Otto Andrew (J'28) (CMJNB), Pres., Gen. Mgr., Star Watch Case Co., S. Rath Ave., Ludington, Mich.
Starke, Thomas Jefferson ('40) (BMJKD), Pres., Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond, Va.
Starke, William Walter ('27; '38) (TSLFCZ), Asst. Plant Engr., Forstmann Woolen Co., 2 Barbour Ave., Passaic; *for mail,* 154 Sheridan Ave., Hohokus, N.J.
Starkweather, James Oliver (J'44) (NOHM), Lt. (j.g.), U.S.N.R., Engr. Officer, U.S.S. *Rodman,* DMS-21, F.P.O., New York; *for mail,* 23 Autenrieth Rd., Scarsdale, N.Y.
Starkweather, William G. ('97) (EFKS), Pres., Starkweather Engrg. Co., Inc., 246 Walnut St., Newtonville, Mass.
Starman, Edward J. (J'41) (CJM), 1035 S. Lombard Ave., Oak Park, Ill.
Staroba, John E. (J'42) (BHL), Aircraft Parts Prod. Engrg., Chicago Screw Co., 1012 S. Homan Ave., Chicago; *for mail,* 3257 S. Austin Blvd., Cicero 50, Ill.
Staroba, Joseph Frank ('43) (BHMNJC), Ch. Engr., Design Engrg. Div., Taylor Forge & Pipe Works, Box 485, Chicago; *for mail,* 1845 S. Maple Ave., Berwyn, Ill.
Starr, Charles Joseph ('19; '35) (MJB), Assoc. in Mech. Engrg., Univ. of Ill.; *for mail,* 606 W. Nevada St., Urbana, Ill.
Starr, Milland Orlando (J'35) (RACS), Instr. in Gen. Engrg. Drawing, Univ. of Ill.; *for mail,* 509 E. Washington St., Urbana, Ill.
Starrett, Richard Howard (J'41) (YNBMTD), Plastics Engr., B. F. Goodrich Co., 500 S. Main St., Akron 18; *for mail,* Box 534, Hudson, Ohio.
Stary, Marvin Leo (J'45), Project Engr., Aerojet Engrg. Corp., Pasadena; *for mail,* 409 W. Ramona Blvd., Alhambra, Calif.
Staszesky, Francis Myron (J'44) (SKBFJH), Jr. Engr., Design Div., Power Engrg., E. I. du Pont de Nemours & Co., Wilmington 98, Del.
States, Charles Bradley (J'44) (BHJMNY), Draftsman, Designer, Ross Gear & Tool Co., Heath St.; *for mail,* 1005 Heath St., Lafayette, Ind.
Statler, Charles W. (J'39) (ACS), Prod. Engr., Pa. Elec. Co., Box 539; *for mail,* 700 Linden Ave., Johnstown, Pa.
Staub, Morton H. (J'36) (CKS), 858 Ridgewood Rd., Millburn, N.J.
Staude, Edwin G. ('08; '17), Pres., E. G. Staude Mfg. Co., 2675 University Ave., St. Paul; *for mail,* 2232 Lake of Isles Blvd., Minneapolis, Minn.
Stauffer, Ralph Daniel ('22; '30) (SFKEDO), Ch. Engr., New England Gas & Elec. System, 727 Massachusetts Ave., Cambridge 39, Mass.
Stauverman, Edward, Jr. (J'38) (EK), 1st Lt., Army Air Forces, U.S.A.; 64 The Prado N.E., Atlanta, Ga.
Staver, Robert Berkeley (J'44), Capt., Liaison Office, Ord. Dept., Univ. of Calif., Pasadena 4, Calif.
Stead, Frank Arthur ('42) (CLS), V.P., Vulcan Iron Works, Main St., Wilkes-Barre, Pa.; *for mail,* Loudon Woods, Rye, N.Y.
Stead, Harry Garnet ('44) (CFKLMS), Pres., Ch. Engr., E. Leonard & Sons Ltd., 381 York St.; *for mail,* 79 Bruce St., London, Ont., Can.
Stearns, Ellis J., Jr. (J'34), 933 Parker St., Falls Church, Va.
Stearns, Frederick Arlington ('20; '24; '30) (SBFJMZ), Assoc. Prof. Mech. Engrg., Northeast. Univ., 360 Huntington Ave., Boston; *for mail,* 66 Florence Ave., Melrose 76, Mass.
Stearns, Karl Thompson ('17; '35) (SKHBDF), Asst. Engr., New England Power Serv. Co., 441 Stuart St., Boston 16; *for mail,* 73 Hawthorne Ave., Auburndale 66, Mass.
Stearns, Thomas Beale ('83; F'36) (CDES), *Vice-President,* '11-'13; Pres., Stearns-Roger Mfg. Co., 1716 California St., Denver, Colo.
Stearns, Walter Joshua ('44) (LOW), Mech. Engr., Oxford Paper Co.; *for mail,* 35 Lochness Rd., Rumford, Me.
Steck, Robert C. (J'44), 242 S. Columbus Ave., Mt. Vernon, N.Y.
Steckel, René Joseph (J'40) (NCLM), Product Engr., Kyle Corp., 9th & Marion Aves., South Milwaukee; *for mail,* 7008 W. Wisconsin Ave., Wauwatosa 13, Wis.
Steckler, Harry (J'41) (EKS), Pvt., Corps of Engrs., U.S.A., Co. B, 800th Sig. Trng. Regt., Camp Crowder, Mo.
Steel, John (J'27) (CLS), Asst. V.P., Ch. Engr., United Sugar Cos. S.A., Los Mochis, Sinaloa, Mex.
Steel, Ned Harry (J'44), Lt., Army Air Forces, 5202 Burt St., Omaha, Neb.
Steele, Andrew William (J'45) (MNLBHS), Sr. Draftsman, Engrg. Div., Canadian Vickers Ltd., Maisonneuve, Montreal, Que., Can.
Steele, Arthur Eugene (J'40) (JCMH), Draftsman, Natl. Cast Iron Pipe Co., Tarrant; *for mail,* 1131 N. 27th St., Birmingham 4, Ala.
Steele, Charles Henry (J'44), 1529 Thomas Ave., Charlotte 4, N.C.
Steele, John H. (J'41) (CKS), Ensign, U.S.N.R., U.S. Naval Argus, Unit 14, F.P.O., San Francisco, Calif.
Steele, Malcolm Aubrey, Jr. (J'40), Ch. of Operas., Supersonic Wind Tunnels Lab., Ballistic Research Lab., Ord. Research & Devel. Center, Aberdeen Proving Ground; *for mail,* 37 Defense Dr., Aberdeen, Md.
Steele, Maurice Gaylord ('26) (CMYDGJ), Secy., Factory Mgr., Kent Co., Inc., 107 Canal St., Rome, N.Y.
Steele, S. ('28), Prof. Mech. Engrg., Canterbury College, Christchurch, New Zealand.
Steele, Thomas W. (J'42) (BKS), 849 Cross Ave., Elizabeth, N.J.
Steele, Walter David ('92; '01) (CDJML), Pres., Benjamin Elec. Mfg. Co., Des Plaines, Ill.
Steen, Arthur Benjamin, Jr. (J'27) (EKS), Ch. Draftsman, Humble Pipe Line Co., 684 Humble Bldg., Houston 2; *for mail,* 2801 Newman St., Houston 6, Tex.
Steen-Johnsen, H. ('33; '41) (BJS), Ch. Turbine Engr., Elliott Co.; *for mail,* Spanish Villa, Jeannette, Pa.
Steere, Harry Wing ('45) (COFMZK), Mgr. Mech. Serv., Constr. & Maint. Div., Marshall Field & Co., 121 N. State St., Chicago 90, Ill.
Stefanac, Joseph Barnaby (J'36) (BFK), Capt., U.S.N., Dist. Intelligence Office, 9th Naval Dist., Great Lakes, Ill.
Stefanov, Frank Joseph (J'45) (KSHZBL), 3rd Asst. Steamship Engr., McCormick Steamship Co. Div., Pope & Talbot, 19 Rector St., New York, N.Y.; *for mail,* 3 Brandes St., Webster, Mass.
Stefansky, Steve (J'41) (ASFEC), 2172 N. Merrimac Ave., Chicago 39, Ill.
Steffa, H. I. ('41), Fox Lake, Wis.
Steffan, Christian H. ('46) (ACL), 8010—78th Ave., Glendale, L.I., N.Y.
Steffani, Edward C. (J'31) (EHK), Office Engr., Gas Dept., Coast Counties Gas & Elec. Co., 22 Pacific St., Santa Cruz; *for mail,* Box 397, Route 1, San Jose, Calif.
Steffens, Jerome (J'44) (ABJ), Stress Analyst, Aero. Engr., Platt-LePage Aircraft Corp., Eddystone; *for mail,* 724 E. 24th St., Chester, Pa.
Steger, George Warren (J'44) (NOMJB), 1st Sgt., Cannon Co., 26th Inf., A.P.O. 1, c/o Postmaster, New York, N.Y.; *for mail,* 411—60th St., West New York, N.J.
Steglich, Herbert Hans ('45) (NBCM), Asst. Ch. Mech. Engr., Zenith Radio Corp., 6001 W. Dickens Ave., Chicago 39; *for mail,* 1027 Locust Rd., Wilmette, Ill.
Stehman, Frederick Francis ('44) (CEMS), Supt. of Mfr., Brooklyn Borough Gas Co., Mermaid Ave. & W. 17th St.; *for mail,* 4005 Beach 40th St., Brooklyn, N.Y.
Stehman, Frederick Godfrey (J'42) (CMDGJZ), Ch. Indus. Engr., E. A. Labs., 144 Spencer St., Brooklyn 5; *for mail,* 4005 Beach 40th St., Brooklyn 24, N.Y.
Steigerwalt, Robert W. ('39) (JR), Metal. Engr., Carnegie-Ill. Steel Corp., Carnegie Bldg., Pittsburgh; *for mail,* E-2 Alder Court Apts., Pittsburgh 6, Pa.
Stein, Fredrick William ('23; '32) (CMGD), V.P., Charge of Mfg., Stand. Register Co., Dayton 1; *for mail,* 256 Beverly Pl., Dayton 9, Ohio.
Stein, Irving Melville ('37) (ZCBJSF), V.P., Dir. of Research, Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44, Pa.
Steinbach, Edward Sargent ('21; '22) (FESLCM), Indus. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.
Steinbeck, Charles Earl ('27) (SFKEHJ), Asst. Engr., Pacific Gas & Elec. Co., 245 Market St., San Francisco 6, Calif.
Steinberg, Alvin Morris (J'43) (ABJKGH), 2nd Lt., Army Air Forces, 3538th Air Base Unit, Brainard Field, Hartford, Conn.
Steinberg, Harry (J'36) (BCM), Prod. Engr., Office of Chief of Ord., SPOIM, Engr. Branch, The Pentagon, Washington 25; *for mail,* 2913 M St., N.W., Washington 7, D.C.
Steinberg, Howard G. (J'36), Engr., Charge Design, Concord Burner Co., Inc., P.O. Box 66, Lawrence, N.Y.
Steinberg, Leonard (J'41) (NBLKC), Designer, E. I. du Pont de Nemours & Co., Wilmington 98, Del.; *for mail,* 5427 Wyndale Ave., Philadelphia 31, Pa.
Steinberg, Max Jacob ('39) (SKFEH), Div. Engr., System Engrg. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Steinbrenner, George Richard (J'42) (OLSFDZ), Engr., Niacet Chemicals Div., U.S. Vanadium Corp.; *for mail,* 1311 Pierce Ave., Niagara Falls, N.Y.
Steiner, Ezekiel Charles ('45) (BLO), Ch. Engr., Harvey Mch. Co., Inc., 6200 Avalon Blvd., Los Angeles 3; *for mail,* 745 N. Laurel Ave., Los Angeles 46, Calif.
Steiner, Kalman ('45) (FELK), Cons. Engr., E. Hoffberger Co., Monument & Forrest Sts., Baltimore 2, Md.
Steiner, Oscar ('29; '35) (CMZ), Ch. Engr., Graflex, Inc., Rochester 8; *for mail,* 117 Colebrook Dr., Rochester 12, N.Y.
Steiner, Walter Adolf ('21; '37) (CLKNFD), Product Mgr., Natl. Carbon Co., 80 E. 42nd St., New York 17, N.Y.
Steiner, William Avery (J'37) (BHZG), Draftsman, Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.; *for mail,* 2307 Lyon Ave., New York 61, N.Y.

Steinfeld, Maurice Eugene (J'39) (EM), Assoc. Mar. Engr., Design Sec., Naval Shipyard, Philadelphia; *for mail,* 6726 Wyncote Ave., Philadelphia 38, Pa.
Steinfeldt, William Martin (J'40) (K), 2nd Lt., Sanitary Corps, Medical Dept., A.U.S., Smoky Hill Army Air Field, Salina, Kan.; *for mail,* 30 Belgard, Rochester 9, N.Y.
Steinke, Bernard John (J'43) (YSCMO), Project Engr., U.S. Rubber Co., Maple St., Naugatuck 2, Conn.
Steinke, John Jacob, Jr. (J'43) (SEDOLK), Mech. Engr., M. W. Kellogg Co., 225 Broadway, New York, N.Y.; *for mail,* 184 Kinderkamack Rd., Oradell, N.J.
Steinman, Herbert I. (J'38) (BWJCMN), Asst. Ch. Prod. Engr., Brunswick-Balke-Collender Co., Muskegon; *for mail,* 2121 Mills Ave., North Muskegon, Mich.
Steinman, John O. (J'39) (CMA), 518 Hillsboro Ave., Edwardsville, Ill.
Steinmetz, Arthur Martin (J'41), Asst. Ch. Engr., Power Plant, Thomas A. Edison, Inc., 51 Lakeside Ave., West Orange; *for mail,* 21 Summit Ave., Cedar Grove, N.J.
Steinmetz, Henry G., Jr. (J'41) (AC), 11 Soundview Ave., New Rochelle, N.Y.
Steinmeyer, John W. ('28) (RJDBML), Research Engr., Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.; *for mail,* 30 Gates Ave., Montclair, N.J.
Steinmeyer, Lawrence Andrew (J'45) (CABDMK), 298 Kohler St., Tonawanda, N.Y.
Steinmuller, John Michael ('44) (FKS), Ch. Mech. Engr., Thompson-Starrett Co., Inc., 444 Madison Ave., New York; *for mail,* 17 Dartmouth St., Valley Stream, N.Y.
Steins, Carleton Kenedon ('21; '35) (R), Mech. Engr., Pa. R.R. Co., Philadelphia; *for mail,* C-1, York Lynne Manor Apts., Overbrook, Philadelphia 31, Pa.
Steinsieck, John M. (J'37) (SKF), Asst. Boiler Rm. Engr., Pub. Serv. Elec. & Gas Co., W. Pearl St., Burlington; *for mail,* Oakford Ave., Delanco, N.J.
Stellwagen, Robt. H. ('24; '30; '35), Assoc. Mech. Engr., Charge of Pumping Stas., City of Detroit, 414 City Hall; *for mail,* 4324 Maryland Ave., Detroit, Mich.
Stelz, Frank F. ('45), Ch. Engr., Ideal Rubber Co., Inc., 273-299 Van Sinderen Ave., Brooklyn 7; *for mail,* 4706—46th St., Woodside, Long Island, N.Y.
Stem, Chester R. (J'40), Lt., 79th Inf. Trng. Bn., Camp Roberts, Calif.
Stem, Frank B. ('15) (CMFSE), Engr., Philadelphia Gas Works Co., 1401 Arch St., Philadelphia 5; *for mail,* 220 E. Mt. Pleasant Ave., Philadelphia 19, Pa.
Stenberg, Eric Roland (J'44) (NMGS), Layout Draftsman, J. W. Greer Co., 119 Windsor St., Cambridge 39; *for mail,* 34 Waverly Ave., Newton 58, Mass.
Stengel, Christian William, Jr. (J'42) (CJS), Lt. (j.g.), U.S.N.R., N.Y. Naval Shipyard, Brooklyn, N.Y.; *for mail,* 375 Bergen Ave., Jersey City, N.J.
Stengren, John S. ('43), 6860—108th St., Forest Hills, N.Y.
Stenhouse, James Alexander (J'45) (JYCDM), Asst. to Plant Engr., Aluminum Goods Ltd., 158 Sterling Rd., Toronto; *for mail,* 205 King St., Weston, Ont., Can.
Stentz, Frank W. ('28), Engr., Charge of Design, Carnegie-Ill. Steel Corp.; *for mail,* 709 Fillmore St., Gary, Ind.
Stepan, Theodore Emanuel ('36; '40) (NHSOBG), Mech. Engr., Ch. Mech. & Elec. Design, Miss. River Comm., Box 80, Vicksburg, Miss.
Stepanoff, Alexey J. ('27; '33) (HSB), *Melville Medallist,* '32; Devel. Engr., Ingersoll-Rand Co., Phillipsburg, N.J.
Stephan, Erich Richard ('39; '46) (BGJLN), Asst. Prof., Univ. of Conn., Storrs, Conn.
Stephan, Walter George ('20) (SFAYE), Pres., Stephan Co., 7016 Euclid Ave., Cleveland 3, Ohio.
Stephens, Ernest Leslie ('28; '37) (SEFHML), Mech. Engr., Tex. Co., 205 E. 42nd St., New York 17, N.Y.
Stephens, Eugene Graham, Jr. (J'45), 621 E. 22nd St., Oakland, Calif.
Stephens, Gordon Albert ('36) (SFCDKH), Managing Dir., Affiliated Engrg. Corporations Ltd., 1606 Univ. Tower, Montreal; *for mail,* 536 Stanstead Ave., Town of Mt. Royal 16, Que., Can.
Stephens, James O. (J'39), Box 171, Swarthmore, Pa.
Stephens, Russell Montgomery (J'43) (AJMRN), Assoc. Ord. Engr., Naval Gun Factory, Washington, D.C.; *for mail,* 2705 N. Florida St., Arlington, Va.
Stephenson, Ardell Moring ('32; '43) (FSWTLK), Dist. Rep., Iron Fireman Corp., 3170 W. 106th St., Cleveland, Ohio; *for mail,* 310 E. Boulevard, Charlotte 3, N.C.
Stephenson, Dale Quam (J'44) (CES), 5038 Luverne Ave. S., Minneapolis, Minn.
Stephenson, Gordon Cosler (J'45) (MOCTAK), Engr., Kurt Salmon Associates, 3000 Albemarle, N.W., Washington 8, D.C.; *for mail,* 3649 Aldrich Ave. S., Minneapolis 8, Minn.
Stephenson, John ('28; '35) (RSJBCF), Asst. Plant Engr., Natl. Steel Car Corp., Kennilworth Ave.; *for mail,* 97 Barons Ave. S., Hamilton, Ont., Can.
Stephenson, Paul A. ('30; '32; '35) (ACM), Ch. Engr., A. B. Dick Co., 720 W. Jackson Blvd., Chicago, Ill.
Stephenson, Revis Lindsay (J'38) (DCLS), Field Engr., U.S. Hoffman Mchy. Corp., 105—4th Ave., New York 3; *for mail,* 40-60 Elbertson St., Elmhurst, N.Y.
Stephenson, Thomas I., Jr. ('20; '27; '35) (CMSDFH), Personnel Mgr., Tenn. Operas., Aluminum Co. of Am., Alcoa; *for mail,* 1118 Oak Park Ave., Maryville, Tenn.
Stephenson, William Bishop (J'38) (HDS), Head, Pump Dept., Allen-Sherman-Hoff Co., 225 S. 15th St., Philadelphia 2; *for mail,* 37 Westview Ave., Philadelphia 19, Pa.
Stephenson, William Kersey (J'44) (WLSMCD), Plant Engr., Supt. Plant Engrg., U.S. Gypsum Co., Lisbon Falls, Me.
Sterbentz, William Henry (J'44), 1142 Addison Rd., Cleveland, Ohio.
Sterling, Albert Alexander, Jr. (J'39) (SEH), Sales Engr., Brance-Krachy Co., 4411 Navigation Blvd., Houston, Tex.
Sterling, Claude H. ('16; '19; '23) (ABC), 6414—8th Ave., Kenosha, Wis.
Sterling, John Carman ('27) (CHM), Supt., Mch. Shop Div., Newport News Shipbldg. & Dry Dock Co.; *for mail,* 2207 Parrish Ave., Newport News, Va.
Stern, Abraham Isaac (J'41) (AHKBSC), Ch. Test Engr., Stratos Corp., Babylon, L.I.; *for mail,* 85-55—115th St., Richmond Hill 18, N.Y.
Stern, Adolf (J'46), 87-11 Elmhurst Ave., Elmhurst, N.Y.
Stern, Allan Philip (J'37) (CJMD), *Charles T. Main Award,* '37; V.P., Gen. Mgr., Colonial Iron Works Co., 17643 St. Clair Ave., Cleveland 10; *for mail,* 26181 Briardale Ave., Euclid 17, Ohio.
Stern, Arthur Cecil ('30; '41 (LFHSKD), *Junior Award,* '38; Ch. Engr., Div. of Indus. Hygiene & Safety Stands., N.Y. State Dept. of Labor, 80 Centre St., New York 13; *for mail,* 55 W. 95th St., New York 25, N.Y.
Stern, Ferdi B., Jr. (J'40) (BJNADC), Project Engr., Magnaflux Corp., 5900 Northwest Highway; *for mail,* 6771 Talcott Rd., Chicago 31, Ill.
Stern, Joseph Henry (J'17) (ACLMN), Process Engr., Ranger Aircraft Engines Div., Fairchild Eng. & Airplane Corp.; *for mail,* 5 Elm St., Farmingdale, N.Y.
Sternal, Norbert J. (J'34) (ACDMNR), Indus. Engr., Minn. Min. & Mfg. Co., 367 Grove St., St. Paul 1, Minn.
Stertzbach, Harry Walter ('20) (RB), Ch. Mech. Engr., Buckeye Steel Castings Co., S. Parsons Ave., Columbus 7, Ohio.
Stessl, Carl Joseph (J'37) (AYZ), Ch. Inspr., Kearney & Trecker Corp., 68th & W. National Ave., West Allis 14; *for mail,* Eagle Springs Lake, Route 2, Eagle, Wis.
Stetler, Charles Orrin ('41) (SKFHBA), Investigations Mech. Engr., Commonwealth & So. Corp., 212 W. Michigan Ave., Jackson, Mich.
Stetson, George A. ('20; '35), Editor, A.S.M.E., 29 W. 39th St., New York 18, N.Y.
Stetson, Ralph Brown (J'45), Ensign, U.S.N.R., 16 Pond St., Georgetown, Mass.
Stetson, Ralph Wetherbee ('21; '37) (BJR), Struc. Designer, N.Y., New Haven & Hartford R.R., 71 Meadow St., New Haven 6; *for mail,* 26 Converse Ave., Meriden, Conn.
Stettler, Robert Harry (J'39) (RCJFMN), Spec. Apprentice, Mech. Dept., Pa. R.R., Philadelphia; *for mail,* 1404—20th Ave., Altoona, Pa.
Steur, William Raymond ('44) (SKFE), Asst. Mech. Engr., Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Steurer, George E. (J'44) (MN), Product Design, Buckeye Tools Corp., Box 966, Dayton 1, Ohio.
Stevens, Alfred Hull ('98; '03), 1st Selectman (Mayor), Town of Clinton; *for mail,* 33 High St., Clinton, Conn.
Stevens, Arthur Joseph (J'44) (ACJ), Estimator, Lockheed Aircraft Corp., Empire Blvd., Burbank; *for mail,* 7518 Farmdale Ave., North Hollywood, Calif.
Stevens, Carl A. ('24; '41) (CMLOSD), Works Engrg. Consultant, 1932 N. Boston Ave., Tulsa 6, Okla.
Stevens, Clarence Colvin ('21; '28; F'44) (BOMN), Ch. Mech. Engr., New Departure Div., Gen. Motors Corp., Bristol, Conn.
Stevens, George Edwin (J'43), 33 High St., New Rochelle, N.Y.
Stevens, George W. ('28; '34; '36) (CEF), 5904 El Campo Terrace, Ft. Worth 7, Tex.
Stevens, Herbert Howe, Jr. ('45) (BCNOJA); Firm of H. H. Stevens, Jr., Cons. Research Engr., 1270—6th Ave., New York 20; *for mail,* 128 Charles St., New York 14, N.Y.
Stevens, Howard Elmore ('24; '29) (MDJCB), *Student Award,* '16; Assoc. Prof. Mech. Engrg., Rensselaer Poly. Inst.; *for mail,* 9 Rankin Ave., Troy, N.Y.
Stevens, John E. ('19; '35) (HFJSL), Mech. Engr., West. Elec. Co., Inc., 195 Broadway, New York; *for mail,* 4 Cedar Pl., Silver Beach, Bronx, N.Y.
Stevens, Lawrence Murat (J'39), 5688 Ash St., Los Angeles 42, Calif.
Stevens, Morris Hamilton (J'42) (ABE), Lt. (j.g.), U.S.N.R., Ammunition & Matl. Officer, c/o Comdr. Fleet Air Rocket Trng., Naval Air Sta., Quonset Point, R.I.
Stevens, Robert H., Jr. ('40) (SFKLCE), Steam Engr., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4; *for mail,* 185 W. Plumstead Ave., Lansdowne, Pa.
Stevens, Roy Resford ('44) (EHRAS), Indus. & Mar. Engr., Westinghouse Air Brake Co., Wilmerding; *for mail,* 108 Elmore Rd., Forest Hills, Pittsburgh 21, Pa.
Stevens, T. D. (J'40), Lt., Bn. Hdq., 302nd Engr. Combat Bn., A.P.O. 77, c/o Postmaster, Los Angeles, Calif.
Stevens, W. R. ('41) (MRS), Asst. Engr., Baltimore & Ohio R.R. Co., Rm. 609, B. & O. Bldg., Baltimore, Md.
Stevens, William D. (J'40) (EHS), Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail,* 456 Parker Ave., Hackensack, N.J.
Stevens, William James ('29; '35) (MZB), Assoc. Prof. Mech. Engrg., Drexel Inst. of Tech., 32nd & Chestnut Sts., Philadelphia 4, Pa.
Stevenson, Alexander Russell, Jr. ('27; F'41) (ACES), *Manager,* '42-'45, *Vice-President,* '45-'47; Staff Asst. to V.P. in Charge of Engrg. Policy, Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Stevenson, Gustie ('43) (NLJKMH), Ch. Design Engr., F. J. Stokes Mch. Co., Tabor Ave. above Adams Ave.; *for mail,* 581½ E. Alcott St., Philadelphia 20, Pa.
Stevenson, Walter Norman ('21) (NJBC), Partner, S. & S. Mch. Works, 4541 W. Lake St., Chicago 24; *for mail,* 921 Monroe St., Evanston, Ill.
Stevralia, Phillip F. (J'42) (MCB), Spec. Engr., Prod. Control Dept., Allis-Chalmers Mfg. Co., 1170 S. 70th St.; *for mail,* 1237 S. 113th St., West Allis 14, Wis.
Stewart, A. R. M. (J'40) (BEW), Mech. Engr., Draftsman, Powell River Paper Co. Ltd.; *for mail,* Box 582, Powell River, B.C., Can.
Stewart, Albert A. (J'32) (CMT), Mech. Engr., Hartford Ord. Dist., War Dept., 95 State St., Springfield; *for mail,* 28 Hanover St., Fall River, Mass.
Stewart, Arnold K. (J'41) (CGM), Asst. to Works Mgr., Am. Blower Corp., Box 58, Roosevelt Park Sta., Detroit; *for mail,* 213 Washington St., Marine City, Mich.
Stewart, Avery Lee (J'40) (ESMA), Mar. Engr., Naval Shipyard, Terminal Island, San Pedro; *for mail,* 1257½ Neptune St., Wilmington, Calif.
Stewart, Burt C. ('30) (SLEG), Mech. Engr., Smith, Hinchman & Grylls, Inc., 800 Marquette Bldg., Detroit 26; *for mail,* 647 Park St., Birmingham, Mich.
Stewart, Carlton Delos ('28; F'45) (RYC), V.P., Westinghouse Air Brake Co., Wilmerding, Pa.
Stewart, Clarence Russell ('25) (SJKFZL), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail,* 92 School St., Arlington 74, Mass.
Stewart, David W. ('22; '35) (CFS), Ch. Engr., New Orleans Pub. Serv., Inc., 317 Baronne St., New Orleans, La.
Stewart, Ethan Allen ('31; '35) (CFS), 31 Vernon Terrace, East Orange, N.J.
Stewart, Frank Y. (A'30) (BC), V.P., Engineering Index, Inc., 29 W. 39th St., New York 18; *for mail,* 35 May St., New Rochelle, N.Y.
Stewart, J. T. (J'33), Prod., W. C. Ritchie & Co., Baltimore St.; *for mail,* 7935 S. Manistee St., Chicago, Ill.
Stewart, James Arthur ('27) (SFEKLJ), Mech. Engr., Traveling Inspr., Air Preheater Corp., 121 S. Main St.; *for mail,* 30 Jefferson St., Wellsville, N.Y.
Stewart, James Clarence (J'45) (OL), Constr. Divisional Supvr., Carbide & Carbon Chemicals Corp., South Charleston 3, W.Va.; *for mail,* 33-19th Ave. N., Texas City, Tex.
Stewart, James German ('23; '35) (FLMS), Maint. Engr., MacAndrews & Forbes Co., 3rd & Jefferson Sts., Camden 4, N.J.
Stewart, James P. ('30) (EFARCL), Automotive Consultant, Research & Devel. Labs., Socony-Vacuum Oil Co., Inc., Paulsboro, N.J.
Stewart, James Pentland ('39) (CEMA), Spec. Rep., Sales Dept., De Laval Steam Turbine Co., Trenton 2, N.J.
Stewart, John A. ('29; '41) (AFS), Plant Engr., Derby Gas & Elec. Co.; *for mail,* Box 332, Derby, Conn.
Stewart, John Carson ('45) (CMS), Plant Engr., U.S. Cartridge Co., 4300 Goodfellow Blvd., St. Louis; *for mail,* 725 Clark Ave., Webster Groves, Mo.
Stewart, John T. ('41) (NMDRB), Engr., Anaconda Copper Min. Co., 25 Broadway, New York 4; *for mail,* 81-17—164th Pl., Jamaica 2, L.I., N.Y.
Stewart, M. G. (A'23) Pres., Pelican Well Tool & Supply Co., 901 McNeil St., Shreveport, La.
Stewart, Murray Douglas (J'39) (JS), Capt., Royal Canadian Elec. & Mech. Engrs., Canadian Army Overseas; *for mail,* 282 Glencairn Ave., Toronto 12, Ont., Can.
Stewart, Oswald, II (J'39) (YLMN), Staff, Engrg. Dept., Dewey & Almy Chem. Co., 62 Whittemore Ave., Cambridge 40, Mass.
Stewart, Randall Edward ('36) (SCFELH), Supvr., Engrg. Dept., Continental Asbestos & Refining Corp., 1 Madison Ave., New York 10, N.Y.; *for mail,* 137 Pasadena Pl., Hawthorne, N.J.

Stewart, Richard Waldorf (J'43) (AES), Mech. Engr., Eng. Testing, Aircooled Motors Corp., Liverpool Rd., Syracuse 8; *for mail,* 211 Comstock Ave., Syracuse 10, N.Y.
Stewart, Robert Addie ('27; '35) (P), Deputy Minister of Labor, Prov. of Man., 336 Legislative Bldg., Winnipeg, Man., Can.
Stewart, Ross R. ('44) (EHS), 863 Bush St., San Francisco 2, Calif.
Stewart, Russel Stoffer (J'43) (SKMEFJ), Mem. Survey Party, Bur. of Highways, State of Idaho; *for mail,* 430 Blue Lakes Blvd., Twin Falls, Idaho.
Stewart, Selden Lewis, II (J'32) (NBJHAE), Consultant, Mech. Engrg., 615 S.W. 2nd St., Miami 36, Fla.
Stewart, Spencer Wilson ('21) (HOS), Pres., Ambursen Dam Co., 295 Madison Ave., New York, N.Y.
Stewart, William Clayton ('37) (BNMJCA), Tech. Adviser, American Institute of Bolt, Nut & Rivet Manufacturers, 1550 Hanna Bldg., Cleveland 15, Ohio.
Stewart, William Fred ('35; '45) (ACBMNO), Ch. Engr., Spartan Aircraft Co., 1900 N. Sheridan Rd., Tulsa, Okla.
Stewart, Wilton Alexander (J'41) (M), 2994 W. 12th St., Los Angeles 6, Calif.
Steyskal, Raymond Francis (J'45), Elec. Draftsman, Kennecott Copper Co.; *for mail,* 316 Anza St., Hurley, New Mex.
Stickle, Harold E. ('41) (CFKS), Mech. Prod. Engr., Boston Edison Co., 39 Boylston St., Boston 12, Mass.
Stickley, Paul E. (J'32) (ZMJCB), Testing Engr., Aluminum Co. of Am.; *for mail,* 10 Forest Pl., Massena, N.Y.
Stickney, Wendell Howard (J'44) (CRS), Brownville, Me.
Stieg, Bernard Overton (J'39) (DNMJC), Ch. Engr., Atlas Conveyor Co.; *for mail,* 61 N. Main St., Clintonville, Wis.
Stieg, Robert Wilson (J'41) (MJNAC), Design Engr., Four Wheel Drive Auto Co.; *for mail,* 57 N. Clinton Ave., Clintonville, Wis.
Stiehl, Harry M. (J'24), Pres., W. N. Best Engrg. Co., Inc., 90 West St., New York, N.Y.
Stienmier, Harold (J'43) (DFCKZL), Asst. Mill Supvr., Empire Zinc Div., N.J. Zinc Co., Gilman; *for mail,* Box 161, Minturn, Colo.
Stiles, Edwin Milo ('39) (DHKJB), Ch. Engr., Consltd. Min. & Smelting Co. of Can. Ltd., Trail, B.C., Can.
Stillman, A. F. ('07; '12), Dir., Watson Stillman Co., Aldene, N.J.; *for mail,* Box 3, Rockledge, Fla.
Stillman, Guy (J'41), R.F.D., Barrington, Ill.
Stillman, Thomas Bliss ('13; '21; '35) (SFE RK), Engrg. Consultant, Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Stimler, Frederick Jacob (J'44) (ABHKOM), Aero. Research Engr., Daniel Guggenheim Airship Inst., 1300 E. Triplett Blvd., Akron 6; *for mail,* 1086 Norka Ave., Akron 7, Ohio.
Stimson, Glen H. (J'39), Gage Sales Engr., Greenfield Tap & Die Corp.; *for mail,* 225 Silver St., Greenfield, Mass.
Stine, Charles Everett ('43) (CBJMND), Pres., Ferguson Gear Co., Box 511, Gastonia, N.C.
Stine, Samuel Spencer ('27; '35) (ABCM), Mgr. Mfg., Westinghouse Elec. Corp., Lester; *for mail,* 176 Bryn Mawr Ave., Lansdowne, Pa.
Stinson, Karl Willson ('17; '26) (EAND), Prof., Ohio State Univ., Columbus 10, Ohio.
Stires, William H. ('19; '27), Mech. Engr., Taylor-Wharton Iron & Steel Co.; *for mail,* Box 242, High Bridge, N.J.
Stitt, Arthur Brown (J'41) (NKFMCD), Sales Engr., Armco Internatl. Corp., Curtis St., Middletown, Ohio.
Stitzer, Donald Joseph (J'41) (DMJH), Matl. Handling Engr., Remington Arms Co., Inc. (E. I. du Pont de Nemours & Co.), Ilion; *for mail,* 163 Wilber St., Utica, N.Y.
Stivers, Frank Orton (J'40) (HCEY), Engr., Humble Pipe Line Co., P.O. Drawer 2220, Houston 1, Tex.
Stix, Lawrence C. (J'11), 1809 Graybar Bldg., 420 Lexington Ave., New York 17, N.Y.
Stock, Arthur James ('26; '35) (FSNLDM), Pres., Stock Engrg. Co., 715 Hanna Bldg., Cleveland 15; *for mail,* 19690 Roslyn Dr., Rocky River 16, Ohio.
Stockham, George F. (J'41) (CDG), Ensign, Naval Proving Ground, Dahlgren, Va.
Stockus, Robert Louis (J'42) (SNKZM), Ch. Draftsman, Page Steel & Wire Div., Am. Chain & Cable Co.; *for mail,* 426 Donner Ave., Monessen, Pa.
Stoddard, Elgin ('11) (CKS), Pres., C. C. Moore & Co., 450 Mission St., San Francisco, Calif.
Stoddard, Marcus Wallace (J'40) (JC), Lt., U.S.N.R., DE-V(g), U.S.N.; *for mail,* 524 Lake Ave., Wilmette, Ill.
Stoeckel, Albert L. (J'37) (J), Ch. Engr., Stock Engr. Co., 715 Hanna Bldg., Cleveland 15, Ohio.
Stoecker, Rudolph John (J'41) (SMAEBC), Assoc. Mech. Engr., War Dept., Signal Lab., Camp Evans, Belmar, N.J.; *for mail,* 283 Hillside Ave., Williston Park, L.I., N.Y.
Stoeckinger, Richard Francis (J'40) (MJYA), Technician, 4th Grade, U.S.A., Co. M, 3rd Ord. Trng. Regt., Aberdeen Proving Ground, Md.; *for mail,* R.R. 2, Mishawaka, Ind.
Stoeckly, Eugene Emil ('44) (ABEKNS), Design Engr., Gen. Elec. Co., Lynn; *for mail,* 16 Lindbergh Rd., Marblehead, Mass.
Stoeltzing, Harry Everett ('16) (CFNS), Supt., Power Dept., N.Y. City Transit System, Bd. of Trans., 250 Hudson St., New York 13, N.Y.
Stoelzer, Walter Herman ('21; '23; '35) (SJK BEF), Engrg. Dept., Am. Gas & Elec. Serv. Corp., 30 Church St., New York 7; *for mail,* 148 Rodney St., Brooklyn 11, N.Y.
Stoessel, Robert Frederick ('34; '44) (ACOJM), Project Serv. Engr., Lockheed Aircraft Corp., Burbank; *for mail,* 362½ Pioneer Dr., Glendale 3, Calif.
Stoever, Alfred Christian ('42) (FKSZ), Supt., Power & Elec. Dept., Granite City Steel Co., 20th & Madison Sts., Granite City, Ill.
Stoever, Herman J. ('35; '37) (SKBFE), Prof. Mech. Engrg., Iowa State College, Ames, Iowa.
Stohldrier, Leonard Joseph (J'41) (JM), Tool Designer, Sunnen Products Co., 7900 Manchester St., St. Louis 17; *for mail,* 1507-A Obear St., St. Louis 7, Mo.
Stokes, Charles William ('44) (ESLBH), Tech. Consultant on Mchy., Cities Serv. Refining Corp.; *for mail,* 1311—11th St., Lake Charles, La.
Stokes, John Stogdell, Jr. (J'43), Spring Valley Farm, Huntingdon Valley, Pa.
Stokey, Wm. Farmer (J'38) (BM), 624 Rockmont Dr., N.E., Atlanta, Ga.
Stolberg, Emil C. ('17) (HKS), Gen. Improvement Engr., Am. Car & Fdy. Co., 30 Church St., New York, N.Y.
Stolfo, Vito Joseph (J'46) (ACY), Devel. Engr., B. F. Goodrich Co., 500 S. Main; *for mail,* 758 Carlysle St., Akron, Ohio.
Stoll, Charles Henry (J'42) (CJN), 1357 E. 141st St., East Cleveland, Ohio.
Stoll, Clarence Griffith ('18), Pres., West. Elec. Co., Inc., 195 Broadway, New York 7, N.Y.
Stoll, Karl Edward ('43) (CFSJ), Asst. Ch. Engr., Commonwealth Edison Co., Box 149, Pekin, Ill.
Stolle, Ernst Byron ('44), Mgr. Engrg., East. Div., Midwest Piping & Supply Co., Inc., 30 Church St., New York 7, N.Y.
Stolp, William John, Jr. (J'42) (JML), 1626 Fountain View, Charlotte, N.C.
Stolper, Walter Herbert (J'38) (HCMBNJ), Ch. Engr., Precision Mfg. Co., 2210 Temple St., Los Angeles; *for mail,* 5409 Russell Ave., Los Angeles 27, Calif.
Stoltz, Fred W. (J'37) (CJW), Plant Engr., Holly Mfg. Co., 875 Arroyo Pkwy., Pasadena; *for mail,* 686 Deninian Pl., Altadena, Calif.
Stolz, Paul Leo ('21) (D), V.P., Anderson-Stolz Corp., 1733 Walnut St., Kansas City 8, Mo.
Stone, Albert Rivington (J'43) (HRAJNY), Dir. of Research, Gerotor May Corp., Maryland Ave. & Oliver St., Baltimore 1; *for mail,* 705 Dunkirk Rd., Baltimore 12, Md.
Stone, Bert L. ('30) (CM), V.P., Gen. Mgr., Hydril Corp., 714 W. Olympic Blvd., Los Angeles, Calif.
Stone, Daniel Victor (J'42) (JCHDMN), Plant Engr., Chicago Mfg. Div., Revere Copper & Brass Inc., 2200 N. Natchez Ave.; *for mail,* 2371 N. Neva Ave., Chicago 35, Ill.
Stone, E. Wadsworth ('15; '18; '27) (OFTLJS), Cons. Engr., 216 Ellington Rd., Longmeadow 6, Mass.
Stone, Gerald Albert (J'45) (NKJ), Detail Draftsman, Link-Belt Co., 500 S. Latimer, Dallas 1; *for mail,* 3114 Holmes, Dallas 15, Tex.
Stone, Henry L. (J'34) (CS), Asst. to Ch. Engr., Mead Johnson & Co.; *for mail,* 1717 Russell Ave., Evansville, Ind.
Stone, James Harry (J'43), Lt., Co. C, 311th Engr. Combat Bn., A.P.O. 450, c/o Postmaster, San Francisco, Calif.
Stone, John Ransom ('19) (CMS), Chief, Mech. Serv., U.S. Penitentiary, Dept. of Justice; *for mail,* 1030 S. Broadway, Leavenworth, Kan.
Stone, Leonard ('19; A'25) (CGTW), Research Statistician, Am. Tel. & Tel. Co., 195 Broadway, New York 7, N.Y.
Stone, Mason Albert ('07; '21) (BS), Shelton Hotel, 49th St. & Lexington Ave., New York 17, N.Y.
Stone, Morris ('35; '35) Mgr., Devel. Dept., United Engrg. & Fdy. Co., 1st Natl. Bank Bldg.; *for mail,* 1023 Milton, Regent Sq., Pittsburgh, Pa.
Stone, R. A. ('45) (CDJT), Ch. Draftsman, Celanese Corp. of Am., 180 Madison Ave., New York; *for mail,* 26 Windsor Pl., Oceanside, L.I., N.Y.
Stone, Ralph Winship, Jr. (J'42) (ABMCNJ), Aero. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail,* Apt. 4, 206 Regent St., Hampton, Va.
Stoneburner, Clarence W. (J'38) (BLP), Lt., Box 807, 503—11th Ave. N., Texas City, Tex.
Stoner, John Henry (J'42) (ABZC), Engr., Wichita Div., Boeing Airplane Co., R.F.D. 5, Wichita 1; *for mail,* 323 S. Crestway Ave., Wichita 8, Kan.
Stopinski, Frank William, Jr. (J'45), Capt., U.S.M.C., 816 Conroy Rd., Birmingham 5, Ala.
Storey, Norman Choate ('18; '35) (HBY), 815 N.W. 72nd St., Miami 38, Fla.
Stork, Alfred Henry ('44) (JMS), 47-12—245th St., Douglaston, N.Y.
Storm, Elmer Ellsworth (J'45) (OGNMAC), Arch. Designer & Engr., 1241 W. 127th St.; *for mail,* 1250 W. 130th St., Los Angeles 44, Calif.
Storrier, James Mitchell (J'42) (MYJCZL), Product Devel. Engr., Bausch & Lomb Optical Co., 635 St. Paul St., Rochester 5; *for mail,* 1212 Park Ave., Rochester 10, N.Y.
Storrow, James J. ('21; '25; '35) (SR), Rm. 607, 80 Federal St., Boston 10, Mass.
Storrs, Robert Stedman ('23) (CMN), V.P., Torrington Mfg. Co., 70 Franklin St., Torrington, Conn.
Storto, Ettore (J'41) (HJOB), Engr., Pump Div., Fairbanks, Morse & Co., 600 S. Michigan Blvd., Chicago 5; *for mail,* 1142 Wisconsin Ave., Oak Park, Ill.
Stott, John Elwood (J'38) (CDMNJ), Ch. Engr., Wallaceburg Brass Ltd., Wallaceburg, Ont., Can.
Stotter, Arnold Harmon (J'44), 234 Atlantic St., Bridgeton, N.J.
Stouffer, Christian S. ('19) (SFDJN), Works Engr., Stanley G. Flagg & Co. Inc., Stowe; *for mail,* 1000 N. Charlotte St., Pottstown, Pa.
Stout, John D. ('21) (SCBJMH), V.P., Terry Steam Turbine Co., 874 Main, Hartford 1, Conn.
Stout, John William (J'41) (MKHCSF), Lt., U.S.N.R., Office of Inspr. of Naval Matl., U.S.N., 600 Bryant St., San Francisco 7; *for mail,* 585 Lincoln Ave., Palo Alto, Calif.
Stovall, Benjamin Lavender (J'44) (SFKZHC), Engrg. Asst., Instrument Engr., San Diego Gas & Elec. Co., 6th & E Sts., San Diego 12, Calif.
Stovel, Russell W. ('02; '07) (FKS), Valuation Engr., Pub. Serv. Comm., State of N.Y.; *for mail,* 1 Christopher St., New York 14, N.Y.
Stover, Alexander Jackson (J'42) (MCSLD), Cost Engr., E. I. du Pont de Nemours & Co., Belle; *for mail,* Apt. 2, 3512 Noyes Ave., Charleston 4, W.Va.
Stover, Howard Roy (J'39) (KBCJH), Design Engr., Stand. Oil Co. of Ind., 119th & Front Sts.; *for mail,* 2027 Davis Ave., Whiting, Ind.
Stowell, Howard Emery ('14; '21) (BDL), Effic. Dept., Carborundum Co., Box 337; *for mail,* 2447 Cleveland Ave., Niagara Falls, N.Y.
Stowers, Richard Frazier (J'45) (BFHNSZ), Ensign, U.S.N.R., ATA-189, F.P.O., San Francisco, Calif.; *for mail,* 2613 Bellevue St., Bluefield, W.Va.
Strachan, Adam Duncan ('44), Elevator Engr., Geo. T. McLauthlin Co., 120 Fulton St., Boston, Mass.
Strachan, Ben Wilson ('28; '35) (SLYC), c/o U.S. Rubber Co., 6600 E. Jefferson Ave., Detroit 32, Mich.
Strader, Roland H. ('15; '23) (CFSDEZ), Gen. Supt., Gas Prod. Dept., Astoria Plant, Consltd. Edison Co. of N.Y., Inc., 20th Ave. & 21st St., Long Island City 5; *for mail,* 710 Chauncey St., Brooklyn 7, N.Y.
Stradley, George Cleland, Jr. (J'37), Maint. Engr., E. I. du Pont de Nemours & Co., Deepwater, N.J.; *for mail,* 607 Concord Ave., Wilmington 270, Del.
Strahan, Charles Edward, Jr. (J'40) (EMSC AF), Draftsman-Engr., Engrg. Sec., Miss. Plant, LeTourneau Co. of Ga., Box 1031, Vicksburg, Miss.
Strahl, O. Robert (J'32) (NJMCD), Ch. Engr., Columbia Mch. Works, Inc., 255 Chestnut St., Brooklyn 8, N.Y.; *for mail,* 427—74th St., North Bergen, N.J.
Strait, John (J'38) (KSNF), Instr. in Agric. Engrg., Univ. of Minn., Univ. Farm, St. Paul 8, Minn.
Stranberg, Don Frederick (J'45) (HNYG), Asst. Ch. Engr., Anchor Coupling Co., Inc., 342 N. 4th St., Libertyville, Ill.
Strand, A. Thomas (J'41) Lt. (j.g.), U.S.N.R., Asst. to Ord. Officer, Carrier Aircraft Serv. Unit 21, F.P.O., New York, N.Y.; *for mail,* c/o Albert C. Strand, Ellendale, N.D.
Strandberg, Francis E. (J'39) (CJKMD), Ensign, U.S.N.R., 301st Naval Constr. Bn., F.P.O., San Francisco, Calif.
Strandhagen, Adolf Gustav (J'42) (BH), Asst. Prof. Mechanics, Carnegie Inst. of Tech., Schenley Pk., Pittsburgh 13, Pa.
Strang, Harold Edward ('44) (CSNZ), Gen. Elec. Co., Schenectady 5, N.Y.
Strange, Charles Alfred (J'44) (KHSBN), Lt. (j.g.), U.S.N.R., Op 23 F2, Rm. 4536, Main Navy Bldg., Washington, D.C.
Stransky, Malcolm Winston (J'44) (KNHBMS), Design Engr., Joshua Hendy Iron Works, Hendy Ave., Sunnyvale; *for mail,* 1162 Sherwood Ave., San Jose 11, Calif.
Strasser, Rolland John ('26) (SEFH), Partner, Sargent & Lundy, 140 S. Dearborn St., Chicago 3, Ill.
Strassman, Robert Charles (J'37) (CM), Partner, Badger Die Casting Co., 201 W. Oklahoma Ave., Milwaukee 7, Wis.
Strate, J. Taylor ('25; '33; '35) (SFE), Prof. & Head, Mech. Engrg. Dept., Colo. Agric. & Mech. College, Ft. Collins, Colo.
Stratmoen, Albert Nelson (A'43) (CJ), Staff Indus. Engr., Natl. Tube Co., 1721 Frick Bldg., Pittsburgh 30; *for mail,* 4722 Stanton Ave., Pittsburgh 1, Pa.
Strattan, William Voltz (J'46), Ensign, U.S.N.R.; 660 Elder Lane, Winnetka, Ill.

Stratton, Charles Brown (J'43) (OCMLHR), Staff Engr., Day & Zimmermann, Inc., 600 Packard Bldg., Philadelphia 2; *for mail,* R.D. 1, Glen Mills, Pa.
Stratton, John Alfred, Jr. (J'37) (SELZON), Asst. Foreman of Machinists, Charge of Refrigeration, Automatic Controls & Gages, Sun Shipbldg. & Dry Dock Co., Chester; *for mail,* R.D. 1, Glen Mills, Pa.
Stratton, John Charles (J'45) (JHCBJM), Student Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1, N.Y.; *for mail,* 1215 E. Gore Ave., Orlando, Fla.
Straub, Eugene D. (J'38) (CG), Lt., U.S.N.R., Tech. Editor, Attention SPOFM, Office of Chief of Ord., War Dept., Washington 25; *for mail,* 3244 Que St., N.W., Washington 7, D.C.
Straub, Lorenz George ('32; '35) (HBOZSA), Prof. Hyds., Head, Dept. of Civil Engrg., Dir., St. Anthony Falls Hyd. Lab., Univ. of Minn., Hennepin Island & 3rd Ave., S.E., Minneapolis 14, Minn.
Strauchen, David Meade ('25) (CDHJMN), Gen. Supt., Cincinnati Milling Mch. Co., 4701 Marburg Ave., Cincinnati 9, Ohio.
Straus, Ferdinand Nicholas ('44) (HKS), Austin Co., 2613 Queenston Rd., Cleveland Heights, Ohio.
Straus, W. Rayner ('13; '23) (CFEM), Mech. & Elec. Engr., City of Baltimore, 320 Mun. Bldg., Baltimore; *for mail,* 2305 South Rd., Baltimore 9, Md.
Strauss, Benjamin Joseph ('42) (CJMDYO), Sales Engr., Am. Can Co., 230 Park Ave., New York; *for mail*—109-29—201st St., Hollis 7, N.Y.
Strauss, Jerome ('36) (BJLR), V.P., Tech. Dir., Vanadium Corp. of Am., 420 Lexington Ave., New York 17, N.Y.
Strauss, John Gerrold (J'42) (NHYDMA), Corps of Engrs., U.S.A., Manila, P.I.; *for mail,* 1490 Hague Ave., St. Paul 4, Minn.
Strauss, Leopold (J'40) (ENBAS), Project Engr., Sterling Eng. Co., 1270 Niagara St., Buffalo; *for mail,* 204 Lancaster Ave., Buffalo 9, N.Y.
Strauss, Robert Wilbur (J'44) (CDSFY), Asst. Publisher, Maujer Publ. Co., 420 Main St., St. Joseph, Mich.
Strayer, Raymond K. ('26; '37) (BLM), Sales Mgr. & Engr., Lancaster Iron Works, Inc.; *for mail,* 833 Martha Ave., Lancaster, Pa.
Streamer, A. Camp ('43) (ACR), V.P., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30, Pa.
Strecker, Charles Edwin ('42) (BHMNKS), Design Engr., Ch. Draftsman, Certified Alloy Valve Co., Bloy St. & Ramsey Ave., Hillside 5; *for mail,* 139 Orchard St., Bloomfield, N.J.
Street, Clement F. ('93), Cons. Engr., St. Johns Inn, Kings Park, L.I., N.Y.
Street, Edward Tatum ('40) (BHJCYW), Ch. Engr., Downingtown Mfg. Co., Washington Ave., Downingtown, Pa.
Street, George Levick, Jr. ('16) (JMR), Pres., J. R. Johnson & Co., Inc., 2400 Maury St., Richmond 24, Va.
Street, Lockwood N. ('23; '35), Elec. Engr., Design & Constr., Nitrogen Div., Solvay Process Co., Hopewell; *for mail,* 1919 Matoax Ave., Petersburg, Va.
Streeter, Victor Lyle ('35; '40) (HKBA), Chmn., Fluid Mechanics & Thermodynamics Research, Armour Research Foundation, 35 W. 33rd St., Chicago 16, Ill.
Streett, John Walter ('43) (AEZKB), Project Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson 3, N.J.
Streid, Dale Downey ('36; '45) (ABKN), Engr., Gen. Elec. Co., 920 Western Ave., West Lynn 3, Mass.
Streit, Clarence T. (J'42) (NHOCD), Design Engr., Pines Engrg. Co. Inc., Highland Ave.; *for mail,* 932 Mountain St., Aurora, Ill.
Strenger, Stanley (J'43) (CMNKFZ), Ch. Engr., Joseph Strenger & Son, 2488 Grand Concourse, New York 58, N.Y.
Stresau, Richard H. F., Jr. (J'45), 3121 Plyers Mill Rd., Kensington, Md.
Stricker, Adam K., Jr. ('29; '41) (ACDMRS), Secy., Mfg. Policy Com., Gen. Motors Corp., 3044 W. Grand Blvd., Detroit 2; *for mail,* 1227 Bishop Rd., Grosse Pointe 30, Mich.
Strickland, Albert W. (A'43) (CDKS), Mech. & Elec. Engr., 402—3rd Ave., Big Timber, Mont.
Strickland, Bert ('28) (CEH), Prin. Elec. Engr., Rock Island Arsenal, Rock Island, Ill.; *for mail,* 1115 E. High St., Davenport, Iowa.
Strickland, Frank Howard ('44), Field Engr., Allis-Chalmers Mfg. Co., 1410 Waldheim Bldg., Kansas City, Mo.
Strickland, Wyatt Bailey (J'42) (HAB), Engrg. Dept., Wright's Automatic Mchy. Co., Calvin St.; *for mail,* 911—2nd St., Durham, N.C.
Strickler, Harrison Kennedy ('41) (AFJY), Pres., Gen. Mgr., Protane Corp., Powell Ave., Erie, Pa.
Strike, James Mathew ('41) (CLS), Gen. Supt., St. Joseph Ry. Light, Heat & Power Co., St. Joseph, Mo.
Strobell, Paul Norris (J'41) (ERFA), Ensign, U.S.N.R.; 499 Norton Pkwy., New Haven 11, Conn.
Strock, Clifford ('45) (KSFHBG), Editor, *Heating & Ventilating,* 148 Lafayette St., New York 13, N.Y.
Strohecker, Robert F. (J'44) (EFNY), 2508 E. Stratford Court, Milwaukee, Wis.
Strom, Russell L. (J'44), Elec. Engr., Charge of Matl., Kaiser Co., Inc., Richmond; *for mail,* 842 Teresita Blvd., San Francisco, Calif.
Stromgren, Charles D. ('42) (ABM), 935 W. 12th St., Los Angeles, Calif.
Stromm, Stanley M. (J'38) (BCD), 1053 N. Huey St., South Bend, Ind.
Strong, Harvey Waldo ('38) (CFKS), Sr. Assoc. Mech. Engr., Pub. Ltg. Comm., City of Detroit, 5425 W. Jefferson, Detroit 9; *for mail,* 16261 Wildemere Ave., Detroit 21, Mich.
Strong, Herbert Davis, Jr. (J'39) (ARCH), Lt. (j.g.), A-V(S), U.S.N.R.; 3437 Las Palmas Ave., Glendale 8, Calif.
Strothman, Edmund Pennell ('24; '45) (SB), Rm. 814, 310 S. Michigan Ave., Chicago 4, Ill.
Strott, John Fred (J'22) (DCNMB), Sales Mgr., Pacific Div., Link-Belt Co., 400 Paul Ave., San Francisco 24; *for mail,* 210 Palm Ave., Millbrae, Calif.
Stroud, Edwin G. ('01; '14) (OMLJCY), Pres., Cleveland Engrg. Agency Co., 2132 E. 9th St., Cleveland 15, Ohio.
Stroud, William Ernest (J'44) (NSCHAJ), Mech. Research Engr., Corning Glass Works, Corning, N.Y.
Strouse, Bernard H. ('30; '41) (KFSO), Propr., S. B. & B. H. Strouse, 307 Guarantee Trust Bldg., Atlantic City, N.J.
Strowe, Charles Gelston (J'45) (YZNLDJ), Devel. Engr., U.S. Rubber Co., Box 2589, Charleston 29; *for mail,* 4004 Venable Ave., Charleston 4, W.Va.
Strowger, Earl B. ('34) (HSBO), Hyd. Engr., Buffalo Niagara Elec. Corp., 535 Washington St., Buffalo 3, N.Y.
Stroyan, George S. (J'36) (HJM), Asst. Mech. Engr., Tenn. Valley Authority, Union Ave., Knoxville; *for mail,* 18 Florence St., Route 81, Knoxville 15, Tenn.
Struble, George Wallace ('17; '21) (CJMRN), Asst. to V.P., Bethlehem Steel Co., Bethlehem, Pa.
Struck, Henry W. ('18; '24) (SERAZF), Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail,* 91 Bay State Rd., Boston 15, Mass.
Struckmann, Holger, II ('30; '37) (ZKLCF), Box 5, Hartsdale, N.Y.
Strunk, Walter Cyrus ('19; '23) (FSRA), Stoker Engr., Westinghouse Elec. Corp., 40 Wall St., New York 5, N.Y.; *for mail,* Parkside Rd., Harrington Park, N.J.
Struthers, Keith James (J'38) (MACD), Sales Engr., Northwest Mill Supply, 4708 Armitage, Chicago 39; *for mail,* 2006 Jarvis, Chicago 40, Ill.
Strutz, Clarence Richard ('36; '41) (BNRJAG), Mech. Engr., Oxweld R.R. Serv. Co., 230 N. Michigan Ave., Chicago 1; *for mail,* 2726 W. 24th St., Chicago 8, Ill.
Stryker, Edgar Harvey (J'42) (CLSK), Engr., Shell Chem. Corp., Box 431; *for mail,* Box 120, Route 2, Pittsburg, Calif.
Stryz, Hillard Charles (J'43) (CDMN), Mech. Engr., Juricagal Div., Johnson & Johnson, 4949 W. 65th St., Chicago; *for mail,* 3733 W. 64th Pl., Chicago 29, Ill.
Stuart, George Norman, Jr. (J'38) (BCEFJK), V.P., Gen. Supt., X-Ray Inc., 4525—12th St., Detroit 8; *for mail,* 913 W. Oakwood, Royal Oak, Mich.
Stuart, James Carroll (J'44) (MB), Box 1341, Renton Highlands, Renton, Wash.
Stuart, Joseph, III (J'36) (ABON), 1022 Trenton Pl., Wilmington 16, Del.
Stuart, Kenneth Emmons ('19) (AL), Cons. Research Engr., Pat. Adviser, Hooker Electrochem. Co., Niagara Falls, N.Y.
Stuart, Milton Caleb ('12; '18; F'45) (SBEH), Prof. Mech. Engrg., Lehigh Univ., Bethlehem, Pa.
Stubau, Carl John (J'42) (ABZHK), Project Engr., Pesco Products Co., 11610 Euclid St., Cleveland 6; *for mail,* 13902 Castalia Ave., Cleveland 10, Ohio.
Stubbs, William Frederick ('21) (ZYKL), Engr., Indus. Instruments, Synthetic Rubber Div., Polymer Corp.; *for mail,* 340 N. MacKenzie, Sarnia, Ont., Can.
Stube, William Masten (J'37) (DJMC), Maint. Engr., Carborundum Co., Buffalo Ave.; *for mail,* 208—79th St., Niagara Falls, N.Y.
Stuber, Bernard Valentine (J'42) (AMCJOZ), Lt., U.S.N.R., Naval Air Test Center, Patuxent River, Md.
Stucki, Arnold ('07), Pres., A. Stucki Co., Oliver Bldg., Pittsburgh; *for mail,* 42 N. Howard Ave., Bellevue, Pa.
Studhalter, Walter Richard (J'45) (KFZEBJ), 2311—18th St., Lubbock, Tex.
Studley, Arthur Benjamin ('45) (TCMA), Asst. Mgr., Boston Office, SKF Industries, Inc., 10 High St., Boston 10, Mass.
Studley, Gerard L. ('24) (ACWY), Pres., Charles I. Allen, Inc., Pequabuck; *for mail,* S. Main St., Cheshire, Conn.
Studley, Gideon, Jr. ('18; '22) (SFEKLB), Steam Engr., Westinghouse Elec. Corp., 40 Wall St., New York, N.Y.; *for mail,* 42 Pine Grove Ave., Summit, N.J.
Stuebing, Albert F. ('17; '23) (RJBMC), Asst. to Mgr. of Sales, High Strength Steel Div., Carnegie-Ill. Steel Corp., 434—5th Ave., Pittsburgh 30, Pa.
Stuecheli, Ralph Bernhard (J'43) (CJMN), Asst. Ch. Tool Engr., Thompson Aircraft Products, Inc., 23555 Euclid Ave., Euclid; *for mail,* 5481 Dalewood, Maple Heights, Cleveland, Ohio.
Stueve, W. H. ('30) (ES), Commercial & Indus. Agt., Okla. Gas & Elec. Co., Box 1498, Oklahoma City, Okla.
Stuntz, John Edw. ('03; '16) (DRS), Pres., Antillian Constr. Co., 61 Obispo, Havana, Cuba.
Sturken, Robert C. (J'40) (L), Design Engr., E. I. du Pont de Nemours & Co.; *for mail,* 207 W. 34th St., Wilmington, Del.
Sturm, George William ('42; '44), 514 Ethan Allen Ave., Takoma Park, Md.
Sturm, Rolland George ('39) (BJR), Research Prof., Purdue Univ., West Lafayette, Ind.
Stutske, William Albert ('44) (BDHNOY), Design Engr., Corning Glass Works, Corning, N.Y.
Stutson, Alverd Chaney ('43) (ERSFJK), Tech. Div., Indus. Lub. Dept., Socony-Vacuum Oil Co., Inc., 26 Broadway, New York 4, N.Y.
Styerwalt, Alfred Jay (J'38) (ZSHANK), Mech. Engr., Saunders Bros., Inc., 112½ S. Pickering Ave.; *for mail,* 409 N. Hoover Ave., Whittier, Calif.
Styles, Thomas William (J'42) (HKNFBS), Assoc. Mar. Engr., Bur. of Ships, Navy Dept., 17th & Constitution Ave., Washington 25, D.C.; *for mail,* 702 Greenwood Ave., Takoma Park, Md.
Styri, Haakon ('25) (JBLNZ), Dir. of Research, SKF Industries, Inc., Front St. & Erie Ave., Philadelphia 34, Pa.
Suarez, Louis (J'37) (E), Inspr. of Engrg. Matls., Office of Inspr. of Naval Matl., U.S.N., 30 Church St., New York 7; *for mail,* 104-37—197th St., Hollis, N.Y.
Suarez, Luis Alejandro (J'41) (CZSNHD), Asst. Engr., United Fruit Sugar Co., Banes, Oriente, Cuba.
Suchy, George Fred ('45) (LBCMKH), Ch. Engr., Borden Co., 1171 Riverside Dr., Waukesha, Wis.
Suczek, Robert ('19) (BKS), Research Engr., Hudson Motor Car Co., Jefferson Ave., Detroit; *for mail,* 591 Fisher Rd., Grosse Pointe, Detroit 30, Mich.
Suda, Stanley (J'39), 20-46—29th St., Astoria 5, L.I., N.Y.
Sudduth, Henry Norton (J'30) (RS), Engr. of Tests & Devel., N.Y. Air Brake Co., Starbuck Ave., Watertown, N.Y.
Sudranski, L. L. (J'38) (CSYHFE), Plant Engr., Packard Elec. Div., Gen. Motors Corp., Warren, Ohio.
Sugrue, John Joseph (J'41) (SFK), Serv. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail,* 27 Lee Ave., New London, Conn.
Suhs, Guy Harold ('40) (SWLD), Supt. of Power, Sorg Paper Co., Manchester Ave.; *for mail,* 2709 Milton Rd., Middletown, Ohio.
Sulentic, Stanislav Albert ('42) (OE), Cons. Engr., 337 New England Bldg., Topeka, Kan.
Sulkin, Maurice Abraham (J'43) (KASNO), Aerodynamicist "A," N. Am. Aviation, Inc., Imperial Highway, Inglewood; *for mail,* 3885½ Denker Ave., Los Angeles 37, Calif.
Sullender, Maurice C. (J'42), U.S.N.R., Apt. 3, 2309 Chenevert, Houston, Tex.
Sullender, William A. ('34; '42) (CLMS), Asst. Head, Catalytic & Distillation Dept., Pan Am. Refining Corp., Texas City; *for mail,* Box 771, Lamarque, Tex.
Sulliger, Arthur Herman (J'39) (A), 1st Lt., Sqd. Engrg. Officer, 27th Bomb. Maint. Sqd., 2nd Air Force, Smoky Hill Army Air Field, Salina, Kan.
Sullins, Samuel L., Jr. (J'40), 706 Maple Ave., Fountain City, Tenn.
Sullivan, Edward L. ('20; '35) (EFS), Dir. of Maint., Boston Housing Authority, 10 Post Office Sq.; *for mail,* 564—5th St., Boston, Mass.
Sullivan, Francis J. (J'41) (ABG), Ensign, U.S.N., Naval Oper. Base, Norfolk, Va.
Sullivan, George Gale (J'35) (CMOD), Engr., Albert Ramond & Associates, Inc., Russ Bldg., San Francisco 4, Calif.
Sullivan, George Leonard ('20; F'41) (CMNKOE), Dean, College of Engrg., Univ. of Santa Clara; *for mail,* Box 447, 725 Madison St., Santa Clara, Calif.
Sullivan, Harry Hewes ('43) (CNMB), Pres., Treas., Paragon-Revolute Corp., 77 South Ave., Rochester 4, N.Y.
Sullivan, John Frederick, Jr. ('24; '34) (CSFBH), Asst. to V.P. in Charge of Serv. & Constr., Commonwealth Edison Co., 72 W. Adams St., Chicago 90, Ill.
Sullivan, Patrick J. (J'34) (CLS), Lt., Ord. Sec., Base Hdq., A.P.O. 600, New York, N.Y.
Sullivan, Raymond H. ('20) (CDM), 7020 Jeffery Ave., Chicago 49, Ill.
Sullivan, Robert James (J'43) (BCP), 7078 N. Wolcott Ave., Chicago 26, Ill.
Sullivan, Sylvester J. (J'35) (SFECK), Custodian Engr., Bd. of Education, New York City, 110 Livingston St., Brooklyn 2; *for mail,* 8924—89th St., Woodhaven 21, N.Y.
Sullivan, William Edward ('28; '32) (CKHMNE), Capt., U.S.N., Supvr. of Shipbldg., 11 Broadway, New York 4, N.Y.
Sullwold, Richard Hale (J'42), 350 E. Armour Blvd., Kansas City 3, Mo.
Sultzer, Norman W. ('41), Sales Mgr., N.Y. Dist., Research Corp., 405 Lexington Ave., New York, N.Y.

Sulzbacher, Joel, Jr. (J'40) (LMS), 206—4th Ave., Rome, Ga.
Suman, John R., Jr. (J'39) (E), Box 192, Columbia, Miss.
Summerfield, John R. (J'38) (CJM), Graduate Student, Univ. of Calif.; *for mail,* 1227-A University Ave., Berkeley 2, Calif.
Summerhays, Louis J. ('25) (SFOKW), Pres., Wm. Summerhays Sons Corp., 620 Clinton Ave. S., Rochester 7, N.Y.
Summerhays, Robert Louis (J'42), Ensign, U.S.N.R., 225 Alpine Dr., Rochester 10, N.Y.
Summerlin, I. W. ('28; '29), c/o R. C. Morrow, Sanatorio La Luz, Morelia, Michoacan, Mex.; *for mail,* Chapel Hill, N.C.
Summers, Bernard Deane (J'42), Dist. Supvr. Stoker Serv., Westinghouse Elec. Corp., 1299 Northside Dr., N.W., Atlanta, Ga.
Summers, James Harold Eugene (J'43) (DFMBRC), Jr. Engr., Drafting Rm., McNally Pittsburg Mfg. Corp.; *for mail,* 120 W. Forest St., Pittsburg, Kan.
Sumner, Henry W. ('00; '12), Retired; 2012—14th Ave. N., Seattle, Wash.
Sumner, Merton Rogers, Jr. (J'37) (S), Mar. Engr. with George G. Sharp, 30 Church St., New York; *for mail,* 45-41—39th Pl., Long Island City 4, N.Y.
Sumnicht, James Warren (J'44), 4357 Verdugo Rd., Los Angeles 41, Calif.
Sundell, Samuel Sigrud ('44) (JNCMD), Pres., SKG Corp., 1200 Chestnut, Minneapolis 3; *for mail,* 3040 Longfellow Ave., Minneapolis 7, Minn.
Sunderland, Richard Norman, Jr. (J'40) (CMLDZ), Stands. Engr., Newport Plant, E. I. du Pont de Nemours & Co., Newport, Del.
Sungur, Sabah (J'46) (ST), Sun-Or T.A.S., Bozkurt-Jeneral Han 15, Galata, Istanbul, Turkey.
Sunnen, Joseph ('37) (RCM), Pres., Sunnen Products Co., 7900 Manchester Rd., St. Louis 17, Mo.
Supove, Lawrence ('42; '43), 1439 S.W. Montgomery, Portland 1, Ore.
Suppe, Charles Arthur (A'41) (REMK), Draftsman, Am. Loco. Co., Schenectady 5, N.Y.
Suppes, Donald Frederick (J'42) (ABEC), Planning Engr., Wright Aero. Corp., Beckwith Ave., Paterson; *for mail,* 244 Teaneck Rd., Ridgefield Park, N.J.
Surdy, Charles J. ('41) (RFKS), Asst. to Pres., Stand. Stoker Co., Inc., Erie, Pa.
Surinak, John Joseph (J'45) (CMNDJL), Ch. Engr., Lake Shore Engrg. Co.; *for mail,* 1019 Kimberly Ave., Iron Mountain, Mich.
Surveyer, Arthur ('44) (OHCSL), Pres., Arthur Surveyer & Co., 1010 St. Catherine St. W., Montreal 2, Que., Can.
Suss, Lester Leon (J'43) (AES), Pvt., 624th Field Arty. Assn. Bn., Ft. Leonard Wood, Mo.
Sussdorff, Edmund L. ('23; '26) (CNMJSF), Mech. Engr., Pal Blade Co., Inc., 39 River St.; *for mail,* 54 Lorraine St., Plattsburg, N.Y.
Sussman, William (J'45) (ABC), Pvt., Army Air Forces, Sqd. 5, 4000th B.U., Wright Field, Dayton, Ohio; *for mail,* 1501 Undercliff Ave., New York 53, N.Y.
Sutcliffe, John William (J'39) (CMT), Tool Designer, Worthington Pump & Mchy. Corp., 87 Appleton St.; *for mail,* 17 Roosevelt Ave., Holyoke, Mass.
Suter John H. ('14) (ESH), Ch. Engr., West. Eng. Co. Div., Vernon Tool Co., Ltd., 1101 Meridian Ave., Alhambra, Calif.
Sutherland, Charles Richard ('45) (BNGJMO), Engr., Charge of Mech. Product Devel., Reliance Elec. & Engrg. Co., 1088 Ivanhoe Rd., Cleveland 10; *for mail,* 3382 Henderson Rd., Cleveland Heights 12, Ohio.
Sutherland, D. Manson (A '26) (W), Pres., Sutherland Refiner Corp., 143 E. State St., Trenton, N.J.
Sutherland, Edwin Keith (J'44), Mech. Research Div., Corning Glass Works, Corning, N.Y.
Sutherland, Howard John (J'44) (MSZW), 102-11—86th Rd., Richmond Hill 18, N.Y.
Sutherland, James Edward (J'38) (COMA), Exec. Engr., MacDonald Bros. Inc., 10 High St., Boston, Mass.; *for mail,* 350 Wallace Dr., Osborn, Ohio.
Sutherland, Kenneth W. ('19) (ES), Valuation Engr., New Rochelle Water Co., 29 Cottage Pl., New Rochelle; *for mail,* 188 Broadway, Dobbs Ferry, N.Y.
Sutherland, Richard Van Arsdale ('36) (JCM), Ch. Engr., Mch. Mfg. Div., Charles Bruning Co., 4700 Montrose St., Chicago 41, Ill.
Sutherland, Robert Alexander ('44) (HSNJ), Hyd. Engr., Ebasco Services Inc., 2 Rector St., New York 6; *for mail,* 56 Sprague Ave., Scarsdale, N.Y.
Sutherland, Robert Leslie ('44) (FSRE), Combustion Engr., Truax Traer Coal Co., 310 S. Michigan Ave., Chicago 4, Ill.
Sutherland, William Harrison (J'36) (ABH), Project Engr., Exper. Towing Tank, Stevens Inst. of Tech., 711 Hudson St.; *for mail,* 406 Hudson St., Hoboken, N.J.
Sutton, E. Clifton ('26) (CLD), Design Project Mgr., E. I. du Pont de Nemours & Co., Wilmington, Del.
Sutton, Edward W. (J'41) (HKS), 411 Elm St., Westfield, N.J.
Sutton, Frank Maynard (J'39) (AYMJ), Field Serv. Rep., Boeing Airplane Co.; *for mail,* 2004 N. Poplar, Wichita, Kan.
Sutton George P. (J'43) (AMHCZ), Instr., Calif. Inst. of Tech., Pasadena; Devel. Engr., Aerojet Engrg. Corp., Azusa, Calif. (*Use former address for mail.*)
Sutton, Harry Melvin ('19; '22) (COWMT), Sr. Partner, H. M. Sutton Engrs., Newton Centre Sta., Boston 59; *for mail,* 31 Westbourne Rd., Newton Centre, Mass.
Sutton, Richard Insley (J'42) (EMS), Mech. Design Engr., Austin Co., 16112 Euclid Ave., Cleveland 12; *for mail,* 2259 Noble Rd., Cleveland Heights 21, Ohio.
Sutton, Russell I. ('20; '35) (BNSJ), Plant Engr., Babcock & Wilcox Co., Stirling Ave., Barberton; *for mail,* 139 Harcourt Dr., Akron 2, Ohio.
Svensen, Carl Lars ('11; '19) (NGABS), Secy., Dir., Tex. State Bd. of Registration for Professional Engrs., 511 Ewell Nalle Bldg., Austin 22, Tex.
Svenson, Carl Louis ('20; '36) (SEKF), Assoc. Prof. Heat Engrg., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Svenson, Eric B. (J'39) (ACM), Inspr., Am. Can Co., 317 St. Paul Ave., Jersey City; *for mail,* 653 Mountain Ave., North Caldwell, N.J.
Svenson, Robert Henry ('22; '35) (LTDY), Plant Engr., Irvington Varnish & Insulator Co., 6 Argyle Terrace, Irvington 11, N.J.
Svenson, Robert Henry, Jr. (J'44) (CMS), Mech. Engr., Design & Constr., B. F. Goodrich Co., 500 S. Main St., Akron; *for mail,* 750 Canyon Trail, Akron 3, Ohio.
Swain, H. Dudley (J'37) (AJK), Design Engr., Chance Vought Aircraft Div., United Aircraft Corp., S. Main St.; *for mail,* 96 Minor Ave., Stratford, Conn.
Swain, Philip William (16; '20; '25) (SFOLZ), Editor, *Power,* McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York, N.Y.; *for mail,* Chapel Lane, Riverside, Conn.
Swain, Robert Onwood (J'46), 38 Hitchens Lane, Newport News, Va.
Swain, Wilbur Alfred ('18; A'24) (CMYS), Jenkins Bros., 80 White St., New York 13, N.Y.; *for mail,* 90 Evergreen Pl., East Orange, N.J.
Swan, John Joseph ('99; '09) (CM), Comptroller, Oper. Mgr., Barnard College, 607 W. 119th St., New York 27, N.Y.
Swan, S. R. B. ('36), Asst. Master of Engrg., South West Essex Tech. College, Forest Rd., Walthamstow, London 17, England.
Swan, Wallace Beech (J'41) (FEH), Engr. Asst. "A", Job Planner & Supvr. of Automotive Maint., So. Calif. Gas Co., 1401 E. Elm St.; *for mail,* 1909 E. Palmer Ave., Compton, Calif.
Swanberg, Edmund Raymond (J'43) (CM), Lt., Ord. Dept., U.S.A., Planning & Control Officer, Base Shop Div., Red River Arsenal, Texarkana, Tex.; *for mail,* 432 Prospect St., Norwood, Mass.
Swaney, Frank Ready, Jr. (J'41) (AMC), Exper. Engr., Chrysler Corp., 12800 Oakland Ave., Detroit; *for mail,* 261 Massachusetts, Highland Park 3, Mich.
Swannack, Jervis Daniel ('35; '46) (BEZ), Ch. Design Calculator, Fairbanks, Morse & Co., Lawton Ave.; *for mail,* 1103 Milwaukee Rd., Beloit, Wis.
Swanson, Carl Axel (J'43) (CMODYB), 7032 Justine St., Chicago 36, Ill.
Swanson, Harry Robert ('43) (LYKC), Pres., Petro-Chem. Devel. Co., Inc., 120 E. 41st St., New York 17, N.Y.
Swanson, John L. (J'45), Engr., Tech. Sec., Belle Works, E. I. du Pont de Nemours & Co., Belle; *for mail,* 11 Estill Dr., Charleston, W.Va.
Swanson, Leslie (J'39) (KSB), 9 Common St., Quincy 69, Mass.
Swanson, Lester V. ('44) (BCKOY), Fdy. Engr., Deere & Co.; *for mail,* 838—19th Ave., Moline, Ill.
Swanson, Maurice Carl (J'39) (REA), Engr., Diesel Loco. Div., Am. Loco. Co., Jay St., Schenectady, N.Y.
Swanson, Nils Evald ('41) (BJM), Project Engr., Hyatt Bearings Div., Gen. Motors Corp., Harrison, N.J.; *for mail,* 55 W. 11th St., New York 11, N.Y.
Swanson, Roger Howard (J'41) (ABMNJ), Stress Analyst, Lockheed Aircraft Corp., Burbank; *for mail,* 1850 Vista Del Mar, Hollywood 28, Calif.
Swanton, Harold R. (J'27) (ACM), Exec. V.P., Precision Bearings, Inc., 1706 S. Grand Ave., Los Angeles 15, Calif.
Swarr, Jay Nelson ('44) (ZLK), Control Engr., Chem. Div., Corn Products Refining Co., Argo; *for mail,* 9344 S. Richmond St., Evergreen Park, Ill.
Swarthout, Harry Putnam (J'40) (SELZTW), Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail,* 50 Jefferson Rd., Wakefield, Mass.
Swartwout, James Fletcher, Jr. ('37) (AEF), Analytical Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St.; *for mail,* 110 Chapel St., East Hartford 8, Conn.
Swartz, Martin D. (J'37) (NBZAJH), Ord. Engr., Frankford Arsenal, Bridge & Tacony Sts., Philadelphia 37; *for mail,* 1714 Pine St., Philadelphia 3, Pa.
Swedler, James William (J'42) (JMKBNC), Mech. Engr., Dept. 7833, West. Elec. Co., Inc., Chicago 23; *for mail,* 5727 W. 22nd Pl., Cicero 50, Ill.
Sweeney, Guy M. (J'39), Supt., Tool & Die Dept., Samuel Stamping & Enameling Co., Manufacturing Rd.; *for mail,* 128 Alden Ave., Chattanooga, Tenn.
Sweeney, Ronald James ('29; '35; '35) (BCJMNA), Ch. Engr., Stover Lock Nut & Mchy. Corp., Bushkill Dr., Easton, Pa.
Sweeney, Roy Lee (J'45) (LSY), Engr., Draftsman, Clinton Engr. Works, Tenn. Eastman Corp., Oak Ridge; *for mail,* 1900 W. Clinch Ave., Knoxville 16, Tenn.
Sweet, Charles E. ('07) (JMSH), Pres., Superior Mch. & Engrg. Co., 1930 Ferry Park, Detroit 8, Mich.
Sweet, Franklin ('03; '13) (NLMYKJ), Estimator, Engrg. Dept., Creamery Package Mfg. Co.; *for mail,* 500 S. Main St., Ft. Atkinson, Wis.
Sweet, Harold Maynard (J'43), Engr., Process Improvement, U.S. Engr. Dept., Oak Ridge, Tenn.; *for mail,* Box 394, 509—5th St., Spencer, N.C.
Sweet, William Harold (J'44), 4488 St. Catherine St. W., Montreal, Que., Can.
Sweet, William Lemuel (J'45) (JNMYC), Sales Engr., Magnavox Co. Ltd., Bueter Rd., Ft. Wayne; *for mail,* 1235½ Huestis Ave., Ft. Wayne 6, Ind.
Sweetland, Ernest J. ('17), Chmn., Bd. of Dirs., Oliver United Filters, Inc., 351 California St., San Francisco; *for mail,* 11 Glen Alpine Rd., Piedmont, Calif.
Sweetland, Ernest John, Jr. (J'43) (AHJ), Asst. Ch. Engr., Charge Exper. Work, Aireon Mfg. Corp., 166 W. Olive Ave., Burbank, Calif.
Sweigard, Jos. L. ('23), Sole Prop., Jos. L. Sweigard & Co., 1342 Lincoln-Liberty Bldg., Philadelphia, Pa.
Sweigert, Ray Leslie ('38) (EFKBSZ), Dir. Gen. Engrg., Ga. Sch. of Tech., 225 North Ave., N.W., Atlanta, Ga.
Swem, George Albert ('42) (HSBM), Engr., Permutit Co., 405 S. Hill St., Los Angeles 13; *for mail,* 1825 Milan Ave., South Pasadena, Calif.
Swengel, Franklin M. ('44; '45) (SEKFZR), Ch. Mech. Designer, Laramore & Douglass, Inc., 327 S. La Salle St., Chicago 4, Ill.
Swenson, Arthur Bernhart (J'41) (ACSO), Sr. Exper. Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 S. Main St., East Hartford; *for mail,* Apt. Cl, 83 W. Middle Turnpike, Manchester, Conn.
Swenson, Carl A. (J'42), 9 Baltusrol Pl., Summit, N.J.
Swenson, Carl O. ('42) (AEHKZ), Mech. Engr., New Devel., Oscar F. Carlson Co., 2600 W. Irving Park Rd., Chicago 18, Ill.
Swenson, Charles Herbert (J'43) (SKLFEC), Radio Technician, 3rd Class, U.S.N., U.S.S. *Grand Canyon* Det., Kirkland, Wash.; *for mail,* 310 Pacific Ave., Jersey City 4, N.J.
Swenson, John D. (J'46) (FKS), Asst. Prof., Civil Engrg. Dept., Mass. State College, Cambridge; *for mail,* 23 E. Pleasant St., Amherst, Mass.
Swenson, Karl Eklund (J'40) (NHBJ), Mech. Engr., Auxiliary Mch. Sec., Bur. of Ships, Navy Dept., Washington 25; *for mail,* Apt. 202, 2310—40th Pl., N.W., Washington 7, D.C.
Swenson, Leonard Kenneth (J'39), 1st Lt., Corps of Engrs., 15 Elliott St., New Haven, Conn.
Swenson, Maurice Eric (J'41) (SCRME), Field Engr., Allis-Chalmers Mfg. Co., Box 512, Milwaukee 1, Wis.; *for mail,* 3736—15th Ave. S., Minneapolis 7, Minn.
Swertloff, Sid. (J'37), 1035 Washington Ave., Brooklyn, N.Y.
Swetting, J. Rodney ('16; '26; '29) (OCMD), V.P., Charge of Engrg., Norris & Elliott, Inc., 85 E. Gay St., Columbus 15, Ohio.
Swift, Donald Charles ('44) (SOLFCA), Mech. Plant Betterment Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.
Swift, James Lester (J'43) (AMEF), Jr. Engr., Wright Aero. Corp., Paterson, N.J.; *for mail,* 1189 Forest Ave., West Brighton 10, N.Y.
Swift, Lewis B. ('29) (ZCLMY), Pres., Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Swinburne, Ralph E. ('27; '35), Applied Mechanics Teacher, Bd. of Education, 500 Park Ave., New York; *for mail,* 3518—168th St., Flushing, L.I., N.Y.
Switack, John Wayne (J'43) (ORBA), Ensign, U.S.N.R., Armament Test, U.S. Naval Air Sta., Patuxent River, Md.
Switzer, Frederick G. ('27) (HB), Div. Engr., N.Y. City Bd. of Water Supply, Rm. 1401, 120 Wall St., New York 5, N.Y.
Switzer, William P. (J'41), 50 Polk Rd., Menands, N.Y.
Sykes, Charles Samuel ('45), Dist. Engr., Gen. Elec. Co., 1405 Locust St., Philadelphia 2, Pa.
Sykes, Robert Elgie ('43) (CHKMSY), Dir. of Research, Elec. Sprayit Co., Illinois St.; *for mail,* 419 Clifton Ave., Sheboygan, Wis.
Sykes, Wilfred ('22) (BMS), Pres., Inland Steel Co., 38 S. Dearborn St., Chicago 3, Ill.
Sylvester, L. Arthur ('30) (CDJMO), Pres., Sylvester Co., 20 N. Wacker Dr., Chicago 6, Ill.
Symon, Maxwell Stanley ('33; '44) (CMLAFB), Sales Engr., De Mornay Budd, Inc., 475 Grand Concourse, New York 51; *for mail,* 180 E. 163rd St., New York 56, N.Y.

Symonds, Nathaniel Gardner ('05; '15) (LOZ), Consultant, Carl L. Norden, Inc., 80 Lafayette St., New York, N.Y.; *for mail* 208 S. La Salle St., Chicago 4, Ill.
Symonds, Ralph Frederick ('38) (BCDMA), Pres., New England Trawler Equip. Co., Eastern Ave., Chelsea; *for mail*, 248 Pleasant St., Marblehead, Mass.
Symons, John Jacob (J'41) (NBAHK), Design Engr., Maytag Co., 512 N. 4th Ave. W.; *for mail*, 310 E. 4th St. N., Newton, Iowa.
Syndennis, Elmer Joseph ('46), Mech. Engr., M. W. Kellogg Co., 225 Broadway, New York; *for mail*, 148 Daniel Low Terrace, New Brighton, S.I., N.Y.
Syrovy, Gerald Henri (J'43) (EASNFB), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 8—5th St., Scotia 2, N.Y.
Syska, Adolph G. ('18; '33) (EFHKOS), Partner, Syska & Hennessy, 144 E. 39th St., New York 16, N.Y.
Szekely, Ernest ('44) (BKHL), Pres., Gen. Mgr., Bayley Blower Co., 1817 S. 66th St., Milwaukee 14, Wis.
Szepan, Henry Frank (J'46) (CJM), 739 Page St., Manchester, N.H.
Sziklas, Endre ('26; '32; '35) (BLS), Design Engr., Solvay Process Co., Hopewell; *for mail*, 1718 Mt. Vernon St., Petersburg, Va.
Szmachlo, Walter (J'44) (ABE), 2nd Lt., Coast Arty. Corps, U.S.A.; 1719 Carrie St., Schenectady, N.Y.

T

Taber, George Aymar ('22) (ACHMS), Pres., Pennell, Dearborn & Hovey, Inc., 715 Washington St., Lynn; *for mail*, 1000 N. Main St., Reading, Mass.
Taber, George H., Jr. ('18; '30) (FECAL), Pres., Sinclair Refining Co., 630—5th Ave., New York 20, N.Y.
Tabshey, Frederick Peter (J'39) (HBN), Design & Product Devel., Waterbury Tool Co., E. Aurora St., Waterbury 91; *for mail*, 100 Sylvan Ave., Waterbury 19, Conn.
Tacchella, Adolph Amadeo ('14; '19) (ERFAKJ), Dist. Mgr., Busch-Sulzer Bros.-Diesel Eng. Co., 116 New Montgomery, San Francisco 5, Calif.
Tackett, James Bernard (J'44) (ABJNF), Lt. (j.g.), Draftsman A, Fuselage Dept., Douglas Aircraft Co., Inc., Santa Monica; *for mail*, 11264½ Mississippi Ave., West Los Angeles 25, Calif.
Tackett, Ray H. (J'38) (BK), 584 N. Wilson, Pasadena 4, Calif.
Taddiken, John F. ('07; '23), V.P., Gen. Mgr., Honolulu Iron Works Co., Box 3140, Honolulu, T.H.
Tafel, Robt. W. (J'35) (BER), Mech. Engr., Szekely Co., Inc., 1011 Chestnut St., Philadelphia; *for mail*, 115 Treaty Rd., Drexel Hill, Pa.
Taft, Theodore Howard ('03; '10) (SFEK), Prof. Emeritus, Mech. Engrg., Mass Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Tag, Walter ('15; '19; '24) (FHS), Mech. Engr., Office of Serv. Comd. Engr., War Dept., 270 Broadway, New York 7; *for mail*, 786 E. 38th St., Brooklyn 10, N.Y.
Taggart, James Moses ('44) (ELS), R.R. 1, Old Country Rd., Huntington, N.Y.
Tait, Gordon Edward (J'42) (JCKABM), 2nd Asst. Engr., Isthmian Steamship Co., 71 Broadway, New York, N.Y.; *for mail*, 283 Hillcrest Ave., Wood-Ridge, N.J.
Tait, Ralph Shannon ('21; '25; '35) (BMJ), Assoc. Prof. Mech. Engrg., Univ. of Kan., Lawrence, Kan.
Takahashi, Saburo (J'45) (BCMNOS), Mech. Engr., Meadows Critoph & Co., 118 Richmond St. W., Toronto; *for mail*, 234 Cottingham St., Toronto 5, Ont., Can.
Takata, Victor Taro (J'44) (BJLOR), Project Engr., Artisan Metal Products, Inc., 211 Congress St., Boston 10, Mass.
Takesuye, Thomas Tetsuro (J'45) (AKDBZN), Edo Aircraft Corp., Roosevelt Ave., Flushing; *for mail*, Apt. 17, 20 La Salle St., New York 27, N.Y.
Talbot, James M. ('13; '19) (CMLJZB), V.P., Charge of Prod., S. S. White Dental Mfg. Co., Prince Bay, S.I. 9, N.Y.
Talbot, Paul Abbott ('26) (AEFRS), Head, Firm of Paul A. Talbot, Empire State Bldg., New York 1, N.Y.
Talbourdet, Guy Julien ('40) (BMJHYZ), Mech. Engr., Research Div., United Shoe Mchy. Corp., Balch St., Beverly, Mass.
Talcott, Agnew A. (J'32) (EP), Tech. Salesman, Wm. S. Gray & Co., Inc., 342 Madison Ave., New York, N.Y.; *for mail*, Scott's Cove, Darien, Conn.
Talley, Samuel Kinter ('45) (BJM), Lub. Engr., Shell Devel. Co., 4560 Horton St., Emeryville 8; *for mail*, 791 Hilldale Ave., Berkeley 8, Calif.
Tallgren, Walter ('28; '35) (CDFKS), Asst. Engr., Luzerne County Gas & Elec. Corp., 247 Wyoming Ave., Kingston; *for mail*, 43 Willow St., Plymouth, Pa.
Tallmadge, Edward C. (J'38) (CMOW), Asst. Ch., Legal Branch, Office of Dist. Engr., Seattle Dist., Army Engrs., 1400 Textile Tower, Seattle 1; *for mail*, 9729 Dayton, Seattle 3, Wash.
Tallmadge, Webster ('43), 526 Park Ave., East Orange, N.J.
Tallman, Roscoe B. (J'40) (BCM), 702 Radcliff, University City, Mo.
Tallman, Wm. S. (A'19), V.P., Gen. Mgr., H. H. Robertson Co., 2400 Farmers Bank Bldg., Pittsburgh 22, Pa.
Talmage, Archibald Alexander (J'27) (CMOSEL), Exec., Elec. Bond & Share Co., 2 Rector St., New York 6; *for mail*, 25 Walworth Ave., Scarsdale, N.Y.
Talmage, Ralph Houston (J'37) (CMSEH), Col., Chem. Warfare Serv., U.S.A., Chief, Readjustment Div., Gravelly Point, Va.
Taney, Ahmed Raif (J'31) (COMLJN), Asst. Prototype Dept. Head, T. H. K. Aircraft Factory, Etimesgut, Ankara, Turkey.
Tang, Tsu-Zao (J'42) (LMYDCJ), Mech. Devel. Engr., U.S. Rubber Co., 355 Valley St., Providence 1, R.I.; *for mail*, Lane 363, 50 Gordon Rd., Shanghai, China.
Tangeman, Walter William (J'16), V.P., Cincinnati Milling Mch. Co., 4701 Marburg Ave., Cincinnati 9; *for mail*, 3450 Observatory Pl., Cincinnati 8, Ohio.
Tankle, Adolph (J'45) (MLDCSR), Ensign, U.S.N.R., Engrg. Officer, U.S.S. *Kearsarge*, CV-33, F.P.O., New York, N.Y.; *for mail*, 24 Stacy Ave., Trenton, N.J.
Tann, Walter Leslie ('34) (CMJDAY), Capt., U.S.N.R., Chief, Prod. Engrg., Bur. of Ord., Navy Dept., Washington 25; *for mail*, 3803 W St., S.E., Washington 20, D.C.
Tannehill, Vernon Louis (A'45) (BHL), Box 515, 4939 S. Lafayette St.; *for mail*, 312 W. Brackenridge St., Ft. Wayne, Ind.
Tanner, Frederick C. ('26; '33; '35) (CZMJNB), V.P., Gen. Mgr., Fed. Products Corp., 1144 Eddy St., Providence 1; *for mail*, 30 Bretton Woods Dr., Cranston 9, R.I.
Tanner, Henry Cranston ('24; '27; '35) (FKLRSZ), Dist. Mgr., Bailey Meter Co., 121 N. Broad St., Philadelphia 7, Pa.
Tanner, Hubert D. ('23), V.P., Asst. Gen. Mgr., Pratt & Whitney Div., Niles-Bement-Pond Co.; *for mail*, 37 Brunswick Ave., West Hartford, Conn.
Tanner, J. Roy ('20) (SBJ), Partner, Tanner & Arnold, 728 Gulf Bldg., Pittsburgh 19, Pa.
Tansil, Charles Louis, Jr. (J'45), Box 279, Nederland, Tex.
Tansley, L. R. ('35) (BCLMN), Dir. of Mfg., White Castle System Inc., 555 W. Goodale St., Columbus 16, Ohio.
Taplinger, Jean R. (J'33) (NMOLJ), Plant Engr. & Supt., Solo Products Corp., Cedar Lane; *for mail*, 181 Liberty Rd., Englewood, N.J.
Tapp, Harry F. ('23; '29) (CFMK), Mgr., Research & Devel. Dept., Gilbert & Barker Mfg. Co., West Springfield; *for mail*, 90 Maple St., Springfield 5, Mass.
Tapparo, John A. ('25; '43) (CMKJL), Plant Supt., Natl. Radiator Co.; *for mail*, 235 S. Lincoln Ave., Lebanon, Pa.
Tarallo, D. Richard (J'40) (CALMYW), Assoc. Consultant, Kenneth A. McIntyre Associates, 2 Rector St., New York, N.Y.; *for mail*, 213 McElroy Ave., Palisades, N.J.
Tarantino, Anthony Francis (J'45), Jr. Test Engr., Ranger Aircraft Engs. Div., Fairchild Eng. & Airplane Corp., Farmingdale, L.I.; *for mail*, 1060 Morris Park Ave., Bronx 61, N.Y.
Tarashik, Nicholas (J'38) (BCMJYS), Capt., Ord. Dept., U.S.A., Harbor Defense Officer, Ft. Williams, Cape Cottage, Me.
Tarbox, John William (J'42) (ACF), Capt., Canadian Army, Fire Control Inst. Specialist, E-4, D.M.E., N.D.H.Q., Ottawa, Ont., Can.
Tardi, Joseph Thomas (J'43) (AM), Mech. Equip. Engr., Behr-Manning Corp., Watervliet; *for mail*, 1513 Bouton Rd., Troy, N.Y.
Tarne, Chester A. (J'42) (MCA), Gp. Leader, Tool Engrg., Dodge Chicago Plant Div., Chrysler Corp., 7401 S. Cicero Ave., Chicago 80; *for mail*, 1811 S. 51st Ave., Cicero 50, Ill.
Tarnowski, Chester A. (J'42), 2327 S. Scoville Ave., Berwyn, Ill.
Tarr, Levi L. (J'36) (CJP), Assoc. Engr., Phillips Petroleum Co.; *for mail*, 1737 Keeler, Bartlesville, Okla.
Tarrson, Irvin Saul (J'43) (NOMBJC), Mch. Engr., Intercontinental Engrs. Inc., 176 W. Adams St., Chicago 3; *for mail*, 1744 W. Roosevelt Rd., Chicago 8, Ill.
Tarwater, J. L. (J'39) (T), Gen. Mgr., Harriman Hosiery Mills, Harriman, Tenn.
Tarzy, Albert Richard (J'45) (FKSYE), Lab. Asst., Babcock & Wilcox Co., Bayonne; *for mail*, 6506 Dewey Ave., West New York, N.J.
Taskin, Haldun Ahmet, Koca (J'43) (EKS), Asst. Prof., Charge Mech. Engrg. Lab., Marquette Univ.; *for mail*, Apt. 4, 1310 W. Wells St., Milwaukee, Wis.
Taspinar, Adnan Halet ('37) (LCMHEN), Cons. Engr., Box 1609, Istanbul, Turkey.
Tate, Benjamin E., Jr. ('33) (SFEK), Ch. Engr., Power Plant, Natl. Cash Register Co., Main & K Sts., Dayton, Ohio.
Tate, George Harold ('39) (CSFKZB), Power Supt., Canadian Kodak Co. Ltd., Toronto; *for mail*, Holland Landing, Ont., Can.
Tate, Malcolm C. (J'29), Mech. Engr., A. H. Emery Co., 682 Main St., Stamford, Conn.
Tate, Matlock K. ('43) (OR), Asst. to Pres., Lima Loco. Works Inc., 60 E. 42nd St., New York, N.Y.; *for mail*, Wardman Park Hotel, Washington, D.C.
Tate, Michael G. ('39), Imperial Chem. Industries Ltd., 521—5th Ave., New York 17, N.Y.
Tate, Thomas Rouse ('26) (SHFEAO), Wash. Rep., Charles T. Main, Inc., 201 Devonshire St., Boston 10, Mass.; *for mail*, 314 Denrike Bldg., 1010 Vermont Ave., N.W., Washington 5, D.C.
Tatersall, Leo ('24) (SC), Ch. Engr., James H. Reed Power Sta., Duquesne Light Co., 435—6th Ave., Pittsburgh 19; *for mail*, 410 McCully St., Pittsburgh 16, Pa.
Tatge, Robert James (J'43) (ADFHJK), Mech. Engr., Aviation Gas Turbine Serv. Dept., Westinghouse Elec. Corp., Box 7348, Philadelphia 1, Pa.
Tatlow, Richard Henry, III ('44) (OMCD), V.P., Abbott, Merkt & Co., 10 E. 40th St., New York, N.Y.; *for mail*, 8 Virgilia St., Chevy Chase 15, Md.
Tatman, Jas. S. ('27) (ACD), Pres., Gen. Mgr., Roots-Connersville Blower Corp., Connersville, Ind.
Tattersfield, James Peyton ('44) (SEFCK), Gen. Mgr., Babcock & Wilcox de Mex., S. A., Apt. 416, Ave. Ejido 7, Mexico, D.F., Mex.
Taub, Paul H. (J'39) (CM), Indus. Engr., Louis Lefkowitz & Bros. Inc., 25 Water St., New Brunswick, N.J.
Taul, Horace Wayne ('45) (HSKORB), Lt. Col., A.U.S., Dir. of Sr. Subjects, U.S. Military Acad., Quarters 277, West Point, N.Y.
Taurman, Alphonso ('16), Supt., Equip. & Struc., Birmingham Elec. Co., 2100—1st Ave.; *for mail*, 848 S. 41st St., Birmingham, Ala.
Tausch, John Henry (J'41) (NHBCA), Engrg. Supvr., Quincy Shipyard, Bethlehem Steel Co., E. Howard St., Quincy 69; *for mail*, 94 Elm Ave., Wollaston 70, Mass.
Tauth, George (J'43) (GYTLW), Chem. Research Engr., Armour Research Foundation, Ill. Inst. of Tech., 1800 S. Prairie Ave., Chicago 16; *for mail*, 6415 Parnell Ave., Chicago 21, Ill.
Tautz, Herbert Eric ('22; '27; '35) (CMNZ), Pres., Gen. Mgr., Arapahoe Mfg. Co.; *for mail*, 4150 S. University Blvd., Englewood, Colo.
Tawresey, John S. ('20; '26; '35), V.P., Charge Engrg., Bunting Brass & Bronze Co., 713 Spencer St., Toledo 9, Ohio.
Taylor, A. W. ('33) (FS), Power Supt., Stand. Oil Co. of N.J., Boston & Eaton Sts.; *for mail*, 1217 Round Hill Rd., Baltimore, Md.
Taylor, Albert Merrick (J'44), 1306 E. Calaveras St., Altadena, Calif.
Taylor, Arthur Hobart ('46), 13695 Sunset Blvd., Pacific Palisades, Calif.
Taylor, Benjamin William (J'22) (RBENY), Asst. Ch. Mech. Engr., Gen. Steel Castings Corp., Eddystone; *for mail*, 684 Chester Rd., Swarthmore, Pa.
Taylor, Carl Warren ('43) (OSFK), Secy., Westcott & Mapes, Inc., 109 Church St., New Haven 5, Conn.
Taylor, Clarence E. ('39), Ch. Engr., Del. Power & Light Co., 600 Market St., Wilmington, Del.
Taylor, DeWitt M. ('14; '20) (SEFKL), Mech. Engr., Jackson & Moreland, Park Sq. Bldg., 31 St. James Ave., Boston 16; *for mail*, 117 Pleasant St., Newton Centre 59, Mass.
Taylor, Edward Carrier ('23; '39) (FS), Pres., Whitty Co., Inc., 86 Western Ave., Boston 34, Mass.
Taylor, Edward George Towle ('39) (DKS), Pres., Taylor Engrg. & Constr. Co. Ltd., 80 Richmond St. W., Toronto 1, Ont., Can.
Taylor, Edward Hall ('41) (BCH), V.P., Taylor Forge & Pipe Works, Box 485, Chicago 90, Ill.
Taylor, Edward Story ('45) (ABE), Prof. Aero. Engs., Mass. Inst. of Tech., Cambridge 39; *for mail*, Trapelo Rd., South Lincoln, Mass.
Taylor, Ernest H. ('22) (FKS), V.P., Gen. Mgr., Internatl. Boiler Works Co., East Stroudsburg; *for mail*, 500 Sarah St., Stroudsburg, Pa.
Taylor, Frank Augustus ('42) (EKS), Capt., Coast Arty. (AA), 6605—32nd St., N.W., Washington 15, D.C.
Taylor, Fred Wilson (J'39) (CJMO), Cost Engr., Ch. Cost Accountant, J. P. Seeburg Corp., 1510 Dayton St.; *for mail*, 2414 Gladys Ave., Chicago 12, Ill.
Taylor, George Albert ('44), Project Engr., Babbock & Wilcox Co., 19 Rector St., New York, N.Y.; *for mail*, 171 N. Maple Ave., East Orange, N.J.
Taylor, George C. ('28) (SCFERA), Port & Steam Engr., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4, Pa.
Taylor, George Osborn ('32; '35) (CMO), Prod. Supvr., Matériel Comd., Army Air Forces, 20 N. Wacker Dr., Chicago 6; *for mail*, 1612 S. 58th Court, Cicero 50, Ill.
Taylor, George R. (J'35) (ELS), Hewitt Rubber Corp., 240 Kensington Ave., Buffalo, N.Y.
Taylor, Gordon Watson ('25; '32; '35) (EFJR), Serv. Engr., Loco. Finished Matl. Co.; *for mail*, 1105 Riverview Dr., Atchison, Kan.
Taylor, H. Birchard ('10; '13; F'27) (HCBES), *Vice-President*, '24-'25; V.P., Cramp Shipbldg. Co., Richmond & Norris Sts., Philadelphia 25; *for mail*, The Barclay, 18th & Rittenhouse Sq., Philadelphia 3, Pa.

Taylor, Hamilton Dana ('41) (BSKJ), Asst. Designing Engr., Turbine-Generator Engrg. Div., Bldg 56, Gen. Elec. Co., Schenectady, N.Y.
Taylor, Irving ('40; '44) (AHLSEZ), Pump Engr., Lummus Co., 420 Lexington Ave., New York 17; *for mail*, 437 Melbourne Ave., Mamaroneck, N.Y.
Taylor, J. Hall ('12), Pres., Taylor Forge & Pipe Works, Box 485, Chicago 90, Ill.
Taylor, James Walton ('42) (BCHN), Lt. Comdr., U.S.N.R., Asst. Supvr. Engrg., Naval Ord. Test Sta. (Inyokern), 1030 E. Green St., Pasadena, Calif.
Taylor, Jerome J. (J'43) (BCHKN), Elec. Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Taylor, Jesse, Jr. ('40) (SKRAFO), Steam Supt., Engrg. & Serv., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4, Pa.
Taylor, John Irvin (J'43) (SFKZ), Mech. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio; *for mail*, Box 843, Syracuse, N.Y.
Taylor, John O. ('21) (SWDM), Cons. Engr., Kroehler Mfg. Co., Naperville; *for mail*, 524 Wenonah, Oak Park, Ill.
Taylor, John Paul (J'40) (ZLSFCD), Editor & Publisher, 520 Pleasant St., St. Joseph, Mich.
Taylor, John Wallace ('09; '10) (BJO), Cons. Engr., 2714 Winslow Ave., Cincinnati 6, Ohio.
Taylor Joseph George ('29) (DHGL), Editor, Aluminum Co. of Am., 801 Gulf Bldg., Pittsburgh 19; *for mail*, 440 Olympia Rd., Pittsburgh 11, Pa.
Taylor, Kenneth H. (J'40), Mar. Engr., U.S. Merchant Mar.; 202 Gramercy Pl., Glen Rock, N.J.
Taylor, Leonard Clark (J'40) (BDM), Jr. Engr., Watson-Flagg Mch. Co., Paterson; *for mail*, 21 Western Ave., Butler, N.J.
Taylor, Leonard Harold (J'45) (BJM), Mfg. Engr. in Trng., Am. Can Co., 511 Railway St.; *for mail*, 2456 W. 6th, Vancouver, B.C., Can.
Taylor, Marion Mackenzie (J'45) (A), P-1 Engr., Bur. of Aero., Navy Dept., 18th & Constitution Ave.; *for mail*, Y.M.C.A., 18th & G St., Washington 6, D.C.
Taylor, Mark Albert (J'43) (A), Engr., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, Clifton Heights Lane, Clifton, Mass.
Taylor, Mark Herschel ('42) (ZMC), Managing Dir., Taylor, Taylor & Hobson Ltd., Stoughton St. Works, Leicester, England.
Taylor, Morris P. ('27; '37), Sr. Engr., Joshua Hendy Iron Works, Sunnyvale; *for mail*, Box 524, Stanford University, Stanford, Calif.
Taylor, Reese H. ('25; '31; '35) (CEFY), Pres., Union Oil Co. of Calif., 617 W. 7th St., Los Angeles 14, Calif.
Taylor, Richard Milton ('34; '44) (DMCSRA), Plant Layout Engr., Westinghouse Elec. Corp., 2519 Wilkens Ave., Baltimore 23; *for mail*, 2519 Ellamont St., Baltimore 16, Md.
Taylor, Robt. M. ('17; '26), Distributor, Power Transmission Mchy., 211 Vine St., Cincinnati, Ohio.
Taylor, Ross W. G. ('39) (BHS), Asst. Engr., Hydro-Elec. Power Comm. of Ont., 620 University Ave., Toronto; *for mail*, 82 Glen Echo Rd., Toronto 12, Ont., Can.
Taylor, Roy Milton (A'22) (CBJ), Exec., Am. Cyanamid Co., 30 Rockefeller Plaza, New York 20, N.Y.; *for mail*, 68 Orchard Dr., Greenwich, Conn.
Taylor, Thomas Smith (J'40) (AZSB), Power Devel. Engrg. Dept., Bell Tel. Labs., 463 West St., New York 14, N.Y.; *for mail*, 124 E. Ft. Lee Rd., Bogota, N.J.
Taylor, Vernon (J'39) (CS), 42 Harbor Dr., Hampton, Va.
Taylor, Walter Stewart (J'42) (AMBYEC), 5624 Hampton St., Pittsburgh 6, Pa.
Taylor, Warren Franklin (J'43) (CJM), Asst. Mar. Engr., U.S. Maritime Comm., 310 S. Michigan Ave., Chicago 4, Ill.; *for mail*, 230 Rose St., La Crosse, Wis.
Taylor, William (J'41) (ABME), Maj., Army Air Forces, Aircraft Maint. & Repair Officer, Army Air Base, Grand Island, Neb.; *for mail*, 70—15th Ave., Columbus 1, Ohio.
Taylor, William Franklin ('40; '45) (YCMD), Mech. Supt., Duplate Can. Ltd., Oshawa, Ont., Can.
Taylor, William Mode ('88; '04) (CS), Sales Mgr., Maluminum Co., Trans. Bldg., Indianapolis 4; *for mail*, 124 W. 41st St., Indianapolis 8, Ind.
Teaf, John H. ('25; '32; '35) (CJLMNZ), Asst. Ch. Engr., Radio Condenser Co., Davis & Copewood Sts., Camden; *for mail*, 37 Bala Rd., Colwick, Merchantville, N.J.
Teague, H. M. ('25; '27; '35) (ABM), Product & Test Engr., Primary Battery Div., Thos. A. Edison, Inc.; *for mail*, 42 Park Ave., Bloomfield, N.J.
Teague, Robert L. (J'42) (CJOH), Sales Rep., Am. Cast Iron Pipe Co., Box 1491, Dallas 1, Tex.
Teal, F. Martin (J'43) (WHBOMY), Field Engr., Merritt Engrg. & Sales Co.; *for mail*, 82 Niagara St., Lockport, N.Y.
Teasley, Gladstone Isham ('45) (HJM), Assoc. Matls. Engr., Tenn. Valley Authority, Knoxville; *for mail*, 4214 Alma Ave., Knoxville 15, Tenn.
Teaze, Moses Hay ('21) (HSDFWO), Partner, Hardy S. Ferguson & Co., 200—5th Ave., New York 10, N.Y.; *for mail*, 31 Clarendon Pl., Bloomfield, N.J.
Teberg, Daniel Ernest (J'42) (MJYCS), Maj., Hdq., Army Air Forces, AC/AS-2, AFBIN-DA, Washington 25, D.C.
Tector, Albert David ('30; '35) (CM), Supt., Canadian Johns-Manville Co., Ltd.; *for mail*, Coulston Ave., Asbestos, Que., Can.
Tedrow, George E. (J'44) (DJW), Box 391, Route 3, Bellevue, Wash.
Teed, Ralph Herman ('29; '35) (CES), V.P., Gen. Mgr., Citizens Elec. Co., Box 576, Hot Springs, Ark.
Teele, William Brown (J'42), Seaman 1st Class, U.S.N.R., Bur. of Aero., Navy Dept., Maint. Rep. Office, Bamro, Bldg. 633, Rm. 165, Naval Air Sta., Pensacola, Fla.
Teeling, Geo. A. ('39) (BEF), Cons. Engr., 1 Columbia Pl., Albany, N.Y.
Teeter, Phillip H. (J'46), Lt. Comdr., U.S.N.R., Gen. Line Sch., Naval Oper. Base, Newport, R.I.
Teetor, Ralph R. ('13; '21; '35) (CF), Pres., Perfect Circle Co., 552 S. Washington St., Hagerstown, Ind.
Teichman, William Alexander (J'42) (YHJL), Capt., Hdq. & Hdq. Det., 3rd QM. Bn. (M), A.P.O. 777, c/o Postmaster, New York, N.Y.
Teichmann, Frederick Kurt ('29; '41) (AB), Chmn., Dept. of Aero., College of Engrg., N.Y. Univ., University Heights, Bronx 53, N.Y.
Teichmann, Henry Fred ('16; '26) (ONMJCD), V.P., Forter-Teichmann Co., 610 Smithfield St., Pittsburgh 22; *for mail*, 114 North Ave., Washington, Pa.
Teichmann, Otmar Erich ('45) (HKNBAZ), Research Engr., Fluid Mechanics, Armour Research Foundation, 35 W. 33rd St., Chicago 16; *for mail*, 5116-A Kimbark Ave., Chicago 15, Ill.
Teitelbaum, Arthur (J'40), Ensign, Torpedo Sqd. 8, F.P.O., San Francisco, Calif.; *for mail*, 514 Irvington Ave., Elizabeth 3, N.J.
Telle, Martin Jerome (J'41) (CNMBZO), Devel. Engr., Goodyear Aircraft Corp., Akron 15; *for mail*, 640 W. Market St., Akron 3, Ohio.
Teller, S. Jay ('21) (CJBA), Counsel, Colt's Pat. Fire Arms Mfg. Co., 17 Van Dyke Ave., Hartford; *for mail*, 28 Cumberland Rd., West Hartford 7, Conn.
Tellett, David Parker (J'45) (CLMO), Student Engr., Allis-Chalmers Mfg. Co., Milwaukee; 931 N. 10th St., Milwaukee 3, Wis.
Tellis, Vsevolod Gregory ('26) (R), System Bldg., Supvr., Chicago, Rock Island & Pacific Ry. Co., 139 W. Van Buren St., Chicago 5, Ill.
Temple, Edward Arthur ('45) (MCDJBN), Mfg. Planning Engr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore 24; *for mail*, 2902 Dunran Rd., Baltimore 22, Md.
Templeton, Phil C. (J'37) (KLP), Chem. Engr., Processing Dept., Tex. Co., 1 Havoline St., Lawrenceville, Ill.
Templin, Richard Laurence ('40) (BJNALZ), Asst. Dir. of Research, Ch. Engr. of Tests, Aluminum Co. of Am., Box 772, New Kensington, Pa.
Tenety, James, Jr. (J'40) (ALBHKZ), Lt. (j.g.), U.S.N.R., Asst. Ord. Officer, c/o Supvr. of Shipbldg., Fed. Shipbldg. & Drydock Co., Kearny, N.J.; *for mail*, 34-15—94th St., Jackson Heights, L.I., N.Y.
Tenney, Albert Ball ('96; '04) (Y), 3 Joy St., Boston, Mass.
Tenney, Ashton M. (A'19), A. M. Tenney Associates, 171 Madison Ave., New York; *for mail*, 15 Oakdale Ave., New Rochelle, N.Y.
Tenney, Edward Henry ('12) (SFKHNC), Ch. Engr., Steam Power Plants, Union Elec. Co. of Mo., 315 N. 12th Blvd., St. Louis 1, Mo.
Tepe, John Bright (J'44) (LKCB), Chem. Research Engr., Experimental Sta., E. I. du Pont de Nemours & Co., Wilmington, Del.
Tepel, Richard Walter (J'43) (BHM), 7462 Schoyer Ave., Pittsburgh 18, Pa.
Teree, Baboo Ram ('45) (AHBKMC), Asst. Head, Structures Dept., Cornell Aero. Lab., Cornell Research Foundation, Inc., Buffalo 5; *for mail*, 15 Linden Ave., Kenmore 17, N.Y.
Ter Haar, James Roger (J'44) (BHKLEG), Asst. Mgr., J. Arthur Moore Co., 342 Madison Ave., New York 17; *for mail*, 70 De Mott Ave., Baldwin, Nassau Co., N.Y.
Terpenny, Gordon E. (J'37) (ACE), Mech. Engr., Army Air Forces, Hdq., Air Tech. Serv. Comd., Wright Field, Dayton, Ohio; *for mail*, 529 Altgeld Rd., South Bend 14, Ind.
Terpo, Christopher Thomas (J'44) (CJM), Grinding & Sales Engr., Norton Co., New Bond St., Worcester 6; *for mail*, 15 Clason Rd., Worcester 5, Mass.
Terrell, Edgar A. ('30), Pres., Treas., Terrell Mch. Co., 1200 N. Church St., Charlotte, N.C.
Terrell, Oscar David (J'43) (NZBAHO), Mech. Engr., Under Water Sec., Naval Ord. Testing Sta., 1030 E. Green St., Pasadena 1, Calif.
Terrell, William Alfred ('35; '35) (ABMO), Owner, Terrell Welding & Mch., 200 Mineola Ave., Roslyn Heights, N.Y.
Terry, Charles Dutton ('02; '08) (T), Secy., Treas., Boss Mfg. Co., Pres., Peoples Natl. Bank; *for mail*, 522 S. Tremont St., Kewanee, Ill.
Terry, Charles G., III (J'43) (CML), Indus. Engrg. Dept., Johns-Manville Corp., Manville; *for mail*, 150 West End Ave., Somerville, N.J.
Terry, Chas. M. ('25; '35), Ch. Engr., A. W. Cash Co., N. 18th St.; *for mail*, 224 W. Prairie, Decatur, Ill.
Terry, Edward B., Jr. (J'41) (MSW), 130-13—226th St., Laurelton, L.I., N.Y.
Terry, Ira Anderson ('43), Works Engr., Gen. Elec. Co., 1635 Broadway, Ft. Wayne 2, Ind.
Terry, Killey E. ('14) (DFS), Ch. Engr., S. D. Warren Co., Cumberland Mills, Me.
Terry, Raymond Holmes ('42) (SEF), 608 Petroleum Bldg., 714 W. Olympic Blvd., Los Angeles 15, Calif.
Terry, Roger Vernon ('31; F'45) (HCESN), *Melville Medallist*, '41; Asst. Ch. Engr., Newport News Shipbldg. & Dry Dock Co., 4101 Washington Ave.; *for mail*, 200 James River Dr., Hilton Village, Newport News, Va.
Terry, Roy V. ('40) (NB), Mem. Tech. Staff, Bell Tel. Labs., 463 West St., New York 14, N.Y.
Terry, Seymour ('20) (BFJKS), Works Mgr., Honolulu Iron Works Co., Box 3140, Honolulu 2, T.H.
Terwelp, Edward Joseph ('28; '35) (MNBCLD), Ch. Devel. Engr., Am. Mch. & Metals Inc., East Moline, Ill.
Terwilliger, Hal Rowe ('31; '42) (SKLHBD), Project Engr., Barrett Div., Allied Chem. & Dye Corp., Philadelphia 37, Pa.
Terwilliger, Harry Lester (A'01) (EK), Retired; 2275 Webster St., Palo Alto, Calif.
Tessin, William (J'37), 415 Bellevue Ave., Oakland 10, Calif.
Tessitor, Frank (J'30) (JBNOH), Engr., Matls. Labs., U.S. Bur. of Reclamation, 440 Customs House, Denver 2, Colo.
Tessmer, Raymond G., Sr. (J'45) (LND), Ch. Engr., Niagara Filter Corp., 3080 Main St., Buffalo 14; *for mail*, 413 Adam St., Tonawanda, N.Y.
Test, Ellis W. ('19) (RYNKHE), Asst. to Pres., Charge Research & Engrg., Pullman-Stand. Car Mfg. Co., 79 E. Adams St., Chicago 3, Ill.
Tetzel, Fred Bruce ('44) (SKFCB), Plant Engr., Louisville Gas & Elec. Co., 311 W. Chestnut St., Louisville 2; *for mail*, 1812 Fleming Rd., Louisville 5, Ky.
Tew, Gilbert Waddell (J'46), 1910 Lewis St., Durham, N.C.
Texada, Arnaud Preot, Jr. (J'36) (ESANZ), Research Engr., Shell Oil Co., Inc., Wood River; *for mail*, 1217 Clawson St., Alton, Ill.
Thaden, Theron John (J'46) (ABY), Goodyear Tire & Rubber Co., Akron; *for mail*, 577 E. Market, Akron 4, Ohio.
Thaker, S. H. ('21; '35) (CFS), Retired; 18 Harihar Soc., Maninager, Ahmadabad, India.
Thalman, Alfred John (J'42) (CSLREO), Application Engr., Spec. Assignment to Gen. Mgr., Clark Controller Co., 1146 E. 152nd St., Cleveland 10, Ohio.
Thayer, Paul William ('18; '24) (MCD), Inspr., Minneapolis-Honeywell Regulator Co., 2753—4th Ave. S., Minneapolis; *for mail*, 2808 Fremont Ave. S., Minneapolis 8, Minn.
Thayer, Robert Ellis ('11; '19) (RSEYJN), V.P., Simmons-Boardman Publ. Corp., 30 Church St., New York 7, N.Y.
Thayer, Royal C. (J'42), Capt., Hdq. Btry., 574th Antiaircraft Arty. (AW) Bn., A.P.O. 519B, c/o Postmaster, New York, N.Y.
Thearle, Ernest Lathrop ('26; '35) (BHKEFA), Head, Mech. Investigations Div., Research Lab., Gen. Elec. Co., Schenectady; *for mail*, R.D. 1, River Rd., Rexford, N.Y.
Theilmann, Fred W. (J'36) (CLMNT), Plant Engr., Personal Products Corp., 4949 W. 65th St., Chicago 38, Ill.
Theiss, Ernest Stephen ('39; '46) (MCJDSO), Asst. Prof. Mech. Engrg., Duke Univ.; *for mail*, 2645 University Dr., Durham, N.C.
Theologitis, James S. (J'44) (ANMBC), Design Draftsman, Republic Aviation Corp., Farmingdale, L.I.; *for mail*, 240 Audubon Ave., New York 33, N.Y.
Theriault, Raymond Joseph (J'33) (BCJRA), Project Engr., Edw. G. Budd Mfg. Co., Red Lion & Veree Rd., Philadelphia; *for mail*, 204 Buttonwood Way, Glenside, Pa.
Theroux, Quentin Oren (J'45) (KNHBJ), Jr. Engr., Devel., Carrier Corp., 300 S. Geddes St., Syracuse 1, N.Y.
Thieblemont, Andre Henry ('44) (LDBMNZ), Plant Engr., Coty Products Corp., 423 W. 55th St., New York 19, N.Y.
Thiel, William A. ('46), Am. Bridge Co.; *for mail*, 225 Chase St., Gary, Ind.
Thielscher, Herman George ('23; '27) (SFCEKD), *Manager*, '44-'45; Mech. Engr., Charge of Generating Dept., Potomac Elec. Power Co., 10th & E Sts., N.W., Washington 4, D.C.
Thieme, Walter Irving (J'42) (ACFB), Capt., Army Air Forces, Aero. Engr. & Test Pilot, TSEST-5B, Wright Field, Dayton, Ohio; *for mail*, 403 Evans Ave., Missoula, Mont.
Thiessen, Linwood (J'35) (FGJ), Plant Engr., Vesuvius Crucible Co., Swissvale; *for mail*, 1037 Old Gate Rd., Pittsburgh 21, Pa.
Thithan, Kling ('25; '31; '35), Siamese Govt. Paper Mill, Bangkok, Siam.
Thoenebe, Herman Sylvester (J'44) (YZGJN), V.P., Charge of Indus. Dept., John Falkner Arndt & Co., Inc., Fidelity-Philadelphia Bldg., Philadelphia 9; *for mail*, 117 Twining Rd., Oreland, Pa.
Tholl, John Edward (A'44), Ch. Engr., Am. Tool & Mch. Co., 1415 Hyde Park Ave., Boston 36; *for mail*, 215 West St., Needham Heights, Mass.
Tholl, John F. ('30), 63 Parkinson St., Needham, Mass.

Thom, George Boyd ('34; '41) (SCKMF), Dir. of Trng., Sun Shipbldg. & Dry Dock Co., Chester; *for mail,* 59 Blackthorn Rd., Wallingford, Pa.
Thomas, A. Emil (J'40) (ABK), Lt. Comdr., U.S.N.R.; *for mail,* 111-22—120th St., Richmond Hill, L.I., N.Y.
Thomas, A. Ernest ('41) (CDFHK), Supvg. Engr., The Dime Savings Bank of Brooklyn, 9 DeKalb Ave., Brooklyn 1, N.Y.
Thomas, Albert Harry ('27) (R), Chmn., Bd. of Dirs., Buckeye Steel Castings Co., 2211 S. Parsons Ave., Columbus 7, Ohio.
Thomas, C. Y. ('46) (CMLSF), Gen. Mgr., Military Chem. Works, Inc., Box 604, Pittsburg, Kan.
Thomas, David Fraser (J'40) (JYNEA), Engr., Cent. Tech. Serv. Co., 520 Sexton Bldg., Minneapolis 15; *for mail,* 4811—37th Ave. S., Minneapolis 6, Minn.
Thomas, Emory Floyd (J'42) (SCKHFO), Asst. Plant Supt., Tenn. Valley Authority, 420—1st Ave. N., Nashville 3, Tenn.
Thomas, Felix ('14; '26), Maj., Book Hill Rd., Essex, Conn.
Thomas, Fred S. ('24) (ACHR), Staff Engr., Westinghouse Air Brake Co., Wilmerding, Pa.
Thomas, Frederick Hayward ('29; '32) (LMC), Prof. Mech. Engrg., Univ. of Tenn., Knoxville 16, Tenn.
Thomas, Harold D. ('40) (CFM), Mech. Engr., Photovend Corp., 30 E. 21st St., New York, N.Y.; *for mail,* 292 Lakeview Ave., Paterson, N.J.
Thomas, Horace Tucker ('09) (A), Retired; 114 Walnutt St. S., Lansing 2, Mich.
Thomas, Jack Arthur (J'42) (HSC), Lt. (E), Royal Canadian Navy, F.M.O., Halifax, N.S.; *for mail,* 184a Albert St., Belleville, Ont., Can.
Thomas, James A. (J'41) (CDK), c/o Mrs. Morton, 829 Braddock Ave., Braddock, Pa.
Thomas, James Rodefer ('45) (CDL), Supt., Kingsport Plant, Pa.-Dixie Cement Corp., Box 351, Kingsport, Tenn.
Thomas, James William ('41) (SFC), Watch Engr., Potomac Elec. Power Co., 1st & V Sts., S.W., Washington, D.C.; *for mail,* 9505—2nd Ave., Silver Spring, Md.
Thomas, John Harper (J'41) (FJK), Box 694, Chickasha, Okla.
Thomas, John Roberts, II (J'41), Aluminum Anodizing Technician, Dept. 52, Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford; *for mail,* Apt. D, 12 Oxford St., Manchester, Conn.
Thomas, Julian Baldwin ('26) (ACS), Pres., Gen. Mgr., Tex. Elec. Serv. Co., 810 Elec. Bldg., Ft. Worth 2, Tex.
Thomas, L. G. Lee ('39) (BCHJMS), V.P., Gen. Mgr., Economy Pumps, Inc., Treas., Gen Mgr., Liberty Planners, Inc., Hamilton, Ohio.
Thomas, Omer L. (J'45), Box 1271, Savannah, Ga.
Thomas, Percy Holbrook ('16) (AHS), Office of Ch. Engr., Fed. Power Comm., Washington 25, D.C.
Thomas, Ralph Llewellyn ('26) (CS), V.P., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Lexington Bldg., Baltimore 1; *for mail,* 803 St. Georges Rd., Baltimore 10, Md.
Thomas, Richard Victor (J'40) (JLM), Lt. (j.g.), O-V(S), U.S.N.R., c/o Babcock & Wilcox Co., Alliance, Ohio.
Thomas, Robert Glynn (J'39) (ZSLCW), Sales, Serv. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio; *for mail,* 1614 N. Decatur Rd., N.E., Atlanta, Ga.
Thomas, Robert Lewis (J'38) (CMB), Staff Engr., Albert Ramond & Associates, 435 N. Michigan Ave., Chicago 11; *for mail,* 1503 W. Chase Ave., Chicago 26, Ill.
Thomas, Rosswell Wilder ('21; '33) (CLEAFY), Mgr., Research & Devel., Phillips Petroleum Co., Bartlesville, Okla.
Thomas, Thalbert Rexford (J'40) (BZE), 3200 Gilbert St., Austin 21, Tex.
Thomas, Thomas Nehle ('45) (CMJDLN), Mgmt. Cons. Engr., Norris & Elliott, Inc., 85 E. Gay St., Columbus 15, Ohio; *for mail,* c/o J. F. Thomas, R.D. 1, West Sunbury Rd., Butler, Pa.
Thomas, Thos. R. ('30), Pres., Bijur Lub. Corp., 43-01—22nd St., Long Island City, N.Y.
Thomas, Warren Milner (J'42) (CLKJDZ), Design Engr., Phillips Petroleum Co.; *for mail,* 710 Jennings St., Bartlesville, Okla.
Thomas, Wiley W., Jr. (J'38), Tenn. Eastman Corp., Box 590, Knoxville 5, Tenn.
Thomas, William Bernard (J'45), Mech. Engr., Power Dept., Port Arthur Works, Tex. Co.; *for mail,* 321 Bledsoe Pl., Port Arthur, Tex.
Thomas, William Emil (J'44), 957 N. Buchanan Ave., Pocatello, Idaho.
Thomas, Willis Phelps ('29) (FSKMCJ), Pres., Diamond Power Specialty Corp., Box 288, Detroit 31, Mich.
Thomason, Joseph J. ('44) (CSHK), Mgr., Westinghouse Elec. Corp., 825 Exch. Bldg., Memphis 3, Tenn.
Thomason, Max Deberry ('33; '41) (MCT), 303 N. Ridge Ave., Kannapolis, N.C.
Thompson, A. Stanley (J'41) (BKAJHO), Research Engr., Franklin Inst., Philadelphia; *for mail,* 218 Cornell Ave., Swarthmore, Pa.
Thompson, Albert Loney ('40) (EKNRSY), Indus. Sales Engr., Gulf Oil Corp., Gulf Bldg., Pittsburgh, Pa.; *for mail,* 19 Parkinson St., Needham 92, Mass.
Thompson, Charles Shepherd, Jr. (J'46), 3909 Spruce St., Philadelphia, Pa.
Thompson, Charles T. (J'37) (BJS), Gear Engrg. Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 51-A Beach St., Marblehead, Mass.
Thompson, Clyde ('19; '23; '35) (CMS), Mech. Engr., Research Dept., Atlantic Coast Fisheries Co., 307 Water St., New York; *for mail,* 8301—12th Ave., Brooklyn, N.Y.
Thompson, Clyde Hendricks (A'44) (ACLMNZ), Supt., Aero Lines Trailor Div., Aero Services Inc., Metro. Airport, Van Noys; *for mail,* 1604 Edison Blvd., Burbank, Calif.
Thompson, Collins, Jr. (J'43) (BDL), Chem. Engr., Indus. Engrg. Div., E. I. du Pont de Nemours & Co., Deepwater, N.J.; *for mail,* 3208 W. 2nd St., Wilmington 172, Del.
Thompson, Daniel Goodwin (J'31) (CDM), Supt., Htg. Device Div., Gen. Elec. Co., Bridgeport; *for mail,* 499 Sherwood Pl., Stratford, Conn.
Thompson, Donald ('42) (GLD), Planning Engr., Carbide & Carbon Chemicals Corp., South Charleston, W.Va.
Thompson, E. Frederick (J'40) (JLMB), Box 18, Blenheim, Ont., Can.
Thompson, Earl A. ('25), Earl A. Thompson Mfg. Co., 1300 Hilton Rd., Ferndale 20; *for mail,* Woodridge Rd., Bloomfield Hills, Mich.
Thompson, Edward Cyrus ('46) (CLO), Head Planning Engr., Chem. Warfare Serv., Rocky Mountain Arsenal, Denver 2; *for mail,* 12 S. Ogden St., Denver 9, Colo.
Thompson, Edward S. (J'31) (AECSKF), Engr., Aviation Div., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail,* 10 Trinity Rd., Marblehead, Mass.
Thompson, Elliott Rogers ('46) (BKP), Mgr., Insidline Div., Baldwin-Hill Co., 501 Klagg Ave., Trenton 2, N.J.
Thompson, Ernest Burke (J'45) (BJNLGM), Mch. Designer, E. D. Jones & Sons Co., 25 Depot, Pittsfield, Mass.
Thompson, Ernest Leon (J'43) (SEFAKR), Engr., Gen. Elec. Co., Western Ave., Lynn; *for mail,* 77 Park St., West Lynn, Mass.
Thompson, Glenn Alan (J'31) (RBEC), Serv. Engr., N.Y. Air Brake Co., 420 Lexington Ave., New York 17, N.Y.
Thompson, Gordon Lionel (J'45), 16 Willow St., Rockland, Me.
Thompson, Harold (J'41) (CDJ), Lt., 671st Engr. Co., Camp Bowie, Tex.
Thompson, Henry Havelock ('44) (ADL), Mgr., Engrg. & Serv., Westinghouse Elec. Corp., 1625 K St., N.W., Washington, D.C.; *for mail,* 104 Elm Ave., Takoma Park, Md.
Thompson, Howard Grant ('40) (CEM), Mgr., Aluminate Chemicals Ltd., 555 Eastern Ave., Toronto 8, Ont., Can.
Thompson, Hugh Lindsay ('94), Admin. Cons. Engr., Scoville Mfg. Co., 99 Mill St., Waterbury 91; *for mail,* 129 Pine St., Waterbury 33, Conn.
Thompson, John Clement (J'42), Lt., U.S.N.R., Wood Apts., Gen. Delivery, Waynesboro, Va.
Thompson, John George Hatch ('34; '35) (MJB CTA), Assoc. Prof., Dept. of Mech. Engrg., Agric. & Mech. College of Tex., College Station, Tex.
Thompson, John I. ('42) (LMOCDY), Partner, John I. Thompson & Co., 921—17th St., N.W., Washington 6, D.C.
Thompson, John Raymond (J'42) (SMLNWC), Mech. Engr., Gen. Elec. Co., 60 Jackson St., Holyoke; *for mail,* 165 Granby Rd., South Hadley Falls, Mass.
Thompson, John Roy ('15) (R), Retired; 1600 N. Stanley Ave., Los Angeles 46, Calif.
Thompson, John William ('44) (BCJLNS), Ch. Mech. Engr., Jackson & Moreland, Park Sq. Bldg., Boston 16, Mass.
Thompson, Joseph Charles (J'42) (OBLC), Design Engr., Gen. Am. Trans. Corp., 209 W. Jackson, Chicago; *for mail,* 2430 N. Auers, Chicago 47, Ill.
Thompson, Mason B. (J'41) (ACH), Sr. Structures Engr., Curtiss-Wright Corp., Columbus, Ohio; *for mail,* 815 W. 60th Terrace, Kansas City 2, Mo.
Thompson, Maxwell Alexander (J'45) (HSCZ), Div. Dispatcher, So. Calif. Edison Co., Big Creek; *for mail,* Edison Powerhouse 3, Auberry, Calif.
Thompson, Milton John ('31; '35; '35) (ABEC HK), Prof. & Chmn., Aero. Engrg. Dept., Assoc. Dir., Defense Research Lab., Univ. of Tex., Austin 12, Tex.
Thompson, Norman (J'41) (EFG), Engr., Research Lab., Socony-Vacuum Oil Co., Inc., Paulsboro, N.J.; *for mail,* 10 Central Ave., Bryn Mawr, Pa.
Thompson, O. C. ('10; '15) (BGJNWY), Retired; 44 Lake Dr., Mountain Lakes, N.J.
Thompson, Paul Wheeler ('15; '25) (SCF), V.P., Charge Engrg., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Thompson, Ralph Emerson ('14; '35) (ABCD JM), Dir., Pres., Exec. Comm., Reed-Prentice Corp., 677 Cambridge St., Worcester; *for mail,* 428 Warren St., Brookline, Mass.
Thompson, Ralph Frederick, Jr. (J'45), 1209 Floral St., N.W., Washington 12, D.C.
Thompson, Ray A. (J'41) (SWKLDO), Steam Application Engr., Westinghouse Elec. Corp., 3451 E. Marginal Way, Seattle 4, Wash.
Thompson, Richard Morgan, Jr. (J'44) (ACM), Lt., U.S.S. *A. J. Luke,* DE-577, F.P.O., New York, N.Y.; *for mail,* 300 Woodlawn St., Fall River, Mass.
Thompson, Robert Alden (J'33) (ABSNEH), Acting Head, Mech. Engrg. Dept., College of Engrg., Univ. of Fla., Gainesville, Fla.
Thompson, Ross E. ('39) (AEKMS), Mech. Engr., Fed. Advisers Inc., 70 Pine St., New York 5, N.Y.; *for mail,* 902 West Tijeras, Albuquerque, New Mex.
Thompson, Sanford E. ('09) (CMJLO), Pres., Thompson & Lichtner Co. Inc., 806 Park Sq. Bldg., Boston 16, Mass.
Thompson, Sidney ('26; '30) (EFK), Ch. Engr., Suc. J. Serralles, Mercedita; *for mail,* Box 1788, Ponce, P.R.
Thompson, Stephen J. ('36), Governing Dir., John Thompson Water Tube Boilers, Ltd., Ettingshall, Wolverhampton, Staffs, England.
Thompson, W. G. (J'34), Parker Paper Co.; *for mail,* 605 Montlieu Ave., High Point, N.C.
Thompson, Walter L. ('45) (CEFOS), Mgr., Garber Plumbing & Htg. Co., 811 Hanna Bldg., Cleveland 15, Ohio.
Thompson, Warren G. C. ('18; '26) (SFEKLY), Assoc. Prof., Mech. Engrg. Dept., Pa. State College; *for mail,* 235 S. Gill St., State College, Pa.
Thompson, William Charles (J'43), 122 Randall Ave., Freeport, N.Y.
Thompson, William Dennison ('44) (FEKLBC), Mgr., Indus. Dept., Laclede Gas Light Co., 1017 Olive St., St. Louis 1, Mo.
Thompson, William H. ('37) (K), Ch. Engr., Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4; *for mail,* 166 Hillside Ave., Chatham, N.J.
Thompson, Willis Frederick ('19; '24; '30) (OS FTKC), V.P., Ch. Mech. Engr., Westcott & Mapes, Inc., 109 Church St., New Haven 5, Conn.
Thompson, Woodard Hull (J'41) (KMC), Student Engr., Gen. Elec. Co.; *for mail,* 314 N. Columbus Ave., Louisville, Miss.
Thompson, Yale E. (J'38) (AC), Contact Engr., Lockheed Aircraft Co., Burbank; *for mail,* 5237 Lockhaven, Los Angeles, Calif.
Thomson, A. MacFarlane ('26; '35) (S), Hotel Suburban, 141 S. Harrison St., East Orange, N.J.
Thomson, Francis duPont ('92; '99) (EHSA), 261 E. Main St., Elkton, Md.
Thomson, George H. ('21; '35) (COSTY), Sales Agt., 143 Broadway, Cambridge 42, Mass.
Thomson, Hans Tennison (J'42) (ABE), Mich. Consltd. Gas Co., 372 Morris Ave., Muskegon, Mich.
Thomson, Herbert Wilbur (J'44) (SF), Engr., Charge of Substa. & Steam Plant Devel., U.S. Bur. of Reclamation, Old Post Office Bldg., Sacramento 14, Calif.
Thomson, James ('21) (NFDJ), Ch. Works Engr., Continental Fdy. & Mch. Co.; *for mail,* 4130 Magoun Ave., East Chicago, Ind.
Thomson, John Burry (J'38) (ACM), Pres., Thomson Industries, Inc., 29-05 Review Ave., Long Island City 1; *for mail,* 1029 Plandome Rd., Plandome, L.I., N.Y.
Thomson, John Stewart (J'01) (DT), 20 Lexington St., Newark 5, N.J.
Thomson, Robert Raymond ('40; '43) (FKS), Sales Engr., Combustion Engrg. Co., Inc., 461 Bd. of Trade Bldg., Kansas City, Mo.
Thomson, Robert Speirs ('36; '44) (KJWYC), Indus. Designer & Engr., 7 E. 48th St., New York 17; *for mail,* 98-50—67th Ave., Forest Hills, L.I., N.Y.
Thomson, Samuel Garver ('16) (RMAJLO), Cons. Engr., 3702—147th St., Flushing, L.I., N.Y.
Thomson, T. Kennard ('07), Cons. Engr., 32 W. 40th St., New York, N.Y.
Thone, John Joseph (J'44) (EFS), Turner Constr. Co., 420 Lexington Ave., New York 17; *for mail,* 5404—32nd Ave., Woodside, N.Y.
Thoren, Anders Victor ('42) (LCDMJE), Res. Engr., Travelers Ins. Co., 508 Builders Exch. Bldg., Duluth 2, Minn.
Thoresen, Ralph S. ('43) (EFSP), Asst. Ch. Engr., Petroleum Heat & Power Co., Stamford, Conn.
Thorn, Frederick Charles ('41) (NHY), Research Dir., Garlock Packing Co., Palmyra, N.Y.
Thornburg, Martin Lynn ('37) (EHJKM), Head, Mech. Engrg. Dept., Univ. of Ariz., Tucson, Ariz.
Thornburg, Robert Wentsorth (J'41) (AEYD SM), Lt., U.S.N.R., Naval Shipyard (P); *for mail,* 78 B. Ashley Ave., Charleston 6, S.C.
Thorne, Harold W. ('38) (CLT), North Star Woolen Mill Co., 200 Madison Ave., New York, N.Y.; *for mail,* West Rd., New Canaan, Conn.
Thorne, Raymond B. (J'40), St. Albans, Me.
Thornley, Albert Edward ('44) (CMJN), Secy., Engr., Narragansett Mch. Co., 70 Vale St.; *for mail,* 139 Ridge St., Pawtucket, R.I.
Thornton, Arthur F. ('21), c/o Am. Brake Shoe & Fdy. Co., Mahwah, N.J.
Thornton, Brian M. (J'25), Asst. Mgr., Boiler Dept., I.C.I. (Alkali), Ltd., Northwich; *for mail,* 29 Walnut Ave., Hartford, Cheshire, England.
Thornton, Frank, Jr. ('37), Engrg. Mgr., Assn. Activities, Westinghouse Elec. Corp., East Pittsburgh, Pa.
Thornton, Harry E. ('45) (ACJM), Devel. Engr., Bound Brook Oil-Less Bearing Co., Bound Brook, N.J.

Thornton, John Allan ('46) (DHT), Chmn., Managing Dir., B. Thornton, Ltd., Turnbridge; *for mail*, 650 Bradford Rd., Bradley Bar, Huddersfield, Yorks, England.
Thornton, John Miller (J'42), Ensign, Civ. Engr. Corps, U.S.N.R., 31st Naval Constr. Bn., F.P.O., San Francisco, Calif.; *for mail*, 222 Chestnut Rd., Sewickley, Pa.
Thornton, W. Lloyd (J'45) (CDML), Partner, Gen. Mgr., Steel Engrg. Co., 2700 Pafford St., Ft. Worth 10, Tex.
Thornton, William Foster ('09) (DHKRS), Retired; 330 N. Gerona Ave., San Gabriel, Calif.
Thorp, Arthur George, II (J'43) (BAH), Mech. Engr., P-2, Natl. Advis. Com. for Aero., Langley Field; *for mail*, 260 Regent St., Hampton, Va.
Thorp, Geo. G. ('92; '04), Lakewood Blvd., Madison, Wis.
Thorpe, C. Lloyd ('37; '44) (CMSJN), Gen. Mgr., Clarkson Co., 1700 El Camino Real, Belmont; *for mail*, 1205 Park Blvd., San Mateo, Calif.
Thorpe, William Alfred Charles ('20) (RFZC), Deputy Ch. Mech. Engr., Indian State Rys., (Retired); *for mail*, c/o Grindlay & Co., Parliament St., London, S.W. 1, England.
Thorson, Allen W. ('29; '39) (FRS), Mgr., Power Plant Div., Am. Engrg. Co., Aramingo & Cumberland, Philadelphia 25; *for mail*, 609 Braeburn Lane, Penn Valley, Narberth, Pa.
Thorson, Wilbur Raymond ('41) (CFKLOS), Asst. Elec. Supt., Belle Works, E. I. du Pont de Nemours & Co., Belle; *for mail*, 4006 Virginia Ave., S.E., Charleston 4, W.Va.
Thrall, Edward Wolff, Jr. (J'39) (AB), Structures Engr., El Segundo Div., Douglas Aircraft Co., Inc., 5651 W. Manchester Ave., Los Angeles 45; *for mail*, 529 Nectarine St., Inglewood, Calif.
Thress, John M. (J'38) (MNZCBA), Ord. Engr., Ord. Dept., U.S.A., Frankford Arsenal, Bridesburg St., Philadelphia 37, *for mail*, 6937 Horrocks St., Philadelphia 24, Pa.
Throckmorton, Edmund Hugh ('44) (EFHKOS), Supvg. Mech. Engr., United Light & Power Serv. Co., United Light Bldg., Davenport, Iowa; *for mail*, 1521—28th St., Rock Island, Ill.
Throckmorton, John Wickliffe ('28; '34) (KEF SBA), V.P., Petro-Chem. Devel. Co. Inc., 120 E. 41st St., New York 17, N.Y.
Throckmorton, Ralph Edgar (J'40) (CJRDMS), U.S. Navy; Watson Ave., North Haven, Conn.
Throne, Robert Franklin ('28) (SFEKHC), Supt. Steam Prod., Pub. Serv. Co. of Colo., 900—15th St., Denver 1, Colo.
Thuering, George Lewis (J'42) (CMEOA), Sr. Mfg. Engr., Lockheed Aircraft Corp., Burbank; *for mail*, 624 W. Wilson Ave., Glendale 3, Calif.
Thuerk, Hugh Campbell ('29; '33) (CSFO), Pres., N.J. Power & Light Co., 9 W. Blackwell St., Dover, N.J.
Thuermann, Wilfred John ('21; '25; '35) (CF DK), Plant Engr., Chain Belt Co., 1600 W. Bruce St., Milwaukee 4; *for mail*, 666 N. 74th St., Wauwatosa 13, Wis.
Thuesen, H. G. ('36; '41) (CMBZ), Head, Dept. of Indus. Engrg., Okla. Agric. & Mech. College, Stillwater, Okla.
Thull, Alvin Julius, Jr. ('45) (DHL), 1830 Sunset Blvd., San Diego, Calif.
Thumim, Carl ('23; '26; '32) (BANKJO), Engr., Charge of Spec. Products Div., Ite Circuit Breaker Co., 19th & Hamilton Sts., Philadelphia; *for mail*, 909 MacDade Blvd., Lansdowne, Pa.
Thumser, Robert C. ('19; '29) (CMSZ), Plant Mgr., Clinton Labs., Monsanto Chem. Co., Box 1991, Knoxville; *for mail*, 165 Outer Dr., Oak Ridge, Tenn.
Thuney, Francis Miles ('32; '42((CML), Exec. Engr., Minneapolis-Honeywell Regulator Co., 1101 Vermont Ave., N.W., Washington 5, D.C.
Thurlo, John Allen (J'39) (SEK), Jr. Mfg. Engr., Westinghouse Elec. Corp., Lester; *for mail*, 639—8th Ave., Prospect Park, Pa.
Thurman, Astor Lewis ('37; '45) (BCJ), Elec. Engr., Aetna Stand. Engrg. Co., Youngstown, Ohio.
Thurn, John Alexander (J'43) (ABZ), Jr. Product Engr., Sperry Gyroscope Co., Inc., Great Neck, L.I.; *for mail*, 55 Hanson Pl., Brooklyn 17, N.Y.
Thurston, Edward Day, Jr. ('09; '12) (FSK), Retired; Sharon, Conn.
Thurston, H. Geo. ('28), Managing Dir., Wantage Engrg. Co. Ltd., Wantage, Berks, England.
Thurston, Sidney (J'44) (ASBKHE), Mech. Research Engr., Natl. Advis. Com. for Aero., Cleveland Airport; *for mail*, 11219 Hopkins Ave., Cleveland, Ohio.
Thutlow, Oscar G. ('15), Box 341, Route 2, Birmingham 9, Ala.
Thybo, Boyd C. (J'39) (AW), Flight Test Mgr., Interstate Aircraft & Engrg. Corp., De Kalb, Ill.; *for mail*, Reva, S.D.
Thyer, George Elwood (J'45) (JO), Sales Engr., Natl. Cylinder Gas Co., 205 W. Wacker Dr., Chicago 6; *for mail*, 4863 N. Rockwell St., Chicago 25, Ill.
Tibbals, George Attwater ('97), 20 E. 76th St., New York 21, N.Y.
Tichvinsky, Leonid Michael ('41) (BEHKOS), Prof., Mech. Engrg. Dept., Washington Univ., St. Louis 5, Mo.
Ticknor, Winton A. (CFKDM), Indus. Planning Engr., Corning Glass Works, Corning, N.Y.
Ticotsky, Matthew Edward (J'46) (BST), Mar. Engr., Design Sec., Bldg. 12, Naval Shipyard, Philadelphia, Pa.; *for mail*, 598 George St., New Haven, Conn.
Tiedeberg, John Waldemar, Jr. (J'44) (K), Student Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.; *for mail*, 254 Merrison St., Teaneck, N.J.
Tiedemann, James Bonn (J'45) (EFK), Ensign, D(L), U.S.N.R., 2331 N. 46th St., Milwaukee 10, Wis.
Tieke, William ('27; '32; '35) (CNMBAJ), Ch. Design Engr., Forgings Div., Aluminum Co. of Am., 2210 Harvard Ave., Cleveland 5; *for mail*, 6705 Southfield Ave., Cleveland 9, Ohio.
Tiernan, James Blaine (J'44) (HMBSLC), M.M., Edgar Bros. Co.; *for mail*, Box 231, McIntyre, Ga.
Tierney, James Edwin ('22; '35) (AR), Lockheed Aircraft Corp., Burbank; *for mail*, 3912 S. Bronson Ave., Los Angeles 43, Calif.
Tierney, William Douglass (J'44) (KBJH), D. I. C. Staff, Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge; *for mail*, 19 Whitridge St., Boston, Mass.
Thiesler, Julius ('43) (BCHJMN), V.P., Ch. Engr., Dallett Co., Mascher at Lippincott St., Philadelphia 33; *for mail*, Huntingdon Pike, Huntingdon Valley, Pa.
Tietjen, William Diedrich (J'43) (BCMZ), Asst. Project Engr., Sperry Gyroscope Co., Inc., Great Neck, N.Y.; *for mail*, 255 Sherman Ave., New Haven 11, Conn.
Tietz, Thomas Edwin (J'44) (JMY), 212 Irving Blvd., Los Angeles 4, Calif.
Tiffin, Charles Martineau (J'46) (ABO), Internatl. Stand. Elec. Co., 67 Broad St., New York; *for mail*, 638 W. 138th St., New York 31, N.Y.
Tiger, Howard L. ('21; '26; '35) (SLT), V.P., Charge Research & Devel., Permutit Co., 330 W. 42nd St., New York 18, N.Y.
Tigges, Alexander Joseph ('41), Air Preheater Corp., 60 E. 42nd St., New York, N.Y.
Tilghman, Richard H. ('39) (SFZ), 3730 Gundry Ave., Long Beach 7, Calif.
Till, Fred (J'42) (ACJ), Seaman, 1st Class, U.S.N.R., Naval Trng. Sch. (Radio Materiel), Navy Pier, Chicago, Ill.
Till, Ralph J. (J'38) (CKS), E. I. du Pont de Nemours & Co., 1007 Market St., Wilmington, Del.; *for mail*, R.D. 3, Winding Lane, Media, Pa.
Tilley, John ('14) (C), Mech. & Elec. Engr., Vermilya-Brown Co., Inc., 100 E. 42nd St., New York, N.Y.
Tilley, Melvin Martin (J'40) (CBJ), 252 Columbia Ave., Berkeley 8, Calif.
Tillquist, David E. (J'41), Lt., c/o B. Tillquist, Apt. 5A, 150 S. Harrison St., East Orange, N.J.
Tilson, Howard ('15; '23; '31) (COMJW), Pres., Donnelly Elec. & Mfg. Co., 3050 Washington St., Boston, Mass.; *for mail*, 26 Congdon St., Providence 6, R.I.
Tilton, Henry Benson ('17) (CMBL), Dir., Morley Co., Islington St.; *for mail*, 1229 South St., Portsmouth, N.H.
Tilton, Peter Deming (J'42) (BCMG), Servomechanisms Lab., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Timbs, Edward ('19; '35) (BJ), Engr., Natl. Supply Co., 1524 Border Ave., Torrance; *for mail*, 5131—10th Ave., Los Angeles 43, Calif.
Timmerman, Harold Wayne (J'44), Ensign, U.S.N.R., U.S.S. *Preserver*, ARS-8, F.P.O., San Francisco, Calif.; *for mail*, 1705 S 7th Ave., Sioux Falls, S.D.
Timmerman, Thomas John (J'44) (BDM), Mech. Engr., P-2, Puget Sound Naval Shipyard; *for mail*, Box 1784, Sta. A, Bremerton, Wash.
Timmis, Pierce ('20) (DKLGW), Indus. Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5, Pa.
Timmons, John Joseph (A'45) (BCEFKS), Supvr., Fidelity & Casualty Co. of N.Y., 810 Arrott Bldg., Pittsburgh 22; *for mail*, 27 Ralston Pl., Mt. Lebanon, Pittsburgh 16, Pa.
Timms, Howard Lister (J'44) (ABC), Sec. Engr., Gen. Elec. Co., 2000 Taylor St., Ft. Wayne, Ind.
Timoshenko, Stephen P. ('24; F'38) (B), *Worcester Reed Warner Medallist*, '35; Prof., Theoretical & Applied Mechanics, Stanford Univ., Stanford University, Calif.
Timpson, Willard Q. ('31; '40) (CJM), 511 Hillside St., Ridgefield, N.J.
Tincher, T. S. ('37), Inspr. of Mchy., Washington Gas Light Co.; *for mail*, Apt. 220, Dorchester House, 2480—16th St., N.W., Washington, D.C.
Tindal, Lorenza Vastine (J'44), 411 Whitney Ave., Pittsburgh 21, Pa.
Tindall, Edwin Lamar (J'29), Maj., Engr. Sec., Hdq., Army Serv. Forces, A.P.O. 887, New York, N.Y.; *for mail*, 1324 Carnegie Bldg., Pittsburgh 30, Pa.
Tindall, Whitney Paul ('23; '35) (BLNS), Asst. Plant Engr., Ecusta Paper Corp., Box 200, Pisgah Forest; *for mail*, 230 W. Main St., Brevard, N.C.
Tingey, Richard Hume ('39) (SBEHKA), Asst. Tech. Mgr., Shipbldg. Div., Bethlehem Steel Co., E. Howard St.; *for mail*, 90 Squanto Rd., Quincy 69, Mass.
Tingley, John W., Jr. (J'38) (SEFCOJ), V.P., E. L. Woolley Co., 60 Dunnell Lane; *for mail*, 235 Cottage St., Pawtucket, R.I.
Tingquist, Stanley Carroll (J'44) (JMN), Matls. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail*, 329 Locust St., Pittsburgh 18, Pa.
Tinker, Oscar Austin (J'42) (CMDWYL), Indus. Engr., Paper Div., Continental Can Co.; *for mail*, 620 S. Washington St., Van Wert, Ohio.
Tinker, Townsend ('29; '37) (KSLFEH), *Junior Award*, '33; Ch. Engr., Ross Heater & Mfg. Co., Inc., 1407 West Ave., Buffalo 13, N.Y.
Tint, Lester Martin (J'42) (AMCN), Aero. Engr., Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field; *for mail*, 468 Hanover St., Palo Alto, Calif.
Tipton, Hobart (J'42) (LMCJ), Mech. Designer, Barber-Colman Co.; *for mail*, 318 Whitman St., Rockford, Ill.
Tirrell, Robert Wilson ('38) (SLOEC), Mech. Engr., Ebasco Services Inc., 2 Rector St., New York 6, N.Y.; *for mail*, 140 Meadowbrook Rd., Englewood, N.J.
Tisch, Frank Phineas ('42) (AMNJCD), Ch. Engr., Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50; *for mail*, 426 N. Humphrey Ave., Oak Park, Ill.
Tishlarich, Ottmar M. ('40) (BLM), Plant Engr., A. M. Byers Co., S. 6th & Bingham St., Pittsburgh 3; *for mail*, 230 W. Riverview Ave., Bellevue, Pittsburgh 2, Pa.
Titherington, William K. (J'41) (SZAF), Test Engr., De Laval Steam Turbine Co., Trenton; *for mail*, 612 Concord Circle, Trenton 8, N.J.
Titus, James W. (J'43) (NZRAB), Sub-Sec. Head, Mech. Components, Fire, Missile & Pilotless Aircraft Control Div., Naval Research Lab., Bellevue; *for mail*, 13 Danbury St., S.W., Washington 20, D.C.
Titus, Robert Max (J'44) (ABH), 47 Warner Pl., South Waterbury, Conn.
Tizard, Thomas Edward (J'39) (KEFJOS), Test Engr., Power Plant, Inspiration Consltd. Copper Co.; *for mail*, House 82, Inspiration, Ariz.
Toben, Bernard E. (J'38) (ZNMA), Asst. Engr., Pat. Dept., Internatl. Business Mchs. Corp., 590 Madison Ave., New York 22; *for mail*, 19 Belmont Terrace, Yonkers 3, N.Y.
Tobey, Julian E. ('35) (FRS), Managing Dir., Fairmont Coal Bur., 122 E. 42nd St., New York 17, N.Y.
Tobey, W. Arthur ('26; '35) (SKFOHM), Sales Engr., Manistee Iron Works Co.; *for mail*, 105 Pine St., Manistee, Mich.
Tobias, Norman (J'40) (OYNJDA), Assoc. Engr., Design, Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland, Ohio; *for mail*, 4 Reed St., Pittsburgh 19, Pa.
Tobin, Roger Keith (J'40) (SCB), Turbine Engr., Design, Elliott Co.; *for mail*, R.D. 2, Woodlawn, Jeannette, Pa.
Toby, Ivon Channing (J'43) (CNBMJ), Mch. Designer, Canadian Acme Screw & Gear Ltd., 207 Weston Rd. S.; *for mail*, 41 Fern Ave., Toronto, Ont., Can.
Todd, James Herbert ('30), Rms. 601-3, 11 Broadway, New York 4, N.Y.
Todd, James Mulherrin ('22; '29; F'44) (OSC HFL), *Manager*, '33-'36, *Vice-President*, '36-'38; Cons. Engr., 217 N. Peters St., New Orleans 16, La.
Todd, John P. (J'41), 207 Mabrick Ave., Pittsburgh 16, Pa.
Todd, Meryl Lynn ('44) (LFK), 100 Highland Blvd., Waterloo, Iowa.
Todd, Paul E. ('23; '35), Blackhawk Hotel, Davenport, Iowa.
Todd, Percy E. (J'41), c/o E. A. Todd, Route 2, 535 Lawrenceville Rd., Decatur, Ga.
Tode, Arthur M. ('23; '35), Hon. Pres., Propeller Club of the U.S., 17 Battery Pl., New York, N.Y.
Toensfeldt, Kurt ('18), Mgr., Pat. Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Toher, John Thomas (J'43) (CES), Lt. (j.g.), U.S.N.R., 3230 E. 2nd St., Long Beach, Calif.
Tole, John Hollis ('44) (AE), Mgr., Westinghouse Elec. Corp., 707 Boyle Bldg., 103 W. Capital St.; *for mail*, 220 Fountain, Little Rock, Ark.
Tolf, McFarland (J'40) (EBS), Engr., Stand. Oil Co. of Calif., Box 397, La Habra, Calif.
Toliver, Lee (J'43) (ACO), Box 1663, Santa Fe, New Mex.
Tolkmitt, Richard Gustave (J'37) (ACJNR), Design & Tooling Engr., Aero Services, Inc., Metro. Airport, Van Nuys; *for mail*, Route 2, 13523 Gladstone Ave., San Fernando, Calif.
Tolman, Charles Prescott ('38; F'41) (MLNZ DC), Cons. Engr., 82-04 Austin St., Kew Gardens 15, L.I., N.Y.
Tolman, Edgar Bronson, Jr. ('29) (DHS), Pres., United Conveyor Corp., 37 W. Van Buren St.; *for mail*, 5333 University Ave., Chicago, Ill.
Tolman, Lee Prescott ('46), Capt., Corps of Engrs., Box 2601, Washington 25, D.C.; *for mail*, 52 Bayview Ave., Northport, L.I., N.Y.
Tolman, Raymond H. (J'40), Spring Rd., Westboro, Mass.
Toma, John (J'44) (MBCJNY), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail*, 2818 E. 118th St., Cleveland 20, Ohio.
Tombaugh, Roy Wilson (J'44) (JBSNMR), Devel. Engr., Extrusion Tooling, Aluminum Co. of Am., 5151 Alcoa Ave., Los Angeles 11; *for mail*, 1103 Richwood Ave., El Monte, Calif.

Tomey, Jerry G. (J'46), Anaconda Wire & Cable Co., Hastings-on-Hudson, N.Y.
Tomita, Arthur Kiyoshi (J'45) (BEP), Mech. Engr., Design, Refinery Engrg. Co., 600 S. Michigan Ave.; *for mail*, 826 S. Wabash, Chicago, Ill.
Tomkins, Stanton Edwin (J'39) (BCS), Instr., Wyomissing Poly. Inst., Wyomissing; *for mail*, 708 Reading Ave., West Reading, Pa.
Tomkinson, Stanley E. (J'46), 8 Allen St., Lebanon, N.H.
Tomlinson, Charles Sprague ('17; '25; '35) (SCFK), Supt., Neosho Sta., Kan. Gas & Elec. Co., Box 399, Parsons, Kan.
Tomlinson, Joseph Robert (J'41) (CJMZY), Asst. to Gen. Mgr., Barden Corp., E. Franklin St.; *for mail*, R.F.D. 2, Danbury, Conn.
Tomlinson, John Reid (J'42) (BEP), Partner, J. R. & E. S. Tomlinson, 1103 N. 11th St., Duncan, Okla.
Tompkins, Francis Meade ('42) (SCA), V.P., Charge Mech. Engrg. & Pur., Chas. H. Tompkins Co., 907—16th St., N.W., Washington 6, D.C.
Tompkins, Harold D. ('17; '26) (BMS), Treas., Engr., Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City, N.J.
Tompkins, Harold Willers (A'46) (HKS), Lt., Turbo-Elec. Sch., U.S. Maritime Serv. Trng. Sta., Sheepshead Bay, New York, N.Y.; *for mail*, 106 Gardner Ave., Glenolden, Pa.
Tompkins, Howard ('41; '42) (EFS), V.P., Secy., Rowland Tompkins & Son, Inc., 420 Lexington Ave., New York 17, N.Y.
Tompkins, J. Gordon (J'32), Field Engr., Norma-Hoffmann Bearings Corp., 155 E. 44th St., New York 17; *for mail*, Box 864, Chatham Rd., Chappaqua, N.Y.
Tompkins, Rowland ('38), Rowland Tompkins & Son, 420 Lexington Ave., New York, N.Y.
Tompkins, Sherman A. (J'39) (CEK), Spec. Rep., Johns-Manville Corp., 22 E. 40th St., New York, N.Y.
Tong, Jack Wellington (J'44) (AES), Asst. Mech. Engr., U.S. Engrs., San Francisco; *for mail*, 1715 E. 23rd St., Oakland 6, Calif.
Tooker, Welburn Clyde (J'43) (MJKZ), Seaman, 1st Class, Radio Technician, Naval Trng. Sch. (Pre-Radio Material), Co. 1086, Hugh Manley Sch., Chicago 12, Ill.
Toolin, Parks Roe (J'39), Research Engr., Research Lab., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Toomer, Lamar Chappell (J'45) (SFRH), 2nd Asst. Engr., U.S. Merchant Mar., 612 N. Washington St., Portsmouth, Va.
Toomey, Thomas Cornelius ('43) (EFOS), Salesman, Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 139 Fairview Ave., West Brighton, Staten Island 1, N.Y.
Toone, James Ann, Jr. (J'43), Engrg. Dept., Winter Steel Works, Gen. Elec. Co.; *for mail*, 305 E. Woodland Ave., Ft. Wayne, Ind.
Toothacker, William Sanford, Jr. (J'38) (EBJM), Maj., Ord. Dept., U.S.A., The Pentagon, Washington, D.C.; *for mail*, 816 S. Highland St., Arlington, Va.
Toppin, Francis V., Jr. (J'37) (CJL), Asst. Plant Mgr., Rohm & Haas Co., Bristol, Pa.; *for mail*, 535 Lake Ave., Lyndhurst, N.J.
Torell, Baldwin W. (J'41) (ACJB), Liaison Engr., Trans Can. Air Lines, Malton Airport, Toronto, Ont., Can.
Torley, John Frederic (J'43) (MJLCZ), Mech. Engr., Natl. Malleable & Steel Castings Co., S. Dock St., Sharon, Pa.
Tornebohm, Hilding ('37) (BLM), Ingeniors Vetenskaps Akademien, c/o Albert Bonnier Publ. House, 665 Lexington Ave., New York 22, N.Y.
Tornel, Pedro Martinez ('42) (AR), Subsecretaria de Comunicaciones y Obras Publicas, Tacuba 5, Mexico, D.F., Mex.
Torok, Elmer ('36) (COMJK), Pres., East. Tenn. Sheet Metal Works, Inc., 620 Broad St., Bristol, Tenn.
Torrance, Henry ('97; '02) (K), Pres., Norwalk Co., Inc., South Norwalk, Conn.; *for mail*, 112 E. 17th St., New York 3, N.Y.
Torrance, Kenneth Rollie (J'39) (OCFBHD), Owner, Mgr., R. C. Torrance & Son, Cascade Rd., Lake Placid, N.Y.
Torreano, James Charles (J'44), 199 E. Greendale Ave., Detroit 3, Mich.
Torrence, George Paull ('32) (CJY), 20001 S. Woodland Rd., Cleveland 22, Ohio.
Torres, Agustin (J'44), East. Dist. Engrg. Dept., Westinghouse Elec. Corp., 40 Wall St., New York; *for mail*, 22 Midwood St., Brooklyn, N.Y.
Torres, Marino (J'39) (CMKB), Abarca Warehouses Corp., Box 2352, San Juan 10, P.R.
Torrey, Donald Fuller (A'31) (FS), V.P., Gen. Mgr., Currie & Campbell Co., 117 S. 17th St., Philadelphia 3, Pa.
Toth, Alex (J'41) (JHBM), Mech. Engr., Asst. to Ch. Engr., Johnson Suture Co., 5001 W. 67th St., Chicago 38; *for mail*, 2443 W. 46th St., Chicago 32, Ill.
Toth, Louis Wendell (J'41) (CMBSZ), Graduate Student in Mech. Indus. Engrg., Univ. of Mich., Ann Arbor, Mich.; *for mail*, Box 339, Route 6, Toledo, Ohio.
Totten, Harold W. ('38) (CE), Pres., Wash. Iron Works, 318 E. Lamar St., Sherman, Tex.
Toulmin, Harry A., Jr. ('23), Toulmin & Toulmin, Mutual Home Bldg., 308 W. 1st St., Dayton 2, Ohio.
Toups, Harry Sidney, Jr. (J'41) (JMBCSD), Ch. Inspr., Ord. Div., Thibodaux Boiler Works, Inc.; *for mail*, 209 W. 6th, Thibodaux, La.
Tousey, William Maurice (J'44) (RBNE), Mech. Draftsman, Gen. Offices, Fla. E. Coast Ry.; *for mail*, Box 1057, St. Augustine, Fla.
Tove, William B. ('44) (LDM), Rm. 803, 125 Worth St., New York 13, N.Y.
Tow, Bennett Kurtzon (J'41), 2116 Grant St., Berkeley 3, Calif.
Towers, James F. ('22; F'42) (AF), Pres., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.
Towle, Harold Pride (J'32) (ABMNSZ), Design, Project Engrg. Div., E. I. du Pont de Nemours & Co., Chemical Rd.; *for mail*, 626 Walnut Ave., Niagara Falls, N.Y.
Town, Frederic Edward ('05) (CA), Retired; 409 Hillside Pl., South Orange, N.J.
Towne, Joseph M. ('18), Pres., Natl. Blank Book Co., Holyoke, Mass.
Towne, Miles Augustus ('44) (CMKOWY), Dir., Indus. Engr., Miles A. Towne & Associates, 328 S. Knight Ave., Park Ridge, Ill.
Towne, Robin Max (J'46), Assoc., Firm of Erwin Weber, 603 Medical Arts Bldg., Seattle; *for mail*, 4020 E. 135th, Seattle 55, Wash.
Townsend, Albert C. ('10; '18; '26) (FKS), Retired; 363 Verplanck Ave., Beacon, N.Y.
Townsend, Albert J. ('40) (REBK), Ch. Mech. Engr., Lima Loco. Works, Inc., Lima, Ohio.
Townsend, Charles Delmar ('34; '45) (MCJLN), Plant Engr., S. K. Wellman Co., 1374 E. 51st St., Cleveland 3; *for mail*, 10510 Park Lane, Cleveland 6, Ohio.
Townsend, James A. (J'44) (AEH), Assoc. Mech. Engr., Army Air Forces, Materiel Comd., Wright Field; *for mail*, 44½ Holt St., Dayton, Ohio.
Townsend, John Selby ('27) (CJMDSE), Maint. Supt., Carnegie-Ill. Steel Corp., 3426 E. 89th St., Chicago 17; *for mail*, 10527 Hale Ave., Chicago 43, Ill.
Townsend, Norman F. ('31; '35) (CNMZTJ), Ch. Engr., Willcox & Gibbs Sewing Mch. Co., 214 W. 39th St., New York 18, N.Y.
Townsend, Thomas Hewell, Jr. (J'43) (AEBNHK), Student, Ga. Sch. of Tech., Atlanta, Ga.; *for mail*, 2714 Louise Ave., Knoxville 15, Tenn.
Townsend, Walker ('25; '35) (SFBCK), Engr., Proposition Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail*, Baywater Dr., Noroton, Conn.
Townsend, Wisner Robinson ('27; '35) (SCLOA), Asst. Ch. Engr., Charles Pfizer & Co., Inc., 11 Bartlett St., Brooklyn 6; *for mail*, 372 Clermont Ave., Brooklyn 5, N.Y.
Tozer, Sydney John ('33; '46) (SKFLZE), Ch. Engr., Englewood Plant, Am. Can Co., 6017 S. Western Ave.; *for mail*, 6723 S. Maplewood Ave., Chicago 29, Ill.
Trabold, Hermann Simmons, II (J'44) (RSA), Aviation Electronics Technician's Mate, U.S.N.R., Carrier Aircraft Serv. Unit (F) 58, F.P.O., San Francisco, Calif.; *for mail*, 199 Dubois St., Newburgh, N.Y.
Trachtenbarg, Eugene Robert (J'43), 322—3rd Ave., Moline, Ill.
Track, Frank A. ('18; '22) (CKS), Cons. Engr., United Nations Relief & Rehabilitation Admin., Shanghai, F.P.O. 907, c/o Postmaster, San Francisco, Calif.
Tracy, Stephen J. ('29; '36) (ESB), Asst. Prof. Mech. Engrg., College of the City of New York, W. 139th St., New York 31; *for mail*, 166-24—26th Ave., Flushing, N.Y.
Traill, John White ('44) (KOZ), Partner, Roderick Ross & Traill, 123 William St., Melbourne, Victoria, Australia.
Trainer, James Edward ('19; '35; 'F'42), V.P., Firestone Tire & Rubber Co., Akron, Ohio.
Trainis, Alphonse (J'43) (AMHC), Weight Engr., Chance Vought Aircraft Div., United Aircraft Corp., 550 Main St.; *for mail*, 362 Laughlin Rd., Stratford, Conn.
Trainor, Joseph James ('44) (SKCFJB), Ch. Engr., Pub. Serv. Co. of Ind., Inc., 110 N. Illinois St., Indianapolis 9, Ind.
Tramontini, Vernon Nelson (J'39) (KMNA), Project Engr., Stewart Warner Corp., 1514 Drover St.; *for mail*, 5120 University Ave., Indianapolis, Ind.
Transou, Adam J. ('36; '46) (LSEBJ), Engrg. Dept., Tide Water Associated Oil Co., E. 22nd St., Bayonne, N.J.
Tranzen, Karl ('34; '35) (BCEFHS), Engr., Southmoor Hotel, 67 Stony Island, Chicago 37; *for mail*, 4758 Lake Park Ave., Chicago 15, Ill.
Trapnell, John Moore ('18; '35) (DJMS), Engr., R. D. Cole Mfg. Co.; *for mail*, 32 W. Washington St., Newnan, Ga.
Trapnell, Nicholas McLean ('28; '35) (RSEJC), Supt. M.P., Chesapeake & Ohio Ry. Co., 9th & Main Sts., Richmond 10, Va.
Traudt, William Frederick ('14; '20) (CHL), Pres., Gen. Mgr., Taber Pump Co., 291 Elm St.; *for mail*, 7 Argyle Pk., Buffalo 8, N.Y.
Traut, Frederick Louis (J'40 (JLYC), Div. Engr., Maint. Dept., Calco Chem. Div., Am. Cyanamid Co., Bound Brook; *for mail*, 855 Broad Ave., Ridgefield, N.J.
Traver, Alfred Ellis ('30; '41) (AZESFC), Engr., Instrumentation, Socony-Vacuum Oil Co., Inc., 412 Greenpoint Ave., Brooklyn 22; *for mail*, 8 Wensley Dr., Great Neck, L.I., N.Y.
Travilla, James C., Jr. ('41) (RJB), Ch. Mech. Engr., Gen. Steel Castings Corp., Eddystone, Pa.
Travis, Leonard John ('24; '35) (JMLCN), Asst. Gen. Supt., Natl. Lead Co., 900 W. 18th St., Chicago 80; *for mail*, 227 S. Catherine Ave., La Grange, Ill.
Trawick, Jack George (J'44) (ZLSTJF), Dist. Rep., Minneapolis-Honeywell Regulator Co., 311 Protective Life Bldg., Birmingham, Ala.
Traxler, Eugene Rice (J'36) (YDT), Belting Engr., B. F. Goodrich Co., 500 S. Main St., Akron 11; *for mail*, 490 Hudson Rd., Stow, Ohio.
Trayler, Will Anderson, Jr. (J'36) (RSF), M.M., Columbus & Greenville Ry. Co., Columbus, Miss.
Treadwell, Burson (J'31) (CJM), Engr., Natl. Cash Register Co., Dayton 9; *for mail*, 1216 Oakdale Ave., Dayton 10, Ohio.
Treat, Robert M. (J'34), Lt., 76 Princeton St., Manchester, Conn.
Trebes, Harold Herman (J'42) (BHAM), Engr., Airplane Div., Airport Plant, Curtiss-Wright Corp., Buffalo; *for mail*, 46 Leroy Rd., Buffalo 21, N.Y.
Trecker, Francis J. (J'38) (ACO), Secy., Kearney & Trecker Corp., 6784 W. National Ave., West Allis 14, Wis.
Trecker, Joseph Leonard (A'45), Exec. V.P., Kearney & Trecker Corp., 6784 W. National Ave., West Allis, 14, Wis.
Trefethen, Lloyd Slade (J'46) (ABC), Layout Designer, Glenn L. Martin Co., Baltimore 3; *for mail*, Apt. D, 804 Wilson Point Rd., Baltimore 20, Md.
Tregenza, William Edward ('43) (CM), Pres., Engr., Canadian Engrg. & Tool Co. Ltd., 920 Mercer St., Windsor, Ont., Can.; *for mail*, 80 Winder St., Detroit, Mich.
Trego, James Theodore (J'42) (ZSFE), Mech. & Field Engr., Bailey Meter Co., 30 Church St., New York 7; *for mail*, 323 W. 89th St., New York 24, N.Y.
Treiber, Kenneth Louis ('32; '41) (FHJNCY), Maj., Corps of Engrs., U.S.A., Chief, Petroleum Distribution Branch, Army Serv. Forces, Engrg. Bd., Ft. Belvoir, Va.
Treiber, Otis Daniel ('38) (ESFBOC), Cons. Engr., Hercules Motors Corp., Canton 2; *for mail*, 1414 Ridge Rd., Canton 3, Ohio.
Treloar, James Bennett (J'36), Process Engr., Somerville Ltd.; *for mail*, Apt. 5, 49 Ridout St. S., London, Ont., Can.
Trent, Clarence E. ('39) (BGM), Asst. Prof. Engrg., Mgr., Bluefield Branch, Va. Poly. Inst.; *for mail*, Box 423, Blacksburg, Va.
Trescott, Donald Albert (J'36) (SJMCOB), Sales Mgr., Cent. Corp., 20 N. Wacker Dr., Chicago 6; *for mail*, 7609 Clyde Ave., Chicago 49, Ill.
Treshow, Michael ('31) (OMFKE), 33222 W. Malibu Rd., Cornell, Calif.
Tresilian, Stewart S. ('35; '35), Home Lea, Chellaston, Near Derby, England.
Trethaway, Joseph D. ('28; '35) (JMCOZ), Mgr., Alloy Sales, Cerro de Pasco Copper Corp., 40 Wall St., New York 5, N.Y.; *for mail*, 34 Watchung Ave., Montclair, N.J.
Trethaway, William, Jr. ('24; '26; '35), 1444 Shore Rd., Linwood, N.J.
Trewin, Chas. S. ('38), Asst. Ch. Engr., N.J. Zinc Co., Palmerton, Pa.; *for mail*, 21 Myrtle Ave., Plainfield, N.J.
Tribus, Myron (J'44) (KAH), Research Engr., Univ. of Calif. at Los Angeles, Westwood Village, Los Angeles, Calif.
Trick, Charles John (J'44) (CLM), 3457 Richard, Pittsfield Village, Ann Arbor, Mich.
Trickett, Joseph Marshall (A'46) (COZ), Ch. Organization Engr., Columbia Steel Co., Russ Bldg., San Francisco 6, Calif.
Trier, James John (J'44) (NMCGJH), Designing Engr., R. Hoe & Co., 138th St. & East River, Bronx; *for mail*, 1525 Jarvis Ave., Bronx 61, N.Y.
Trigger, Kenneth J. ('38; '43) (JML), Prof., Mech. Engrg., Univ. of Ill., Urbana, Ill.
Trimmer, Chas. A. ('39) (EHOS), Cons. Engr., Hotel Lawler, Mitchell, S.D.
Trinkle, Robert Jacob ('30; '35) (SEFALM), Prof. Mech. Engrg., Va. Military Inst.; *for mail*, 606 Stonewall St., Lexington, Va.
Trinks, W. ('05) (KJZOF), Cons. Engr., Associated Engrs., 1423 Farmers Bank Bldg., Pittsburgh 22, Pa.
Triolo, Joseph Francis (J'40) (ZSF), Sales Serv. Engr., Bailey Meter Co., 403 City Centre Bldg., Philadelphia 7; *for mail*, 105 N. Lansdowne Ave., Lansdowne, Pa.
Triplett, John Louis ('40) (RBJ), Engrg. Asst. to Supt. of M.P., West. Md. Ry. Co.; *for mail*, 819 Woodland Way, Hagerstown, Md.
Tripp, Wilson ('38; '42) (KSBHEA), Assoc. Prof. Mech. Engrg., Kan. State College, Manhattan, Kan.
Trishman, Harry A. ('31; '35) (YNBCJM), Ch. Engr., William R. Thropp & Sons Co., 968 E. State St., Trenton 5, N.J.
Tritle, Edward M. (J'41) (ABE), Mech. Engr., Supercharger Engrg. Dept., Gen. Elec. Co., 920 Western Ave.; *for mail*, 12 Atlantic St., Lynn, Mass.
Tritton, Julian S. ('40) (DER), Cons. Engr., Rendel, Palmer & Tritton, 55 Broadway, Westminster, S.W.1, England.
Trofimov, Lev Alexander ('45) (O), Cons. Engr., Route 1, Hanna Rd., Willoughby, Ohio.
Troger, George F. (J'36) (LDSK), Designing Engr., Colgate-Palmolive-Peet Co., 105 Hudson St., Jersey City; *for mail*, 123 Baker St., Maplewood, N.J.

Troger, Henry Herman ('27; '46) (CDM), Ch. Prod. Planner, Fed. Shipbldg. & Dry Dock Co., Lincoln Highway, Kearny, N.J.
Troller, Theodor Henry ('41) (ABKSO), Prof. Aero. Engrg., Case Sch. of Applied Sci., Univ. Circle, Cleveland; *for mail,* 1250 Sunsetview Dr. W., Akron 2, Ohio.
Trosper, Ralph S. ('30; '35) (AEFCKO), Prof. Mech. Engrg., Head, Mech. Engrg. Dept., Univ. of Louisville, Belknap Campus, Louisville, Ky.
Troth, H. C. ('36; '42) (CMK), Pres., Gen. Mgr., Auto Coolers Inc., Pacific Mfg. Corp., 5308 Blanche Ave., S.E., Cleveland 4, Ohio.
Trott, Daniel Lloyd (J'43) (DMC), Pvt., 12th Bn., T343, 69th QM. Trng. Co., Camp Lee, Va.
Trotter, Arthur Halsey ('14) (DFKJHS), Plant Engr., Solvay Process Co., Hopewell, Va.
Trotter, Richard Adelbert ('24; '32) (BNRMSD), Assoc. Prof. Mech. Engrg., Ga. Sch. of Tech., Atlanta, Ga.
Troup, John Dawson ('28) (FSKZDE), Managing Dir., Engrg. Publications, John D. Troup Ltd., 90 High St., Holborn, London, W.C.I., England.
Trout, Charles L. ('46) (FPS), Engrg. Draftsman, Atlantic Refining Co., 260 S. Broad St., Philadelphia; *for mail,* 241 Barrington Rd., Upper Darby, Pa.
Trout, Walter C. (A'19), Pres., Mgr., Lufkin Fdy. & Mch. Co., Lufkin, Tex.
Trowbridge, Francis C. (J'89), Retired; 723 Dayton St., Hamilton, Ohio.
Troy, Max ('41), Gen. Linen Supply & Laundry Co., 835 Myrtle Ave., Brooklyn; *for mail,* 51—5th Ave., New York, N.Y.
Truax, William H. ('42) (CM), Commr. of Conciliation, U.S. Conciliation Serv., Washington 25, D.C.; *for mail,* 32 Arlington Rd., Stamford, Conn.
Trudel, Allen Robert (J'44), Lt. (j.g.), U.S.N.R., U.S.S. *Knapp,* DD-653, F.P.O., San Francisco, Calif.
True, Charles Henderson ('13; F'45) (CRS), V.P., Superheater Co., 151st St. & Railroad Ave., East Chicago, Ind.
True, Laurence Melbern ('32; '35) (CB), Methods & Equip. Dept., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3; *for mail,* 2915 Courtland Blvd., Shaker Heights 22, Ohio.
True, M. Austin ('45) (CS), Supvr. Mech. Equip., Dept. of Pub. Safety, City of St. Louis, 426 City Hall, St. Louis 3; *for mail,* 4247 Louisiana Ave., St. Louis 11, Mo.
Truedsson, Gosta Ragnar ('34; '39) (OMHJNB), Cons. Engr. & Physicist, 311 S. Main St., Holden, Mass.
Trueheart, Harry P., Jr. (J'39) (CJM), Indus. Engr., Time & Motion Study, Eastman Kodak Co., 343 State St.; *for mail,* 275 Susquehanna Rd., Rochester, N.Y.
Truett, Bertram Strelau (J'36) (CEMYZH), Lt. Comdr., U.S.N., 600 Bryant St., San Francisco 7; *for mail,* 1356 Drake, Burlingame, Calif.
Truitt, John Robert (J'43), Prod. Foreman, Truitt Mfg. Co.; *for mail,* 405 E. Whittington St., Greensboro, N.C.
Truman, Frederick ('36; '46) (CJMSZ), Ch. Engr., James Morrison Brass Mfg. Co. Ltd., 276 King St. W.; *for mail,* 284 Deloraine Ave., Toronto, Ont., Can.
Trumbull, Alonzo G. ('27) (RBJKS), Gen. Mech. Engr., 1038 Midland Bldg., Cleveland; *for mail,* Park Lane Villa, Cleveland 6, Ohio.
Trummel, John Merle (J'44) (BNE), Asst. Prof. Mech. Engrg., Engrg. Bldg., State Univ. of Iowa, Iowa City, Iowa.
Trump, Harry Walker ('28) (ENBAR), Dist. Mgr., Timken Roller Bearing Co., 409 Olive St., Dallas 1, Tex.
Trumpler, Alfred L. (J'40) (LYTKCA), Sales Engr., Dicalite Co., 1411 Walnut St., Philadelphia 2; *for mail,* 569 Fairway Terrace, Philadelphia 28, Pa.
Trumpler, Paul Robert (J'36) (KSEBF), Heat Transfer Devel. Engr., M.W. Kellogg Co., 225 Broadway, New York 7, N.Y.; *for mail,* 252 Lincoln Ave., Ridgewood, N.J.
Trumpler, William E. ('40) (ENHSB), Engr., Clark Bros. Co., Inc.; *for mail,* 710 Front St., Olean, N.Y.
Trumpler, William E., Jr. (J'37) (BJS), Palmer Mill Rd., R.D. 1, Media, Pa.
Trundle, George Thomas, Jr. ('41) (CDMNWY), Pres., Trundle Engrg. Co., 901 Bulkley Bldg., 1501 Euclid Ave., Cleveland 15, Ohio.
Trusler, Walter Ralph (J'37) (CDFKLY), Works Engr., Canadian Natl. Carbon Co. Ltd., 805 Davenport Rd., Toronto 4, Ont., Can.
Truxal, Orin S. ('21; '35) (NBDJMH), Mech. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail,* Box 156, Manor, Pa.
Tschappat, William Harvey (H'38) (BEST), Maj. Gen., U.S.A., Retired; Box 264, Falls Church, Va.
Tschorn, Frank Henry (J'31) (CDFKMS), Supt. Bldgs. & Grounds, Bennington College Corp., Bennington, Vt.
Tse, Foo-Sing (J'42) (CMOR), Student Grad. Div., Wharton Sch., Univ. of Pa.; *for mail,* 226 S. 38th St., Philadelphia 4, Pa.
Tucker, Alfred Scott (J'44) (ACJ), 1472 Filbert St., San Francisco 9, Calif.
Tucker, Carl Leonard (J'43) (BKJNAF), Design Engr., Automatic Heating Equip. Sec., Gen. Elec. Co., 7 Lawrence St., Bloomfield; *for mail,* 227 N. Walnut St., East Orange, N.J.
Tucker, Carroll Clayton ('38) (CMJDB), Factory Mgr., Matthews Mfg. Co., 104 Gold St., Worcester 3, Mass.
Tucker, David Anderson ('38) (SHR), Engr., Mech. Div., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 7; *for mail,* 116 E. Providence Rd., Lansdowne, Pa.
Tucker, Jesse Mack ('32; '40) (BKS), Assoc. Prof. Mech. Engrg., Univ. of Tenn., Estabrook Hall, Knoxville 16, Tenn.
Tucker, John Bonar (J'44), 2141 Campbell Ave., Schenectady 6, N.Y.
Tucker, Raymond Roche ('41) (EFS), Prof., Washington Univ., St. Louis, Mo.
Tucker, Robert G. (J'37) (FK), Capt., Trans. Corps, U.S.A., N.Y. Port of Embarkation, Brooklyn; *for mail,* 35-45—81st St., Jackson Heights, L.I., N.Y.
Tucker, Robert Nathaniel ('28; '35) (SCFEK), Supt., Div. of Elec., City of Columbus, 90 W. Broad St., Columbus 15; *for mail,* 1520 Bryden Rd., Columbus 5, Ohio.
Tucker, Stanley A. (J'36) (FKS), Assoc. Editor, *Power,* McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York; *for mail,* 21 Rockville Dr., Baldwin, L.I., N.Y.
Tucker, William B. ('23; '30; '35), Mgr., Aircraft Div., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail,* 3154 N. 54th St., Milwaukee 10, Wis.
Tuechter, August Herman (A'18) (CJN), Pres., Cincinnati Bickford Tool Co., Oakley, Cincinnati 9, Ohio.
Tuininga, Pieter (J'40) (FS), Power Engr., Curaçaosche Petroleum Industrie Maatschappy; *for mail,* Emmastad 10, Curaçao, N.W.I.
Tullar, Irving V. (J'33) (HKS), Creole Petroleum Corp., Sombrero Guarico, Venezuela, S.A.
Tulus, Eugene A. ('36) (BHKALS), Mgr., Research Dept., Nash Kelvinator Corp., 14250 Plymouth St., Detroit 32; *for mail,* 18176 Stout Ave., Detroit 19, Mich.
Tupling, Charles Gordon (J'37) (BCM), Sales Engr., Bearing Sales & Serv., Inc., 1109 N.W. Gilson, Portland; *for mail,* Box 99, Route 5, Portland 1, Ore.
Tupper, Edward B. (J'42), Aircraft Inspr., Boeing Aircraft Co.; *for mail,* 6231—34th Ave., N.E., Seattle 5, Wash.
Turck, Fenton Benedict ('43) (CJL), Pres., Turck, Hill & Co., Inc., 10 Rockefeller Plaza; *for mail,* 44 E. 67th St., New York, N.Y.
Turk, J. Clay (J'43) (HKM), Ch. Engr., Charge of Design, Reynolds Mfg. Co., 600 N. Prospect; *for mail,* Box 818, R.F.D. 8, 2005 S. Ferguson, Springfield, Mo.
Turley, Comer ('46), Mech. Supt., Pepperell Mfg. Co., Lindale, Ga.
Turley, H. T. (J'40) (AHJ), Hydro-Elec. Power Comm.; *for mail,* Fitzroy Harbour, Ont., Can.
Turnbull, Geo. B. ('14; '23), V.P., Gen. Mgr., Great Lakes Engrg. Works, River Rouge, Mich.
Turnbull, William A., Jr. (J'39) (CES), Power Plant Engr., Am. Potash & Chem. Corp.; *for mail,* Box 1365, Trona, Calif.
Turner, Arthur Dixon (J'43) (BHKARS), Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 6 Catherine St., Schenectady, N.Y.
Turner, Channing ('16; '25) (CMJK), Business Mgr., Radiation Lab., Mass. Inst. of Tech., 77 Massachusetts Ave., Cambridge 39, Mass.
Turner, Charles Huntington ('05; '17) (RJMB), Prin. Engr., Pullman-Stand. Car Mfg. Co., 27 W. Mountain St., Worcester; *for mail,* 16 Dean St., Worcester 2, Mass.
Turner, Charles Philip (A'07) (SB), Retired; Butler, Ill.
Turner, Edwin Archer ('40) (RC), Pres., Stand. Stoker Co., Inc., 350 Madison Ave., New York 17, N.Y.
Turner, Ernest Sidney (J'38) (BHS), Asst. Research Engr., Natl. Research Council of Can., Montreal Rd.; *for mail,* 184 Lisgar St., Ottawa, Ont., Can.
Turner, Frank B. ('28; '38) (FOEWKC), Engr., Supvr., The Budget Bur., State of N.C.; *for mail,* Box 6165, Five Points Sta., Raleigh, N.C.
Turner, John Wallace ('41) (SK), Ch. Engr., Pacific Steel Boiler Div., U.S. Radiator Corp., 1500 United Artists Bldg., Detroit 26, Mich.
Turner, Lewis (J'26) (CMJD), V.P., Gen. Mgr., Dollin Corp., 600 S. 21st St., Irvington 11, N.J.
Turner, Mahlon C. ('45) (BGJ), Design Engr., Mch. Design, S. M. Langston Co., 6th & Jefferson St., Camden; *for mail,* 216—4th Ave., Haddon Heights, N.J.
Turner, Mebane Ewing ('45) (CDLMNZ), Mech. Engr., Leaf Processing, R. J. Reynolds Tobacco Co., Winston-Salem; *for mail,* 814 Roslyn Rd., Winston-Salem 5, N.C.
Turner, Ralph Emerson ('15; '24) (SFEZKH), Editor, *Power Plant Engineering,* Tech. Publ. Co., 53 W. Jackson Blvd., Chicago 4; *for mail,* 4313 Central Ave., Western Springs, Ill.
Turner, Richard Harold ('43) (C), Pres., Mgr., Falconer Plate Glass Corp., Falconer, N.Y.
Turner, Richard S. (J'39), Capt., Field Arty., 96th Ord. Heavy Maint. Co., Ft. Bragg, N.C.
Turner, Robert Edwin ('39) (HSCZOJ), Hydrographer, Susquehanna Elec. Co., Conowingo, Md.
Turner, William Emerson (J'44) (MNRDCG), Chassis Designer, Autocar Co., Ardmore; *for mail,* 2742 Prescott Rd., Havertown, Pa.
Turner, Wm. W., Jr. (J'36), Lt., U.S.N., Blimp Sqd. 12, F.P.O., New York, N.Y.; *for mail,* 101 Clinton Ave., Montclair, N.J.
Turno, Walter G. ('13; '16; '21), 71 Lafayette Ave., East Orange, N.J.
Turnwald, Wolfgang ('23) (SRBE), Nordberg Mfg. Co.; *for mail,* 2980 S. Logan Ave., Milwaukee 7, Wis.
Turpin, Wayne D. (J'30) (SKFZD), Partner, Lee, Pace & Turpin, 142 S. 5th W., Salt Lake City 1, Utah.
Turton, James George ('44) (ACDHMY), Tech. Rep., Lockheed Hyd. Brake Co. Ltd., Tachbrook Rd., Leamington Spa, Warwicks, England; *for mail,* 8-10 Bridge St., New York 4, N.Y.
Turzicky, Francis C. ('27) (BJM), Engr., Ill. Malleable Iron Co., 1801 Diversey Pkwy., Chicago 14, Ill.
Tuthill, Elmer Sprague (J'23) (YWAG), Arch., Engr., 43 Maple St., Summit; *for mail,* 69 S. Oraton Pkwy., East Orange, N.J.
Tutt, Charles Leaming, Jr. ('33; '41) (NMEJYA), Asst. to Dir., Gen. Motors Inst., Flint 2; *for mail,* 3401 Westwood Pkwy., Flint, Mich.
Tuttle, Aubrey Stephen (J'43) (MJOB), Mech. Engr., Internatl. Nickel Co. of Can. Ltd., 25 King St. W., Toronto 1, Ont., Can.
Tuttle, George Woodbury ('42) (ACH), Reece Button Hole Mch. Co., 502 Harrison Ave., Boston; *for mail,* 7 Denton Rd., Wellesley, Mass.
Tuttle, Irving Edwin ('09; '15; '26) (HSLO), Mech. Engr., Contrg., Otto Epstein Inc., 40 Elm St., Hartford; *for mail,* R.R. 3, Tuttle Rd., Wallingford, Conn.
Tuttle, Norman J. (J'33) (SLFDMO), Asst. Ch. Engr., Continental-Diamond Fibre Co., S. Chapel St.; *for mail,* 216 Kells Ave., Newark, Del.
Tuttle, R. B. ('37), *The Oil & Gas Journal,* Tulsa, Okla.
Tuttle, Robert Allan (J'45) (EKHS), Ensign, U.S.N.R., YMS-306, c/o Postmaster, San Francisco; *for mail,* 5680 Ocean View Dr., Oakland 11, Calif.
Tuttle, William Buckhout ('05; F'45) (ESHC), Chmn., Bd. of Trustees, City Pub. Serv. Bd., Box 1771, 201 N. St. Mary's St., San Antonio 6, Tex.
Tuve, George Lewis ('22; '26; '33) (KZFMEA), Prof. Mech. Engrg., Head, Dept. of Mech. Engrg., Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6, Ohio.
Tuya, Eduardo, Jr. (J'42) (ASR), Engr., Antilles Div. Engr. Office, Fernandez Juncos Ave., Stop 7½, San Juan; *for mail,* Marina 13, Ponce, P.R.
Twaddell, Russell W. (J'35) (JMW), Draftsman, Anaconda Wire & Cable Co., Hastings-on-Hudson; *for mail,* 27 De Witt Dr., Sidney, N.Y.
Twinberrow, J. O. ('29), Sales Engr., Babcock & Wilcox Ltd., 32 Farringdon St., London, E.C. 4; *for mail,* 17 Wendover Rd., Bromley, Kent, England.
Twining, Frederic Edward ('37) (DEFJKL), Owner, Twining Labs., 2527 Fresno St., Box 1472, Fresno 16, Calif.
Twist, Howard E. (J'37), Mech. Engr., Pierce Foundation, 601 W. 26th St., New York, N.Y.; *for mail,* 293 W. Passaic Ave., Rutherford, N.J.
Twitchell, Clarence H. ('21; '35) (JL), Mech. Engr., Nitrogen Div., Solvay Process Co., Hopewell, Va.
Twombly, Robert Cantrell (J'45), 16 Lawson Rd., Winchester, Mass.
Tydeman, W. A. ('41), 129 Ferry St., Easton, Pa.
Tylaska, Ted Thomas ('39; '45) (CHJMN), Asst. Gen. Supt., A. O. Smith Corp., Houston; *for mail,* 6704 Community, Houston 5, Tex.
Tyler, Clarence Mallory, Jr. (J'42) (ABN), Graduate Asst., Aero. Engrg. Dept., Univ. of Pittsburgh, 213 State Hall, Pittsburgh 13; *for mail,* 583 East End Ave., Pittsburgh 21, Pa.
Tyler, Frederick G. (J'41) (AH), Lt. (j.g.), U.S.N.R., VF-80, F.P.O., New York, N.Y.
Tyler, Walter Duane ('42) (BNJHRA), Ch. Estimator, N.Y. Air Brake Co.; *for mail,* 201 California Ave., Watertown, N.Y.
Tymstra, Sybren Ruurd ('40) (BGK), Prof. Mech. Engrg., Univ. of Wash., Seattle 5, Wash.
Tyren, Ted T. (J'40) (DJM), Engr., Charge of Design & Estimating Htg., Durham Plumbing & Htg. Co.; *for mail,* Cole Rd., Durham, N.C.
Tyroff, Carl E. ('22; '35), Constr. Engr., Stand. Oil Co. of N.J., Elizabeth; *for mail,* 247 W. 4th Ave., Roselle, N.J.
Tyrrell, Cecil Clough ('42) (FKNS), Sr. Mech. Engr., Anthracite Inst., Primos, Del. Co.; *for mail,* 115 School Lane, Springfield, Pa.

U

Udale, Stanley Martin ('30) (AHK), Pat. Atty., Holley Carburetor Co., Detroit 4; *for mail,* 893 Lothrop Ave., Detroit, Mich.
Udall, Phillip Aderzuch ('39) (AMZ), Pres., Gen. Mgr., Udall & Landan Mfg. Co., Inc., 295 Lafayette St., New York, N.Y.
Udstad Sigvald F. ('44) (BJR), Asst. to Gen. Mech., Am. Car & Fdy. Co., Berwick 6; *for mail,* 139 E. 7th St., Berwick, Pa.
Uehling, Ed. A. ('83) (A), Retired; 2360 S. 81st St., West Allis, Wis.
Uehling, F. F. ('18), Pres., Combustion Engr., Uehling Instrument Co., 473 Getty Ave., Paterson; *for mail,* 86 Park Ave., Passaic, N.J.
Uehling, Vyrle Martin (J'39) (CDM), Engr., Hastings Grain Bin Mfg. Co.; *for mail,* 1715 W. 5th St., Hastings, Neb.

Ugalde, Hermann Herrera (J'43) (AHRSTW), Student, Ga. Sch. of Tech.; *for mail,* 279—5th St., N.W., Atlanta, Ga.
Uhde, William (J'36) (JLM), Tool Engr., Hyatt Roller Bearing Div., Gen. Motors Corp., 4th St., Harrison, N.J.; *for mail,* 8210 Lefferts Blvd., Kew Gardens, N.Y.
Uhl, William F. ('12) (SHOE), Pres., Chas. T. Main, Inc., 201 Devonshire St., Boston 10, Mass.
Uhle, David John ('45) (BEFKNS), V.P., Coplay Cement Mfg. Co., Coplay; *for mail,* 1411 Hamilton St., Allentown, Pa.
Uhler, Wilmer Paul ('41) (BCJMNY), Ch. Engr., Asst. Mgr., S. S. White Dental Mfg. Co., Prince Bay 9; *for mail,* 111 Patton St., Tottenville 7, N.Y.
Uhlhorn, William F. (J'45) (CJS), 3751—89th St., Jackson Heights, N.Y.
Uicker, George Bernard ('39; '46) (SJMKH), Instr., Mech. Engrg. Dept., Swarthmore College, Swarthmore, Pa.; *for mail,* 78 Crescent Ave., Woodbury, N.J.
Uicker, John Joseph ('32; '38) (KSEFB), Asst. Prof., Acting Dir., Mech. Engrg. Dept., Univ. of Detroit, McNichols Rd. at Livernois, Detroit 21; *for mail,* 16261 Littlefield Ave., Detroit 27, Mich.
Uihlein, H. C. ('19; '35), Secy., Treas., Cincinnati Engrg. Tool Co., 1241 Knowlton St.; *for mail,* 704 Wakefield Dr., Cincinnati, Ohio.
Ulander, Per Helge Natanael ('45), Dir., James Howden Co., Ltd., 195 Scotland St., Glasgow, C.5; *for mail,* Busby House, Busby, Lanarkshire, Scotland.
Ulanowsky, Sam (J'41) (LKSEH), Piping Designer, Lummus Co., 420 Lexington Ave., New York 17; *for mail,* 4504—15th Ave., Brooklyn 19, N.Y.
Ulbert, August (J'42) (HKMZ), East. Mgr., Alco Valve Co., 122 E. 42nd St., New York 17, N.Y.
Ullman, John Ralph (J'45) (BHM), 1108 N.E. Halsey St., Portland 12, Ore.
Ullrich, Walter (J'43) (CDMS), Supt. Asst., Maint., Consltd. Edison Co. of N.Y., Inc., 20th Ave. & 21st St., Astoria; *for mail,* 309 E. Mosholu Pkwy., New York 67, N.Y.
Ulmann, Edward Frazer (J'43) (SJKCEF), Research Project Engr., Langley Memorial Lab., Natl. Advis. Com. for Aero.; *for mail,* Apt. 528-B Southhampton Apts., Elizabeth Rd., Hampton, Va.
Ulrich, Fred L. (J'42), Ensign, U.S.N.R., 117 Kensington Ave., Jersey City, N.J.
Ulrich, R. ('40) (CEFRSZ), V.P., Bacharach Indus. Instrument Co., 7000 Bennett St., Pittsburgh 8; *for mail,* 304 Edgewood Rd., Pittsburgh 21, Pa.
Ulrich, Woodrow Wilson (J'44) (ANMBG), Ch. Draftsman, Tool Design, Perfect Circle Co., Hagerstown, Ind.
Umbehocker, Frank ('21; '29) (SFCJKE), Ch. Oper. Engr., Chicago Dist. Elec. Generating Corp., 122 S. Michigan Ave., Chicago 3, Ill.; *for mail,* 47-172nd Pl., Hammond, Ind.
Umeda, Shigetoki (J'40) (ABL), 425 Euclid Ave., Los Angeles 33, Calif.
Umerez-Blanco, Fco ('32; '42) (DEHLSW), Managing Dir., C. A. Tecnomat, Apartado 108, Caracas, Venezuela, S.A.
Umland, Christian John, Jr. (J'44), 430 E. 149th St., Bronx 55, N.Y.
Underberg, Samuel (J'43) (ACW), 44 Hanson Pl., Brooklyn 17, N.Y.
Underwood, Arthur F. ('45) (BEJAM), Head, Mech. Engrg. Dept. No. 5, Research Labs. Div., Gen. Motors Corp., Box 188, N. End Sta., Detroit 2, Mich.
Underwood, Carl M. ('45), Mgr., Welding Dept., No. Ord. Inc.; *for mail,* 25 Westwood Dr., Minneapolis, Minn.
Ungar, Gustave Adolphe ('39) (HMKABO), V.P., Ch. Engr., Equi-Flow, Inc., 1841 Broadway, New York 23, N.Y.
Unger, Fred Charles (J'44), 109-65—201st St., Hollis, L.I., N.Y.
Unger, Louis Frank ('33; '43) (MCLB), Capt., Ord. Dept., U.S.A., Ord. Dist., Philadelphia 2; *for mail,* 809 N. 8th St., Allentown, Pa.
Unger, Paul Raymond ('44) (EFCSK), Partner, Combustion Engr., Mid-West Heat Serv., 3336 Franklin Blvd., Chicago 24, Ill.
Ungerleider, Samuel Arthur ('46) (CDL), Tech. Appraiser, Surplus Properties & Reconversion, War Assets Admin., 52 Wall St., New York, N.Y.; *for mail,* 60 N. Arlington Ave., East Orange, N.J.
Unkles, Edward Harrison ('28), Mgr., Film Splicer Div., Bloomfield Tool Corp., 35 Farrand St., Bloomfield; *for mail,* 883 Boulevard East, Weehawken, N.J.
Unofsky, Jack (J'41) (ABE), 1009 Rivington St., Roselle, N.J.
Updike, David M. ('20) (FKSE), Combustion Engr., Bradley Mahony Coal Co., 223 E. 129th St., New York, N.Y.
Updike, Ronald W. (J'34) (MCEJ), Tool Supvr., Schwitzer-Cummins Co., 1125 Massachusetts Ave., Indianapolis 7; *for mail,* 5201 W. 16th St., Indianapolis 8, Ind.
Upham, Kirtland Hall (J'45) (HSKN), Ensign, U.S.N.R., Engrg. Officer, PCE(R)-853, F.P.O., New York, N.Y.; *for mail,* 31 Washington Pl., Meriden, Conn.
Upp, Raymond Ferguson (J'43) (AJCNE), Lt., Army Air Forces, 61st Troop Carrier Sqd., 441st Troop Carrier Gp., A.P.O. 57, c/o Postmaster, New York, N.Y.
Upson, Charles Ayrault ('43) (CNW), Pres., Upson Co., Upson Point; *for mail,* 453 Willow St., Lockport, N.Y.
Upson, Maxwell Mayhew ('01; '07; '12; F'41), Pres., Raymond Concrete Pile Co., 140 Cedar St., New York 6, N.Y.
Upson, Robert Seymour (J'38) (MCJ), Prod. Engr., Gen. Elec. Co., 2109 E. Lake Rd., Erie; *for mail,* R.D. 1, North Girard, Pa.
Upton, Sylvester James (J'43) (ACJ), Lt., Army Air Forces, Aircraft Maint. Div., Army Air Field, Charleston, S.C.
Ur, Joseph Andre (J'44) (BJSCNM), Draftsman, Alloy Steel Products Co., Inc., 1300 W. Elizabeth Ave., Linden; *for mail,* 145 Chadwick Ave., Newark 8, N.J.
Uran, Nejdet F. (J'46), 27, Aci Badem Yolu 27, Kadikoy-Istanbul, Turkey.
Urbach, Henry Harley ('42) (R), Mech. Asst. to Exec. V.P., Chicago, Burlington & Quincy R.R. Co., 547 W. Jackson Blvd., Chicago 6; *for mail,* 2523 Central St., Evanston, Ill.
Urban, Fred Oberg ('45) (B), Design Engrg., Air Conditioning, Gen. Elec. Co., 5 Lawrence St., Bloomfield, N.J.
Urbauer, Hugo F. ('23; '35), Chmn., Bd. of Dirs., Midwest Piping & Supply Co., Inc., 1450 S. 2nd St., St. Louis, Mo.
Urech, Henry Herman (J'41), Pump Engr., Magnolia Petroleum Co.; *for mail,* 2031 Park St., Beaumont, Tex.
Urquhart, Noel ('45) (CZBNGJ), Tech. Dir., Heyer Industries Inc., 14 Nassau Blvd., Garden City, N.Y.
Ursprung, Herman ('39) (CDFL), Supt., Lone Star Cement Corp., Nazareth, Pa.
Usdin, Eugene (J'44), Pvt., 1st Class, Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.
Usher, Ewart Sheldon (J'45), Rehabilitation Counselor (Engrg.), Mining Bldg., Univ. of Toronto; *for mail,* 166 Glenmanor Dr., Toronto 8, Ont., Can.
Usher, George Clemens ('22; '43) (CFMABD), Managing Dir., Internatl. Combustion, Ltd., 19 Woburn Pl., London, W.C.1, England.
Usher, Wellington James ('44) (CKS), Engr., Arthur S. Leitch Co. Ltd., 1123 Bay St., Toronto 5, Ont., Can.
Ustabashieff, Jordan Anastas (J'45) (ABH), 3194 Kestner Ave., Brentwood, Pittsburgh 10, Pa.
Ustin, Peter K. ('40) (JLKN), Engr., Shell Devel. Co., 4560 Horton St., Emeryville 8; *for mail,* 1858—38th Ave., San Francisco 22, Calif.
Utz, John Richard (J'45), Design Engr., Technifinish Lab., Inc., Rochester 3; *for mail,* 180 Nunda Blvd., Rochester 10, N.Y.

V

Vacca, Gennaro Anthony (J'41) (AHBENJ), Sr. Mech. Engr., Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail,* 4310 N. Henderson Rd., Arlington, Va.
Vail, Allan P. ('39) (BCS), Ch. Engr., Pub. Serv. Co. of Colo., Alamosa, Colo.
Vail, Carl Waldemar ('19; '24; '35) (CDMSZ), Plant Engr., Morse Chain Co., Box 568; *for mail,* 207 Ithaca Rd., Ithaca, N.Y.
Vail, David Porter ('20; '27) (SKEBFL), Mgr., Los Angeles Branch, C. C. Moore & Co., Engrs., Inc., 555 S. Flower St., Los Angeles 13, Calif.
Vaill, John Locke ('44) (NBHJMA), Owner, Vaill Engrg. Co., 70 N. Elm St., Waterbury 89; *for mail,* 68 Hewlett St., Waterbury 30, Conn.
Vais, Lawrence Andrew (J'42), Ensign, U.S.N.R., 180 Granada Ave., San Francisco 12, Calif.
Vaisman, William (J'45) (BHK), Staff Mem., Dept. of Math., Case Sch. of Applied Sci., University Circle, Cleveland, Ohio.
Vaklyes, John William, Jr. (J'41) (SAHMRK), Lt., U.S.N.R., 532 N. Bancroft Pkwy., Wilmington 137, Del.
Vaksdal, Alfred ('24) (CDEFMS), Plant Engr., Corning Glass Works, Corning, N.Y.
Vaksdal, Steiner R. ('35; '44) (CDS), Hotel Murray Hill, Westfield, N.Y.
Valaitis, Anthony Frank (J'42) (MJO), Instr. in Mech. Trades, West. Elec. Co., Inc., 22nd & 48th St., Hawthorne 23; *for mail,* 2525 W. 70th St., Chicago 29, Ill.
Valentine, Deane Beers (J'41) (CDJKMY), Indus. Engr., Minnequa Plant, Colo. Fuel & Iron Corp.; *for mail,* 141 Goodnight Ave., Pueblo, Colo.
Valentine, Kendall Comstock (J'40) (BJS), Lt. (j.g.), U.S.N.R., U.S.S. *John J. Powers,* DE-528, F.P.O., New York, N.Y.; *for mail,* 777 East St., Walpole, Mass.
Valentyne, Peter Henry (J'41) (ASM), 2118 Delaware Blvd., Saginaw, Mich.
Valeur-Jensen, Soren ('40) (NMLDFS), Ch. Engr., F. L. Smidth & Co., 11 W. 42nd St., New York 18, N.Y.
Vallario, John Baptist, Jr. (J'42) (YNMLDO), Jr. Mech. Engr., Plant Engrg. Dept., Dominion Tire Factory, Strange St.; *for mail,* 103 Queen St. S., Kitchener, Ont., Can.
Valyi, Emery I. ('43) (BMJNL), Pres., ARD Corp., 65 Pine St., New York 5, N.Y.
Vana, John J. (J'41) (CMN), Mech. Engr., Inland Steel Co.; *for mail,* 1913 Broadway, East Chicago, Ind.
Vana, Vladimir Joseph (J'40) (ACF), 2248 S. Keeler Ave., Chicago 23, Ill.
Van Alstyne, John J. (J'40) (DMS), Lt. (j.g.), U.S.N.R., Constr. Bn. Maint. Unit 569, F.P.O., San Francisco, Calif.; *for mail,* Box 242, Sunnyside, Kinderhook, N.Y.
Vanaman, Francis H. ('29; '35) (AJM), Designer, Edw. G. Budd Mfg. Co., 25th St. & Hunting Park Ave.; *for mail,* 1053 Granite St., Philadelphia 24, Pa.
Van Auken, Howard Charles ('46) (COZ), Engrg. Mgr., Sperry Gyroscope Co., Inc., Great Neck; *for mail,* 57 Vanderbilt Rd., Manhasset, N.Y.
Van Blarcom, Gilbert Francis (A'45) (TCM), Treas., Ralph E. Loper Co., Box 648, Fall River, Mass.
Van Bomel, Leroy A. ('18), Natl. Dairy Products Corp., 230 Park Ave., New York, N.Y.
Van Brunt, John ('14) (FKOS), V.P., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Van Buskirk, Geo. L. ('21; '25; '35) (FMS), Routine Engr., M.P. Dept., Interborough Rapid Transit Co., 600 W. 59th St.; *for mail,* 340 W. 86th St., New York, N.Y.
Van Buskirk, James ('45) (CMZ), Indus. Engr., Chicago Ord. Dist., War Dept., 38 S. Dearborn St., Chicago 3; *for mail,* 531 Michigan Ave., Evanston, Ill.
Vance, Carl B. (J'44) (CSFZ), Mech. Engr., Gen. Plants Dept., Indianapolis Power & Light Co., 17 N. Meridian St.; *for mail,* 5016 Park Ave., Indianapolis 5, Ind.
Vance, David Albert (J'46), 2412 Carleton St., Berkeley 4, Calif.
Vance, Harold ('39) (CEFHK), Head, Petroleum Engrg. Dept., Agric. & Mech. College of Tex., College Station; *for mail,* 607 E. 32nd St., Bryan, Tex.
Vance, Ledcreich Stuart ('23; '24) (CH), 9' Hawthorne Hill, Louisville 4, Ky.
Vance, Warren Lewis (J'44) (ENFBH), A. B. Chance Co.; *for mail,* Box 85, Centralia, Mo.
Vancil, Edgar Donald ('29) (CHJN), Mgr. of Engrg., Cincinnati Milling Mch. Co., Marburg, Cincinnati 9; *for mail,* 30 Observatory Hill, Cincinnati 8, Ohio.
Van Dame, Fredrick Peter (J'44) (KADCN), Mech. Engr., Ready Power Co., 11231 Freud Ave., Detroit 14; *for mail,* 15711 Snowden Ave., Detroit 27, Mich.
Van de Cop, Karel ('22; '25; '35), Mech. Engr., Fruit Dispatch Co., Pier 7, N.R., New York, N.Y.; *for mail,* Harriet Ave., Harrington Park, N.J.
Vandegrift, Clarence Gilbert ('30; '46) (BMS), Assoc. Prof. Mech. Engrg., 204 Engrg. D, Pa. State College; *for mail,* 723 W. Nittany Ave., State College, Pa.
Van DeMark, Robert Lewis (J'42) (AMGCSY), Seaman, 1st Class, U.S.N., Co. 208, Billet 429, Navy Pier, Chicago, Ill.
Van Denburg, John W. ('38) (BDE), c/o Dept. of Pub. Works, 125 Worth St., New York, N.Y.
Vanden Heuvel, George R. (J'39), Lt., Army Air Forces, 109 Mountain View Ave., Staten Island, N.Y.
Van de Pol, Herbert (J'43) (ADM), Mech. Prod. Engr., Kimble Glass Co., Vineland; *for mail,* 265—92nd St., Stone Harbor, N.J.
Vanderhoof, Arnold Hines ('18) (HLOS), Cons. Engr., Paragon Bldg., Asheville, N.C.
van der Jagt, Barend Gysbertus Hendrikus ('43) (COT), Cons. Engr., 1727—16th St., Washington 20, D.C.
vander Straeten, Herman Joseph (J'44) (FER ASH), Gen. Labs., Socony-Vacuum Oil Co., Inc., Paulsboro, N.J.
Vander Veer, John Henry, Jr. (J'42) (BEP), Mech. Engr., Naval Ord. Field Engr., Sperry Gyroscope Co., Inc., Great Neck; *for mail,* Box 29, Bayville, L.I., N.Y.
Van Derveer, William Williamson ('43) (ACE JNS), Serv. Engr., Metalock Casting Repair Serv., 27-05—43rd Ave., Long Island City; *for mail,* 80-09—35th Ave., Jackson Heights, N.Y.
Vander Velde, Martin Hugo (J'38) (ZLKM), Sales Engr., Powers Regulator Co., 2720 Greenview, Chicago, Ill.
Vander Vennen, Roy J. (J'44) (NHJMYK), 1749 Coit Ave., N.E., Grand Rapids, Mich.
Van Dervort, Adrian Orsemus ('00; '18) (DKN), Mech. Engr., Behr-Manning Corp.; *for mail,* 146 Oakwood Ave., Troy, N.Y.
Vanderweil, Raimund G. ('44) (KSEOC), Project Engr., Heat Transfer, Chase Brass & Copper Co., Waterbury 91; *for mail,* Woodbury, Conn.
Van Deursen, Charles Augustus ('44) (CEKS), Mgr., Utility Bd. of the City of Key West, Fla., Duval & Greene, Key West, Fla.
Vande Ven, Alvin W. (J'38), Asst. Plant Engr., Buda Co., Harvey; *for mail,* 931 S. Clinton Ave., Oak Park, Ill.
Van Deventer, Frank M. ('18; '24; '26) (SFR BJH), Asst. Ch. Engr., Walworth Co., Inc., 60 E. 42nd St., New York 17, N.Y.
Van De Verg, Nathaniel (J'40) (HN), Research Engr., Jet Propulsion Lab., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4; *for mail,* 1226 Rosalind Rd., San Marino 5, Calif.

Van Dongen, Dirk (J'41) (EDKH), Mar. Engr., Dravo Corp., Pittsburgh 25; *for mail,* Box 221, Coraopolis, Pa.
Van Doren, William Durham ('21) (DFHKS), Mech. Engr., Barrett Div., Allied Chem. & Dye Corp., 40 Rector St., New York 6; *for mail,* 37-20—84th St., Jackson Heights, N.Y.
Van Driest, Edward Reginald ('44) (BHKA), Asst. Prof. Mech. Engrg., Mass. Inst. of Tech., Cambridge 39, Mass.
Van Dusen, C. Theron (J'20) (ACEHS), Engr., Power Sales Div., Detroit Edison Co., 2000—2nd St., Detroit 26, Mich.
Van Duzer, Robert M., Jr. ('38) (BCFJS), Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Van Dyck, William Cuyler (J'45), Supvr., College Graduate Trng., Caterpillar Tractor Co., Peoria 8, Ill.
Van Dyke, James Richard ('30; '34) (BEKNSA), Prof., Acting Head, Mech. Engrg. Dept., Univ. of Nev., Reno, Nev.
Van Dyke, Richard Vickery (J'38) (CELN), Fire Underwriter, Mill Mutuals, 3rd Natl. Bank Bldg., Nashville 3; *for mail,* 2129 Bernard Ave., Nashville 5, Tenn.
Vane, Francis F. (J'34) (ABJ), Sr. Physicist, David W. Taylor Model Basin, Bur. of Ships, Navy Dept., Washington, D.C.; *for mail,* Box 206, Virginia & Grandin Aves., Rockville, Md.
Vanek, Stanley Donald (J'41) (JKNSDA), Design Engr., with C. J. Nordstrom, Naval Arch., 905—2nd Ave. Bldg., Seattle; *for mail,* 1460 S.W. 151, Seattle 66, Wash.
Van Gilst, Peter C. ('39), V.P., Charge Opera., Ky. W.Va. Gas Co., 2nd Natl. Bank Bldg., Ashland, Ky.
Van Gorden, LeRoy C. ('38) (NBJZ), Ch. Draftsman, Mch. Design, Kimble Glass Co., Crystal Ave.; *for mail,* Box 89, Route 3, E. Walnut Rd., Vineland, N.J.
Van Hamersveld, John J. N. ('20; '31) (NHBAMJ), Mech. Engr., Pat. Dept., Warner & Swasey Co., 5701 Carnegie Ave., Cleveland 3, Ohio.
Van Hengel, Gerrit H. ('22; '27) (SKBHAJ), Mech. Engr., Prod. Dept., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail,* 19 Elm Park Blvd., Pleasant Ridge, Mich.
Vaniman, Russell Eldon (J'44) (MCBZJN), Gen. Mgr., Applied Physics Corp., 40 S. Oak Knoll, Pasadena 1; *for mail,* 760 E. Mountain St., Pasadena 6, Calif.
Vanis, Albert Anton (J'39) (CM), Supt., Ch. Engr., Craft Mfg. Co., 3949 W. Schubert Ave., Chicago 27; *for mail,* 1917 Maple Ave., Berwyn, Ill.
Van Kammen, Isaac John ('43), Commodore, U.S.N.R., 1552 Hall St., S.E., Grand Rapids 7, Mich.
Van Leer, Blake Ragsdale ('20; '25) (HCMO), Pres., Ga. Sch. of Tech., 225 North Ave.; *for mail,* 153—4th St., N.W., Atlanta, Ga.
Van Loo, Carl George (J'37) (DNMJ), Ch. Engr., Rapids Stand. Co., 535 Bond, Grand Rapids; *for mail,* 1716 Silver Ave., S.E., Grand Rapids 7, Mich.
Van Meter, Homer (J'42) (EARJMN), Gen. Mgr., Universal Diesel Products Co., 3607 Orchard Ave., Indianapolis, Ind.
Vann, Fred Ellis (J'39) (MDNCK), Mech. Engr., Stockham Pipe Fittings Co., 4000—10th Ave. N., Birmingham 2, Ala.
Van Norman, Frederick Dewey ('19), V.P., Van Norman Co., 3640 Main St., Springfield 7; *for mail,* 120 Clarendon St., Springfield 9, Mass.
Van Nostrand, Erwin Skidmore ('43) (CEFOS), Oper. Supt., Southwest. Pub. Serv. Co., 5th & Polk St., Amarillo, Tex.
Van Osten, Philip Gould (J'40) (NZBMA), Test Engr., Moore Products Co., H & Lycoming Sts., Philadelphia 24; *for mail,* 4556 Aldine St., Philadelphia 36, Pa.
Vanous, Charles Joseph ('37; '46) (ANHKE), Design Engr., Food Mchy. Corp., 6000 N. Figueroa St., Los Angeles 42; *for mail,* 760 S. Griffith Park Dr., Burbank, Calif.
Van Overveen, Joseph P. (J'40) (ERSCYF), Apt. 5-C, 83-06 Victor Ave., Elmhurst, N.Y.
Van Patter, Hugh Stanley ('46) (BHM), Ch. Engr., Hyd. Div., Dominion Engrg. Works, Ltd., Box 220, Montreal; *for mail,* 612 Belmont Ave., Westmount, Que., Can.
Van Riper, Francis H. ('41), Exec. to Dir. of Prod., U.S. Maritime Comm., Commerce Bldg.; *for mail,* 2817 Connecticut Ave., N.W., Washington, D.C.
Van Saun, W. G. (J'40), Asst. Gen. Mgr., Janesville Cotton Mills Co., 248 N. River St.; *for mail,* 18 Elm St., Janesville, Wis.
Van Schwartz, Zolly Carleton ('43; '44) (BJMN), Ch. Mchy. Engr., Peck, Stow & Wilcox Co., Southington; *for mail,* Russell Rd., Milldale, Conn.
Vanselow, John Clifford ('27; '35) (CMZ), Shop Supt., West. Elec. Co., Inc., 900 Laurel Ave., Minneapolis 3; *for mail,* 5245 Chowen Ave., Minneapolis 10, Minn.
Van Valkenburg, James Fredrick (J'43) (S), Inspr., Dept. W-20, Westinghouse Elec. Corp., Lester; *for mail,* 228 Lafayette Ave., Collingdale, Pa.
Van Valkenburgh, Hugh (J'38) (EKAS), Ch. Test Engr., Ranger Aircraft Engs. Div., Fairchild Eng. & Airplane Corp.; *for mail,* 177 Conklin St., Farmingdale, L.I., N.Y.
Van Valkenburgh, Robert Mills ('38; '46) (CMJSZ), Maj., Ord. Dept., U.S.A., Exec. Officer, Army Serv. Forces, Frankford Arsenal, Philadelphia 37; *for mail,* 1209 Wakeling St., Philadelphia 24, Pa.
Van Valzah, Robert W. ('42) (MDNJH), Plant Engr., Danly Mch. Specialties, Inc., 2100 Laramie, Cicero; *for mail,* 297 Northwood Rd., Riverside, Ill.
Van Vleet, James G. ('38; '43) (BNL), Ch. Engr., Blackthorne Co., South Milwaukee; *for mail,* 4063 N. 39th St., Milwaukee 9, Wis.
Van Vlerah, Sylvan ('39), Asst. Supvg. Engr., Detroit Bd. of Education, 1354 Broadway, Detroit 26; *for mail,* 14212 Piedmont St., Detroit 23, Mich.
van Voorhees, Richard Miller (J'44), c/o Marion U. van Voorhees, 105 Danbury Rd., Compo Beach, Westport, Conn.
van Wezel, Myndert (J'45) (SKJLZH), Engr., Measuring Dept., Curaçaosche Petroleum Industrie Maatschappÿ, Emmastad, Curaçao, N.W.I.
Van Wie, John Allen, Jr. (J'43) (CAMBJ), 492 Pawling Ave., Troy, N.Y.
Van Winkle, Edward M. ('44), East. Sales Mgr., Am. Steel Fdys., 230 Park Ave., New York 17, N.Y.
Van Wyck, Philip Sleight ('19; '27; '35) (ROMJ), Dir., Occupational Analysis & Indus. Services Div., U.S. Employment Serv., 1778 Pennsylvania Ave., N.W.; *for mail,* Apt. 327, 2500 Que St., N.W., Washington 7, D.C.
Van Yorx, John H., Jr. ('17), Tech. Sales Promotion, Bullard Mch. Tool Co., Bridgeport; *for mail,* Lordship Manor, 1st Ave., Stratford, Conn.
Van Zandt, Paul Claiborne ('00; '07; '09) (BCNZ), Retired; 156 N. Eleanor Pl., Peoria, Ill.
van Zelm, Henri Baldwin ('23; '35) (OS), Cons. Engr., 11 Asylum St., Hartford 3, Conn.
Van Zytveld, Roger Frank (J'41) (LKHJFS), Mech. Engr., White Star Div., Socony-Vacuum Oil Co., Inc.; *for mail,* 2232 Riverside Dr., Trenton, Mich.
Varady, John C. ('36) (CDMNO), Pres., Gen. Mgr., Varady Engrg. Serv., 507 West. Reserve Bldg., Cleveland 13; *for mail,* 1213 St. Charles, Lakewood 7, Ohio.
Varagona, Joseph J. (J'40) 1829—8th Ave. S., Birmingham, Ala.
Varani, Ubaldo Alfred (J'40) (ANBJKH), Mech. Engr., P-3, Aero. Exper. Lab., Naval Air Matl. Center, League Island, Philadelphia 12; *for mail,* 643 E. Woodlawn St., Philadelphia 44, Pa.
Varga, Gideon M. ('29; '40) (CDM), A. Schrader's Son Div., Scovill Mfg., Inc., 470 Vanderbilt Ave., Brooklyn; *for mail,* 500 St. John's Pl., Brooklyn 16, N.Y.
Varian, Howard M. ('16; '25) (SC), Ch. Engr., Great Lakes Engrg. Works, River Rouge 18; *for mail,* 937 S. Denwood Dr., Dearborn, Mich.
Varnado, Osmond D. M. ('45) (EFS), Sr. Research Engr., Charge of Devel. & Design, Reynolds Metals Co., 709 Taylor; *for mail,* 1012 Washington St., Corinth, Miss.
Varner, Ira Stanford (J'43) (AJSENK), Assoc. Engr., The Engr. Bd., War Dept., Ft. Belvoir; *for mail,* 3320-B S. Wakefield St., Arlington, Va.
Varner, Stanley Evert (J'42) (ABFHKZ), Sr. Engr., Unit Head, Wright Aero. Corp., Wood-Ridge; *for mail,* 293 Park St., Montclair, N.J.
Varnes, Samuel Kepner (J'10) (BLS), Ch. Engr., Ammonia Dept., E. I. du Pont de Nemours & Co., Wilmington; *for mail,* 500 Rockwood Rd., Wilmington 220, Del.
Varnum, Richard Saltonstall, Jr. (J'40) (NASKBO), Mech. Engr., P-3, Designer, Natl. Advis. Com. for Aero., Langley Field; *for mail,* 11 Langley Ave., Beaconsdale, Newport News, Va.
Varona, Miguel Chinchilla ('27; '35), Cons. Engr., Central Washington, Manacas, Las Villas, Cuba.
Vassar, Hervey Sackett ('14) (CGS), Lab. Engr., Pub. Serv. Elec. & Gas Co., 200 Boyden Ave., Maplewood, N.J.
Vasselli, Anthony J. ('20; '35), Mgr., Zenith Engrg. & Mch. Co., 46 Clinton St.; *for mail,* 85 Hedden Terrace, Newark, N.J.
Vassily, George Raymond (J'41) (MCKH), Aviation Electronics Technician's Mate, 3rd Class, U.S.N., Naval Air Sta., Box 43, Miami, Fla.; *for mail,* 497 Oakdene Ave., Ridgefield, N.J.
Vaughan, James Wilson, Jr. ('36) (TBHLMW), Pres., Treas., J. W. Vaughan Co., Box 627, Greenville, S.C.
Vaughan, John F. ('17) (ACHRS), Retired; 21 Cedar Rd., Belmont 78, Mass.
Vaughan, Lillian Lee ('16; '21) (FESKHC), Prof. & Head, Mech. Engrg. Dept., N.C. State College, State College Sta., Raleigh, N.C.
Vaughn, Everett William (J'40) (BNYZ), Design Engr., Westinghouse Elec. Corp., 2519 Wilkins Ave., Baltimore 3, Md.
Vaule, Sven A. ('23; '37) (CDM), Indus. Engr., Fairbanks, Morse & Co.; *for mail,* 1218 Hinsdale Ave., Beloit, Wis.
Vawter, W. Dale ('33; '43) (NFLEZS), Research Design Engr., Elliott Co., Jeannette; *for mail,* 416 Center Ave., Greensburg, Pa.
Vazsonyi, Andrew (J'42) (ABKH), Engr., Research & Devel., Elliott Co., Jeannette, Pa.
Veal, C. B. ('07; '12) (AE), Coordinating Research Council, 30 Rockefeller Plaza, New York, N.Y.
Veatch, Nathan Thomas, Jr. ('21) (OSH), Partner, Black & Veatch, 4706 Broadway, Kansas City 2, Mo.
Vecchiotti, Frank (J'43) (AEH), Pvt., 1st Bn., Field Arty. Sch., E.M.C., 118 (T369), Ft. Sill, Okla.
Veck, Milton Frank (J'27) (AMBCHK), Project Engr. & Sec. Head, Prod. Methods Devels., Armaments Dept., Cornell Aero. Lab., Cornell Research Foundation, Inc., 4455 Genesee St., Buffalo 5; *for mail,* 985-A Beach Rd., Buffalo 21, N.Y.
Vedder, W. O. ('24; '34) (DBL), Mgr., Dust Control Dept., Pangborn Corp., Pangborn Blvd.; *for mail,* Fountain Head Heights, Hagerstown, Md.
Veenschotan, Vincent V. ('19; '20) (SHCBM), V.P., No. Equip. Co., Delaware Ave. & Grove Dr., Erie, Pa.
Vegosen, Jerome (J'45) (ABM), Test Engr., Wright Aero. Corp., Paterson, N.J.; *for mail,* 1564 Walton Ave., New York 52, N.Y.
Vehslage, Harold E. ('12; '19; '20) (FJS), Supvg. Engr., Westchester Div., N.Y. Hospital, 121 Westchester Ave., White Plains; *for mail,* 32 Jefferson Rd., Scarsdale, N.Y.
Veil, John Wesley (J'42), 126—2nd Ave., Westmont, Johnstown, Pa.
Veinote, Thomas Harvey ('21; '35) (SBF), Ch. Engr., Wm. Filene's Sons Co., 426 Washington St., Boston 1; *for mail,* 147 Cypress St., Newton Centre, Mass.
Vendeleers, Auguste F. (J'36) (FBCS), Constr. Engr., M. H. Treadwell Co., 140 Cedar St., New York; *for mail,* 5415 Netherland Ave., New York 63, N.Y.
Venezia, Louis Alfred (J'43) (BJMNHO), Jr. Product Engr., Arma Corp., 254—36th St., Brooklyn, N.Y.; *for mail,* 26 W. 38th St., Bayonne, N.J.
Venneman, Justin B. (J'42) (BMZJND), Project Engr., Charge of Engrg. Lab., Utility Appliance Corp., 4851 S. Alameda St., Los Angeles 11; *for mail,* 3500 W. Santa Barbara Ave., Los Angeles 43, Calif.
Vera, Francis Leo (J'44) (NZT), Ord. Engr., Army Serv. Forces, Springfield Armory, Federal Sq., Springfield 1; *for mail,* 27 Bowdoin Terrace, Springfield 9, Mass.
Verity, Conrad Edward Howe ('45), Ch. Devel. & Testing Engr., London Power Co., Ltd., Horseferry Rd., Westminster, London, S.W.1, England.
Verkamp, Walter Francis ('18; '25) (DKRS), Pres., Verkamp Corp., Losantiville Ave., Cincinnati 13, Ohio.
Vermillion, Victor Verdon ('44) (MACN), Supvr., Process Planning, Pratt & Whitney Aircraft of Mo., 95th & Holmes, Kansas City; *for mail,* 6316 Chestnut St., Kansas City 5, Mo.
Verner, Hyman J. (J'42) (ONJCKB), Devel. Engr., Syncro Mch. Co., 611 Sayre Ave.; *for mail,* 106 Kearny Ave., Perth Amboy, N.J.
Vernon, Robert S. (J'41) (BEM), Requisition & Design Engr., Gen. Elec. Co., Schenectady; *for mail,* 2756 Broadway, Schenectady 6, N.Y.
Verrall, Godfrey T. ('28; '35) (CZMYJS), Mgr., Hall Tel. Accessories, Ltd., Dudden Hill Lane, London, N.W. 10, England.
Verrier, Arthur Norman (J'46), U.S. Navy; 103 Waldo St., Providence, R.I.
Versaw, William D. (J'43) (ASECBR), 3341 S. 48th St., Lincoln, Neb.
Verschoor, Jack Dahlstrom (J'45) (BNK), Lab. Asst., Mech. Engrg. Lab., Calif. Inst. of Tech., Pasadena 4, Calif.
Vershbow, Arthur Emmanuel (J'43) (BJCZ), 104 Somerset Rd., Brookline, Mass.
Vertrees, Rodney Albert (J'41) (CDNO), Cons. Engr., Rio Oso, Calif.
Vescuso, Anthony (J'43) (JMONBC), Lab. Engr., Consultant, Designer, Induction Htg. Apparatus, Lepel High Frequency Lab. Inc., 39 W. 60th St., New York; *for mail,* 1144 E. 37th St., Brooklyn 10, N.Y.
Vesely, Charles ('41; '45) (CHJMS), Gen. Foreman, Allis-Chalmers Mfg. Co., 1126 S. 70th St., West Allis; *for mail,* 197 N. 123rd St., Wauwatosa 13, Wis.
Vestal, Donald M. (J'38) (CJ), Capt., 2408 Harrison Ave., Ft. Worth, Tex.
Vetlesen, Hans Jorgen ('26; '35) (SFE), Cons. Engr., Gilbert Associates, Inc., 412 Washington St.; *for mail,* 130 N. 5th St., Reading, Pa.
Vevurka, William Edward ('41) (BLP), Mech. Designer, Sinclair Refining Co., East Chicago, Ind.; *for mail,* 1108 W. 103rd St., Chicago 43, Ill.
Viall, Richmond ('40) (BCM), Retired; 141 George St., Providence 6, R.I.
Viberg, Ernest Richard ('18) (ACR), Asst. V.P., Canadian Car & Fdy. Co., Ltd., 631 Craig St. W., Montreal; *for mail,* 5757 Cote St., Antoine Rd., Montreal 28, Que., Can.
Vick, Maurice R. (J'38) (EFSYWC), Buyer, Mech. Equip., Commonwealth Edison Co., Rm. 906, 72 W. Adams St., Chicago; *for mail,* 2139 E. 98th Pl., Chicago 17, Ill.
Vickers, Harry F. ('44) (ACH), Pres., Vickers, Inc., 1400 Oakman Blvd., Detroit 32, Mich.
Vickers, Robert Squire (J'44) (KLC), Design Engr., Stand. Oil Co., Whiting; *for mail,* 816 Fillmore St., Gary, Ind.
Vidosic, Joseph Paul ('32; '42) (BGJC), Assoc. Prof., Mech. Engrg., Ga. Sch. of Tech.; *for mail,* 836 Cherry St., N.W., Atlanta, Ga.
Vierck, Charles John ('42), Asst. Prof., Ohio State Univ.; *for mail,* 1626 Doone Rd., Columbus, Ohio.
Vierck, Robert King ('36; '41) (BJYN), Asst. Prof. Engrg. Mechanics, Sch. of Engrg., 212 Engrg. A, Pa. State College, State College, Pa.

Vierling, Arthur Joseph (J'41) (MJCDLY), Methods & Research Engr., Gorham Mfg. Co., Adelaide Ave., Providence; *for mail*, 223 Clarendon St., Cranston 10, R.I.
Viessman, Warren ('42) (SEFCKL), Mech. & Elec. Engr., Arch. Design Office, Inst. of Inter-Am. Affairs, 12 Calle, Oriente 8, Guatemala, Guatemala, C.A.
Vigne, Albert ('31; '35) (DJM), Pres., Bronze Alloys Co., 6264 St. Louis Ave., St. Louis 20; *for mail*, 260 Blackmer Pl., Webster Groves 19, Mo.
Viles, Gordon Lunceford (J'42), V.P., Charge of Serv. & Maint., Fred M. Viles & Co. Inc., 124 E. Trent Ave., Spokane 8; *for mail*, 2211 S. Grand, Spokane 10, Wash.
Vincent, Arthur S. ('09; '23), Retired; 21 Dartmouth Rd., Mountain Lakes, N.J.
Vincent, Edward T. ('29) (EBFASK), Prof. Mech. Engrg., Univ. of Mich.; *for mail*, 2115 Melrose Ave., Ann Arbor, Mich.
Vincent, Gilbert I. ('13) (CSEHFM), Cons. Engr., 103 Burlingame Rd., Syracuse 3, N.Y.
Vincent, Harry S. ('21) (RKEFSN), Main St., East Harwich 2, Mass.
Vincent, Jesse Gurney ('08; '10) (ABCEFO), V.P., of Engrg., Dir., Packard Motor Car Co., 1580 E. Grand Blvd., Detroit 32, Mich.
Vincze, Andrew ('30; '35) (OEBNKS), Cons. Engr., 58-31—219th St., Bayside, N.Y.
Vinnedge, Earle Walter ('17; '26) (SHKER), Dist. Mgr., Worthington Pump & Mchy. Corp., 8th & Race Sts., Cincinnati 1, Ohio.
Viohl, Christian William, Jr. ('44) (JKLDCG), Engr., H. K. Viohl Co., 2203 W. 18th St., Wilmington 51, Del.
Viohl, Herbert Kurt Wetherell ('23; '29; '35) (SEMNOA), Engr., H. K. Viohl Co., 2203 W. 18th St., Wilmington 51, Del.
Viscardi, John E. (J'40) (NMKBLA), Ch. Engr., Halliwell Inc., 621 Broadway, New York 12; *for mail*, 227 Riverside Dr., New York 25, N.Y.
Visnick, Alexander (J'42) (MJLNCB), Ch. Engr., Trageser Copper Works, Inc., 5000 Grand Ave., Maspeth; *for mail*, 22 Metropolitan Oval, New York 62, N.Y.
Vitale, Joseph Anthony (J'44) (NJMBZY), Mech. Engr., Design, Kimberly-Clark Corp.; *for mail*, 251 E. Doty St., Neenah, Wis.
Vitali, Ercole Julian (J'40) (AH), Engr., Chandler-Evans Corp., Charter Oak Blvd., West Hartford; *for mail*, 21 Ward St., New Haven 11, Conn.
Viti, John (J'40) (BMYJ), Design Draftsman, Gen. Elec. Co., 6901 Elmwood Ave.; *for mail*, 6704 Malvern Ave., Philadelphia 31, Pa.
Vitolo, Robert Adelbert (J'44), Lt., U.S.N.R., Staff Engrg. Officer, Comdr. Tex. Gp., 16th Fleet, c/o Supt. Ships, Orange, Tex.; *for mail*, 3903—48th St., Sunnyside, L.I., N.Y.
Vittucci, Rocco Victor (J'36) (ECKOS), Power Engr., Design, E. I. du Pont de Nemours & Co., Wilmington 98; *for mail*, 707 Coverdale Rd., Wilmington 56, Del.
Vizbara, Mindaugis (J'44) (LNZCKB), Test Engr., Ingersoll-Rand Co., Phillipsburg, N.J.; *for mail*, 855 Linwood St., Brooklyn 8, N.Y.
Vlachos, James Nicholas (J'44) (KFEBCH), Am. Loco. Co., 30 Church St., New York, N.Y.
Vlahakes, John Louis (J'35) (NMDJBC), Methods Engr., Design & Layout, Crucible Steel Co. of Am., S. 4th St., Harrison; *for mail*, 34 Stirling St., Newark 3, N.J.
Voak, Stanley Patterson (J'41) (OZNSR), Project Engr., Huffman-Wolfe Co., 1800 U.B. Bldg., Dayton 2; *for mail*, 115 Nimitz Dr., Dayton 3, Ohio.
Voce, John Denis (J'41) (BAJKFH), Mathematician (Sr.), Armstrong Siddeley Motors Ltd., Parkside, Coventry; *for mail*, 107 Woolmore Rd., Erdington, Birmingham 23, England.
Vockell, Wm. H. (J'16), Pres., Cincinnati Engrg. Tool Co., 1241 Knowlton St., Cincinnati, Ohio.
Vodvarka, Frank John ('40) (MHANCB), Planning Engr., West. Elec. Co., Inc., 2500 Broening Rd., Baltimore; *for mail*, 802 Brunswick Rd., Baltimore 21, Md.
Voelkel, Henry Wyler (J'36) (EHS), Mech. Engr., Design Constr., A. M. Lockett & Co., Ltd., 308 Whitney Bldg.; *for mail*, 2235 Magazine St., New Orleans, La.
Vogdes, Russell Thayer, Jr. (J'43) (JEA), Asst. Welding Engr., Naval Shipyard, Philadelphia; *for mail*, 1847 Widener Pl., Philadelphia 41, Pa.
Vogel, Arthur Philip (J'45) (MNCDJL), Seaman, 1st Class, U.S.N., Bks. 3207, Camp Green Bay, Great Lakes; *for mail*, 1018 Argonne Dr., North Chicago, Ill.
Vogel, Erwin (J'41) (BZAH), Mathematician, Damage Control Project, Stevens Inst. of Tech., Hoboken, N.J.; *for mail*, 47 Perry St., New York 14, N.Y.
Vogel, Henry (J'42) Devel. Engr., Gen. Elec. Co., 100 Woodlawn Ave.; *for mail*, 29 Northumberland Rd., Pittsfield, Mass.
Vogel, Howard Henry (J'38) (EMNDJY), Charge of Engrg., Champion Spark Plug Co., 900 Upton Ave.; *for mail*, 2534 Beaufort Ave., Toledo 6, Ohio.
Vogel, Lawrence Benjamin (J'45) (ACDMNL), Stands. Engr., Westinghouse Elec. Corp., Lester; *for mail*, 813 Mitchell Ave., Morton, Pa.
Vogel, Raphael (J'40) (ODHFZS), Engr., Maint. Dept., Shell Oil Co., Inc., 30 Rockefeller Plaza, New York 20; *for mail*, 60 E. 96th St., New York 28, N.Y.
Vogel, Wm. H. (J'40) (CJK), 268 E. 148th St., Harvey, Ill.
Vogel, William John ('21; '30) (FKS), Exec. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.; *for mail*, 405 N. Fullerton Ave., Montclair, N.J.
Vogelsang, Lewis Oscar ('36) (CFESKY), Lt. Col., Corps of Engrs., U.S.A., Chief, Utilities Branch, Engr., Hdq., 8th Serv. Comd., Dallas 2; *for mail*, 524 James Dr., Garland, Tex.
Vogt, Albert Henry ('39) (HJBCMY), Ch. Engr., United Products Inc., 24 Union Ave., Framingham; *for mail*, 16 Tilton Ave., Ashland, Mass.
Vogt, Axel George ('44) (CDL), Supvr., Prod., Arty. Sec., Springfield Ord. Dist., 95 State St., Springfield; *for mail*, Apt. 9, 161 Broad St., Lynn, Mass.
Vogt, Charles Clifford ('46) (BCD), Chief, Mech. Sub-Div., Naval Ord. Lab., Naval Shipyard, Washington 25, D.C.; *for mail*, 6808 Denton Rd., Bethesda, Md.
Vogt, Clarence W. ('15; '23) (YL), Devel. Engr., Route 1, Norwalk, Conn.
Vogt, Herman William (J'42) (MNCL), Time & Motion Study, Eng. Dept., Wilcox-Rich Corp., Div. of Eaton Mfg. Co., 1000 Rust St.; *for mail*, 1008 S. Webster St., Saginaw 55, Mich.
Vogt, Paul Richard (J'40) (BEASHY), Project Engr., Chrysler Corp., Box 1919, Detroit; *for mail*, 14130 E. Jefferson Ave., Detroit 30, Mich.
Vogt, Richard Stanley (J'43), 41-26—73rd St., Jackson Heights, N.Y.
Vogtsberger, Arthur (J'44) (SEFKC), Engr., Power Div., Sun Oil Co., Woodville St.; *for mail*, 524 Dover Pl., Toledo 5, Ohio.
Voigt, Frederick Albert (J'41) (BJAHC), Lt. Col., Inf., War Dept.; *for mail*, 4328 N. Henderson Rd., Arlington, Va.
Voigt, Louis Edwin ('40) (LSHKEF), Ch. Draftsman, Atlantic Refining Co., 260 S. Broad St., Philadelphia 1; *for mail*, 6343 Ross St., Philadelphia 44, Pa.
Voisinet, Walter Eugene ('43) (KFOS), Mfrs. Rep., Air Conditioning Equip., 250 Delaware Ave., Buffalo 2, N.Y.
Vojir, Jerry E. (J'43) (KBSH), Jr. Mech. Engr., N.Y. Naval Shipyard, Brooklyn; *for mail*, 29 Sweezy Ave., Highland Falls, N.Y.
Vokac, Charles W. ('34; '42) (HJSCBN), Ch. Engr., Hydro-Arc Sec., Whiting Corp., Harvey; *for mail*, 4818 W. 23rd Pl., Cicero 50, Ill.
Vokoun, Otto Henry ('28; '34; '37) (CDLMOY), Gen. Foreman (Assisting M.M.), Swift & Co., Master Mechanics Office, Union Stock Yards, Chicago 9, Ill.
Volck, A. George ('41), Calif. Bank Bldg., 9441 Wilshire Blvd., Beverly Hills, Calif.
Volckhausen, Walter J. (J'28) (BJM), Mch. Designer, West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail*, 1325 Dickerson Rd., West Englewood, N.J.
Volks, Herbert ('42) (CMNL), Pres., Pioneer Indus. Corp.,, 70 Wall St., New York 5; *for mail*, Hotel Tudor, New York, N.Y.
Vollbrecht, Justus Thomas ('38) (SLCOT), Pres., Energy Control Co., Inc., 5 Beekman St., New York 7, N.Y.
Voller, James P. ('39) Box 643, Route 11, Fresno, Calif.
Vollmer, Paul L. ('21; '35), Mech. Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail*, 3785 Greensburg Pike, Wilkinsburg, Pa.
Vollmer, William E. (J'41) (ACJ), Prod. Engr., Candler-Hill Corp., 405 Midland; *for mail*, 948 Chicago Blvd., Detroit, Mich.
Volpe, Vincent F. (J'39) (E), Naval Ord. Lab., Naval Shipyard, Washington, D.C.; *for mail*, 1437 N. Kostner Ave., Chicago, Ill.
Von Arb, Edward Joseph (J'44) (MJCD), Process Engr., Servel, Inc., 119 N. Morton St., Evansville 20; *for mail*, 837 Bellemeade, Evansville 13, Ind.
von Bremen, Daniel W., Jr. (J'40) (ACHMOY), Engr., David E. Kennedy, Inc., 58—2nd Ave., Brooklyn; *for mail*, 3-01—150th St., Whitestone, L.I., N.Y.
Vondersmith, Earl L. ('46) (DHP), Engrg. Draftsman, Atlantic Refining Co., 260 S. Broad St., Philadelphia; *for mail*, 1032 Rees Ave., Bryn Mawr, Pa.
von Fabrice, Roman ('22; F'41) (CDEFOS), Designing Engr., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
von Gontard, Adalbert ('29; '35) (BCKMS), Lt., U.S.N., Charge Diesel Schs.; V.P. & Ch. Engr., Anheuser-Busch, Inc., 721 Pestalozzi St., St. Louis 18, Mo.
Von Herrmann, Charles F., Jr. ('33; '44) (NJCAYZ), Owner, Turkey Creek Mfg. Co., 1927—1st Ave. S., Birmingham 3, Ala.
Von Kármán, Theodor ('31), *A.S.M.E. Medallist*, '41; Dir., Guggenheim Aero. Lab., Calif. Inst. of Tech., Pasadena, Calif.
von Kaske, Karl Homer ('44), Res. Engr., Research Labs., Pittsburgh Plate Glass Co., Creighton; *for mail*, 1030½ Carlisle St., Tarentum, Pa.
von Maltitz, Helmut Karl ('45) (WHYNMZ), Secy., Engr., Plycor Co., 154 E. Erie St., Chicago 11, Ill.
Von Ohlsen, Louis H. ('21; '25; '35) (BR), Asst. Engr., Safety Car Htg. & Ltg. Co., Box 904, New Haven; *for mail*, 42 Vista Terrace, New Haven 15, Conn.
Von Pein, Edward J. ('13), 335 Watervliet Ave., Dayton 10, Ohio.
von Phul, William ('07; F'41), Retired; 64 Grove Ave., Larchmont, N.Y.
von Rehberg, Hugo L. ('39) (SZOK), Charge of Engrg. Dept., Braman, Dow & Co., 239 Causeway St., Boston 14, Mass.
Von Rotz, Robert ('45) (HJMABC), Exec. Engr., N.Y. Air Brake Co., 420 Lexington Ave., New York 17; *for mail*, 43 Payne Whitney Lane, Manhasset, N.Y.
von Schlegell, Frederick ('15), Retired; 888 Arroyo Blvd., Pasadena, Calif.
Von Till, Louis A. ('23; '31; '35) (CJLMNO), Mech. Project Engr., Closure Div., Armstrong Cork Co., Liberty & Mary Sts.; *for mail*, Route 2, Millersville Pike, Lancaster, Pa.
Von Voigtlander, Oscar ('27) (BHEFSK), Cons. Engr., Route 1, Eugene, Ore.
von Wehrden, Carl F. (J'42) (ZNMYDC), Engr., Sound & Projection, Motion Picture Serv., U.S.A., 3327-A Locust St., St. Louis 3; *for mail*, 5328-C Gladstone Pl., St. Louis 21, Mo.
von Wehrden, F. Walter ('33; '35), Supt., Broderick & Bascom Rope Co., 4233 N. Union Blvd., St. Louis; *for mail*, 8412 Glen Echo Dr., Normandy, St. Louis Co., Mo.
Voorhees, Guy Ashley ('44) (FKOCH), Cons. Engr., Box 73, Route 1, Monrovia, Ind.
Voorhees, John Redding ('42) (OCSWZ), Mech. Engr., Turner Constr. Co., 420 Lexington Ave., New York 17, N.Y.; *for mail*, 26 Woodcrest Ave., Millburn, N.J.
Voorhees, Stanley Van (J'38) (ABH), Hyds. Div., Engrg. Dept., Douglas Aircraft Co., Inc., Santa Monica; *for mail*, 2730 Hollyridge Dr., Hollywood, Calif.
Voorhees, Stephen F. ('23), Sr. Partner, Arch., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York, N.Y.
Voorheis, James Tuttle (J'42) (KHEFA), Engr., Coen Co., 40 Boardman Pl., San Francisco 3; *for mail*, 50 Inverleith Terrace, Piedmont 11, Calif.
Voorheis, Temple Stadtwan (J'37) (SKM), Mech. Engr., Coen Co., 40 Boardman Pl.; *for mail*, 989 Lombard St., San Francisco 3, Calif.
Voos, Frederick William, Jr. (J'45) (CEK), 412 Greenwood Pl., Syracuse 10, N.Y.
Vopat, William Arthur ('31; '41) (SFEK), Asst. Prof., Cooper Union, Cooper Sq., New York 3; *for mail*, 150 Park Blvd., Malverne, L.I., N.Y.
Vorhees, Roy W., Jr. (J'38) (CMAE), Spec. Assignment to Factory Mgr., Plymouth Motor Co., Div. of Chrysler Corp., Detroit; *for mail*, 13449 Camden St., Detroit 13, Mich.
Vose, Fred Hale ('06; '10) (SFK), Prof. Mech. Engrg., Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6, Ohio.
Vose, Richard Shepard ('46) (ABE), Research Engr., Peco Div., Bendix Aviation Corp., Philadelphia 33; *for mail*, 225 Princeton Ave., Swarthmore, Pa.
Vose, Robert Weston ('32; '43) (BT), Dir. of Research, Chicopee Mfg. Corp., Chicopee Falls, Mass.
Voss, Johann H. H. ('29) (BEJS), Pres., J. H. H. Voss Co., 785 E. 144th St., New York 54, N.Y.
Voss, Wilbur Fred (J'35) (BHJNOM), Design Engr., Crucible Steel Co. of Am., 4th St., Harrison; *for mail*, 287 Franklin Ave., Cliffside Park 9, N.J.
Voss, William Jesse (J'44) (ACMWON), Mech. Engr., Axial Compressor Research, Aircraft Eng. Research Lab., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland 11; *for mail*, 9813 Cumberland Ave., Cleveland 4, Ohio.
Vossbrinck, William John Henry ('43) (CEFASL), Asst. to Mgr., Compressor Div., Chicago Pneumatic Tool Co., 6 E. 44th St., New York 17, N.Y.
Voyce, Lee Charles ('43) (CJMLDS), Mgr., East. Div., Midwest Piping & Supply Co. Inc., 30 Church St., New York 7, N.Y.
Voyles, Richard M. (J'40) (EBK), Head, Vibrator Engrg. Dept., Electronic Labs., Inc., 122 W. New York St., Indianapolis 4; *for mail*, 3444 Washington Blvd., Indianapolis 5, Ind.
Voysey, Alfred Everett ('31; '45) (CMJL), Maj., Ord. Dept., Watertown Arsenal, Arsenal St., Watertown 72; *for mail*, 414 Waltham St., West Newton 65, Mass.
Vragel, Kurt Edward (J'44), Ensign, U.S.N.R., U.S.S. *Banquet*, DE-739, F.P.O., San Francisco, Calif.; *for mail*, 3619 Lakewood Ave., Chicago, Ill.
Vreeland, Milton A. ('35) (SOFC), Project Engr., Burns & Roe, Inc., 233 Broadway, New York 7; *for mail*, King's Highway, Middletown, N.Y.
Vroman, Erwin C. ('29), Mech. Engr., N.Y. Air Brake Co.; *for mail*, 208 S. Indiana Ave., Watertown, N.Y.
Vroom, Robert C. ('29; '35), Ch. Engr., Peabody Engrg. Corp., 580—5th Ave., New York, N.Y.; *for mail*, 60 Edgemont Rd., Montclair, N.J.
Vuilleumier, Albert ('16; '35) (CDHM), Ball Plant Supt., New Departure Div., Gen. Motors Corp., N. Main St.; *for mail*, Box 1077, Bristol, Conn.
Vukan, Frank S. (J'42), 1st Lt., Army Air Forces, 43rd Sta. Complement Sqd., A.P.O. 4740, c/o Postmaster, New York, N.Y.
Vye, George P. (A'43) (OCS), Duff & Phelps, 208 S. La Salle St., Chicago 4; *for mail*, 1259 Old Mill Rd., Deerfield, Ill.

W

Waage, John J. ('29; '35) (CM), Pres., Stimpson Computing Scale Co., Inc., 837 S. Logan & Breckinridge Sts., Louisville, Ky.
Wachs, Charles Ludwig ('18) (JSC), Pres., E. H. Wachs Co., 1525 Dayton St., Chicago 22, Ill.
Wachs, Theodore ('11; '18; '19) (CMLOKD), Gen. Plant Engr., RCA Victor Div., Radio Corp. of Am., Front & Cooper Sts., Camden, N.J.
Wachter, Lester W. ('44) (HJL), Ch. Engr., Anthony Co., Inc.; *for mail*, 309 W. Grant St., Streator, Ill.
Wachunas, Joseph F. (J'44) (FKS), 1505 Metropolitan Ave., New York 62, N.Y.
Wacker, Emil John, Jr. (J'38) (ECS), Mech. Engr., Magnolia Pipe Line Co., Box 900, Dallas 1, Tex.
Waddell, Charles Lorne ('44) (KHSC), Ch. Engr., Worthington Pump & Mchy. Corp., Worthington Ave., Harrison; *for mail*, 81 Fair Haven Rd., Fair Haven, N.J.
Waddell, George Fulton (A'98) (EABJ), Retired; 1975—18th Ave., San Francisco 16, Calif.
Waddell, Joseph L. (J'42) (YSKJHE), B. F. Goodrich & Co., 500 S. Main St., Akron; *for mail*, 955 Delia Ave., Akron 2, Ohio.
Waddell, William Colvin ('43) (MBLC), Engr., Mgr., Plant 2, John W. Jones Co., 425 S. State St., Indianapolis 1; *for mail*, 5440 College Ave., Indianapolis 5, Ind.
Waddington, Lester Edwin ('37; '45) (CGWYAJ), Design Engr., C. G. Conn, Ltd., 1101 E. Beardsley Ave., Elkhart, Ind.
Wade, Edward A. ('46) (FHS), Yerkes Works, E. I. du Pont de Nemours & Co., Buffalo, N.Y.
Wade, Edward Annesley ('34; '46) (YML), Product Engr., Am. Hard Rubber Co., Park Pl.; *for mail*, Box 14, Catherine St., Butler, N.J.
Wade, Franklin S. ('18), Pres., Gen. Mgr., So. Counties Gas Co., Rm. 609, 810 S. Flower St., Los Angeles, Calif.
Wade, Garth S. (J'41) (JCMBY), Plant Engrg. Dept., Canadian Gen. Elec. Co. Ltd.; *for mail*, 269 John St., Peterborough, Ont., Can.
Wade, Irving L. ('43) (SFC), Supt., Elec. Prod., Pub. Serv. Co. of No. Ill., 72 W. Adams St., Chicago; *for mail*, 4127 Wolf Rd., Western Springs, Ill.
Wade, Walter Anderson ('30; '34) (BJSCMH), Inspr. of Mchy., Govt. of Union of S. Africa, Box 164, Germiston; *for mail*, 65 Mars St., Kensington, Johannesburg, S. Africa.
Wade, William Morgan ('31; '35) (SFC), Ch. Engr., Westfield State Sanatorium; *for mail*, Box 186, 20 Pearl St., Westfield, Mass.
Wadge, Gordon Frederick (J'43) (JRS), Ensign, U.S.N.R., c/o Office of Indus. Mgr., Compass Sec., 8th Naval District, 1 Canal St., New Orleans, La.
Wadiaeff, Michael (J'46) (CJY), 39-88—49th St., Long Island City 4, N.Y.
Wadleck, Joseph Paul ('44) (BDHJNO), Spec. Engr., Charge of Design of Spec. & New Equip., Aetna Stand. Engrg. Co., W. Federal Ave., Youngstown, Ohio.
Wadleigh, George Robinson ('07) (LSDOEK), Cons. Engr., 7 Dey St., New York; *for mail*, 70 Flower Ave., Hastings-on-Hudson, N.Y.
Wadman, Harold A. ('25; '35) (BKNHJF), Ch. Physicist, Hartford-Empire Co., 333 Homestead Ave., Hartford; *for mail*, 45 Ten Acre Lane, West Hartford 7, Conn.
Wadsworth, John Frederic ('15; '17; '22) (OSY), 818 Commerce Bldg.; *for mail*, 254 W. 8th St., Erie, Pa.
Wager, Robert C. (J'36) (CDM), 2663 W. North Ave., Baltimore 16, Md.
Wagner, Donald Eugene (J'38) (CM), Ch. Cost Clerk, Cost Dept., Mchy. Div., Crown Cork & Seal Co., Inc., 1508 Guilford Ave., Baltimore 2, Md.
Wagner, Edmond M. ('29; '41) (MHCBLJ), *Junior Award*, '32; Cons. Engr., 1165 Lorain Rd., San Marino 9, Calif.
Wagner, Elijah Rudolph (J'41) (CJA), Pvt., 1st Class, U.S.A., Co. F, 134th Inf., A.P.O. 35, c/o Postmaster, New York, N.Y.
Wagner, Frank Caspar (M'45) (JMS), Ch. Engr., Power Plant, Cent. Power & Light Co.; *for mail*, 3105 Lawnview Dr., Corpus Christi, Tex.
Wagner, Frederick David (J'41) (CLD), Mech. Engr., Swift & Co., Union Stock Yards, Chicago 9; *for mail*, 6545 S. Union Ave., Chicago 21, Ill.
Wagner, Harlan Cleveland, Jr. (J'41) (RAH), Technician, 5th Grade, Signal Corps, 9463rd Tech. Serv. Unit, Radio Propagation Unit, Holabird Signal Depot, Baltimore 19, Md.; *for mail*, 1040 W. Front St., Plainfield, N.J.
Wagner, Harry Dallas (J'45) (SHBDZN), Jr. Engr., Mech. Engrg. Div., Philadelphia Elec. Co., 900 Sansom St., Philadelphia; *for mail*, 7530 Mayland St., Philadelphia 38, Pa.
Wagner, Harvey Arthur ('28; '38) (SEBJFK), System Engr., Detroit Edison Co., 2000—2nd Ave., Detroit 26; *for mail*, 12900 E. Outer Dr., Detroit 24, Mich.
Wagner, Howell Charles (J'42) (ZME), Asst. Research Project Engr., Sperry Gyroscope Co., Inc., Great Neck; *for mail*, 230 S. Linden St., West Hempstead, L.I., N.Y.
Wagner, Laurence E. ('28) (FJCEKT), Ch. Indus. Engr., Providence Gas Co., 100 Weybosset St., Providence 1; *for mail*, 20 Taft St., West Warwick, R.I.
Wagner, Paul William (J'45) (BCL), Box 479, Charlotte Amalie, St. Thomas, Virgin Islands.
Wagner, Richard Edwin (J'43) (CMKSAN), Asst. Plant Engr., Southwind Div., Stewart-Warner Corp., 1514 Drover St., Indianapolis 7, Ind.
Wagner, Robert Jacob (J'45) (NBJHAY), Mech. Design Engr., Westinghouse Elec. Corp., 2519 Wilkens Ave.; *for mail*, 1021 Wilmington Ave., Baltimore 23, Md.
Wagner, Stanley Thomas (J'45), U.S. Navy, 217 Rector St., Philadelphia 28, Pa.
Wagner, Stephen Stanley (J'39) (MNLBJC), Instr. in Engrg., Univ. of Buffalo, Buffalo 14; *for mail*, 114 N. Parade Ave., Buffalo 11, N.Y.
Wagner, Warren Orval (J'39) (BCH), Asst. Prof. Hyds., Project Supvr., Navy Engrg. & Research, Calif. Inst. of Tech.; *for mail*, Caltech, Pasadena 4, Calif.
Wagoner, Philip D. ('14), Chmn., Underwood Corp., 1 Park Ave., New York 16, N.Y.
Wahl, Arthur M. ('28; '30; '35) (BNOJ), *Junior Award*, '29; Research Engr., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Wahl, Henry R. ('15), Retired; 1535 N. Las Palmas St., Los Angeles 28, Calif.
Wahl, James Howard ('44) (CEJS), Asst. Sales Mgr., Chapman Valve Mfg. Co., 203 Hampshire St., Indian Orchard, Mass.
Wahlgren, R. Donald (J'43) (CJM), Ensign, Civil Engr. Corps, U.S.N.R., ACEPD, Navy 3149, F.P.O., San Francisco, Calif.
Wahrenburg, Lester E. F. ('26; '37) (SOFDEC), V.P., Peter F. Loftus Corp., 632 Oliver Bldg., Pittsburgh, Pa.
Waibler, Paul John (J'45) (ABEKS), Instr. in Mech. Engrg., Mason Lab., Yale Univ., 400 Temple St., New Haven 11, Conn.
Waidelich, James Richard ('44) (BCL), Mgr., Spec. Engrg. Dept., Bristol Co., Waterbury; *for mail*, Chestnut St., Naugatuck, Conn.
Wainwright, Arthur M. ('41) (FKS), Gen. Elec. Co., 570 Lexington Ave., New York 22, N.Y.
Wait, Wm. B. (J'14) (CLS), Plant Engr., Am. Sugar Refining Co., 132 N. Peters St.; *for mail*, 2011 Palmer Ave., New Orleans, La.
Wait, William Bell ('42) (R), Pres., Valve Pilot Corp., 230 Park Ave., New York 17, N.Y.
Waite, Fred Lee (J'46), Triceps Design & Devel. Co., 115 Broadway, New York; *for mail*, 51-01—39th Ave., Long Island City 4, N.Y.
Waite, Howard Winfred ('42) (BHANM), Cons. Engr., 3727 Arbolada Rd., Los Angeles 27, Calif.
Waite, Robert Henderson ('43) (RJ), Sales Agt., Am. Car & Fdy. Co., 80 E. Jackson Blvd., Chicago 4, Ill.
Waite, William Henry ('22) (DEO), Sales Mgr., Browning Crane & Shovel Co., 16226 Waterloo Rd., Cleveland 10; *for mail*, 2475 Lee Blvd., Cleveland 18, Ohio.
Waitkus, Joseph ('31; '42) (SFKNZM), Mgr., Erection & Serv. Dept., Air Preheater Corp., Main St., Wellsville, N.Y.
Wakefield, Frederick Irving (J'45) (SEFHKA), Portsmouth Ave., Stratham, N.H.
Walch, Richard Hyrum (J'42), Engr., Caterpillar Tractor Co., Peoria; *for mail*, 259 Randolph Ave., Peoria 5, Ill.
Walcher, Edwin Alfred (J'42) (CM), Asst. Oper. V.P., Am. Steel Fdys., 410 N. Michigan Ave., Chicago, Ill.
Walcott, H. G., Jr. (J'38) (BGP), Chas. L. Henry & Associates, 3115 Westminister, Dallas, Tex.
Walden, Robert F. (J'40) (MS), Mch. Designer & Draftsman, Mchy. Div., Republic Supply Co. of Calif., 2122 E. 7th St., Los Angeles; *for mail*, Box 581, Avenal, Calif.
Walden, Robert Rice ('21; '30; '35) (SFKLDN), Sr. Mech. Engr., Pa. Salt Mfg. Co., 1000 Widener Bldg., Philadelphia 6; *for mail*, 420 Thayer St., Ridley Park, Pa.
Walden, William R. (J'38) (BLH), 2nd Lt., Ord. Dept., Proof Officer, Research Center, Aberdeen Proving Ground, Md.
Waldrep, James E. (J'37) (FMS), Supply Sgt., Co. B, 5th Inf., U.S.A., A.P.O. 360, c/o Postmaster, New York, N.Y.; *for mail*, 9 Hampton Ave., Greenville, S.C.
Waldron, Everett H. ('20) (TCF), Pur. & Mech. Engr., Mt. Hope Finishing Co., North Dighton; *for mail*, 228 Weir St., Taunton, Mass.
Waldron, J. Laurence ('15; '35) (SCDFEK), 652-A Lafayette Ave., Brooklyn 16, N.Y.
Waldron, Jerome Hubeli (J'28) (TCDN), Box 74, R.R. 2, New Brunswick, N.J.
Walendy, Marlin Frederick (J'40) (CDM), Capt., Ord. Dept., Army Serv. Force Depot, Columbus 15, Ohio.
Wales, C. Clarke ('30; '35) (CFJ), Ch. Project Engr., Natl. Tube Co.; *for mail*, 4247 Lake Rd., Sheffield Lake, Lorain, Ohio.
Wales, Edgar William (J'46) (CES), Ensign, U.S.N.R.; 2903 Glenview Ave., Philadelphia 24, Pa.
Wales, Richard Noyes (J'41) (A), Pvt., Army Air Forces, Sqd. L, 3020th Base Units, La Junta, Colo.
Wales, Robert ('36) (EFJS), Tech. Dir., Compania Cubana de Fianzas, Amargura 203, Havana, Cuba.
Wales, Royal Linfield ('23) (EFKS), Dean Emeritus, Engrg., R.I. State College; *for mail*, Campus Ave., Kingston, R.I.
Walin, Don R. (J'42), Engr., Asst. A, Structures Dept., San Diego Div., Consltd. Vultee Aircraft Corp., San Diego; *for mail*, 12 Torrey Pines Homes, La Jolla, Calif.
Walkama, Toivo Edward ('27; '34; '35) (FKSZ), Research Engr., Airkem, Inc., 206 E. 50th St., New York 22, N.Y.; *for mail*, 351 Armstrong Ave., Jersey City 5, N.J.
Walkem, George Alexander ('29) (BFO), Pres., Managing Dir., Vancouver Mchy. Depot Ltd., 6th Ave., West Vancouver, B.C., Can.
Walker, Alexander George ('36; '45) (BHM), Ch. Engr., Kent Mch. Co.; *for mail*, 2767 Northland Ave., Cuyahoga Falls, Ohio.
Walker, Alexander Marriott (J'37) (KLMBCO), Ch. Engr., C. E. Rogers Co., 8731 Witt St., Detroit 9, Mich.
Walker, Alvin Robert (J'38) (BKHOEF), Mar. Engr., Naval Shipyard, Mare Island; *for mail*, 1717 Oxford St., Berkeley, Calif.
Walker, Carroll Lambert, Jr. (J'43) (BACZGH), Engrg. Research Dept., Univ. of Mich.; *for mail*, 3309 Edgewood, Route 7, Ann Arbor, Mich.
Walker, Chapman J. ('40) (SEKNFB), Devel. Engr., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, Loudonville, N.Y.
Walker, Charles Eugene (J'43) (KJCR), Pierce, Idaho.
Walker, David Stewart, Jr. (J'46), 1105 Park St., Route 2, Charlottesville, Va.
Walker, Donald S. ('36) (CFS), V.P., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Walker, Emery LaFayette ('10) (BDESW), V.P., Kieckhefer Container Co., 1715 W. Canal St., Milwaukee 3; *for mail*, 7406 Hillcrest Dr., Wauwatosa 13, Wis.
Walker, Frederick Wiley ('98; '09) (RSEAHC), V.P., Charge of Bond Investments, Northwest. Mutual Life Ins. Co., 720 E. Wisconsin Ave., Milwaukee 2, Wis.
Walker, Gilbert S. ('04) (OBNGH), Cons. Engr., 28 Jonquil Pl., Pittsburgh 16, Pa.
Walker, Glen Hart ('45) (CDMORS), Asst. Ch. Engr., Plant Facilities, Boeing Aircraft Co., Seattle 14; *for mail*, 1623 Calhoun St., Seattle 2, Wash.
Walker, Hamilton Torrey, Jr. (J'42) (AKS), 959 Brunswick Rd., Cleveland Heights 12, Ohio.
Walker, Harold Francis (J'28) (LDJMCB), Mech. Engr., West. Elec. Co., Inc., Chicago; *for mail*, 1157 S. Taylor Ave., Oak Park, Ill.
Walker, Harold Louis ('30; '35) (NBCDLM), Owner, Wakefield Mch. Co., 4510 White Plains Rd., Bronx 66; *for mail*, 716 E. 22nd St., Brooklyn 10, N.Y.
Walker, Henry Sherriff ('30; '35) (ABCSY), Dir. of Research, Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Walker, James Herbert ('29) (C), V.P., Detroit Edison Co., 2000—2nd Ave., Detroit 26, Mich.
Walker, James Joseph (J'45) (HKO), Fire Protection Engr., Carbide & Carbon Chem. Corp., South Charleston; *for mail*, 1586 Virginia St. E., Charleston, W.Va.
Walker, John Edgar ('26; '33; '35) (CERFSK), Supt. Trans., Hawaiian Commercial & Sugar Co.; *for mail*, Box 83, Puunene, Maui, T.H.
Walker, John Howe (J'43) (AJ), 3569 Broadway, New York 31, N.Y.
Walker, John Joseph (J'41) (ACJ), Box 67, Crystal City, Tex.
Walker, Nelson Crosby (A'44) (MLWYCZ), Ch. Engr., Noma Elec. Co., 55 W. 13th St., New York, N.Y.
Walker, Norbert James ('36) (SEHFKN), Asst. to Mech. Engr., Philadelphia Elec. Co., 900 Sansom St., Philadelphia 5, Pa.
Walker, Philip John (J'46), 75 Wilcox Ave., Meriden, Conn.
Walker, Ralph S. ('39), Mech. Engr., Hecker Products Corp., 1437 W. Morris St.; *for mail*, 759 Berkley Rd., Indianapolis, Ind.
Walker, Richard Kenneth (J'42) (NZBJLM), Project Engr., Research Labs., Sperry Gyroscope Co., Garden City; *for mail*, 357 Jackson St., Hempstead, L.I., N.Y.
Walker, Robert Bell (J'39) (AMR), 239 Kenyon Ave., Swarthmore, Pa.
Walker, Robert Bernard (J'43) (NS), Sr. Draftsman, Aluminum Co. of Am., Alcoa; *for mail*, 2519 Yarmouth St., Knoxville, Tenn.
Walker, Rolland Clinton (J'41) (EFCSMO), Lt., Ord. Dept., U.S.A., Depot & Property Disposal Officer, Rock Island Arsenal, Rock Island, Ill.
Walker, Roy Pierce (J'34) (CNL), Sr. Mech. Engr., Plant Engr.'s Dept., Johns-Manville Products Corp.; *for mail*, 705 W. Ocean Ave., Lompoc, Calif.
Walker, Sidney G. ('07) (C), 404 Portlock Rd., Honolulu 49, T.H.
Walker, Terrance James (J'43) (MJKYF), Tech. Foreman, Styrene Div., Dow Chem. Co., Box B; *for mail*, 218 Brazos Court, Freeport, Tex.
Walker, Terrell Hamilton ('44), Mgr., Serv. & Erection Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York, N.Y.; *for mail*, 816 Valley Rd., Merion, Pa.

Walker, Thorp Byrd (J'43) (NHAZKJ), Devel. Engr., Aerojet Engrg. Co., Irwindale Rd., Azusa; *for mail,* 376 S. Marengo Ave., Pasadena 5, Calif.
Walker, Vincent John (J'40) (JNLKYZ), Design Engr., Phillips Petroleum Co.; *for mail,* Box 728, Bartlesville, Okla.
Walker, William Lloyd (J'45) (LKHZSB), Design Engr., So. Calif. Gas Co., Box 3249, Terminal Annex, Sta. 140, Los Angeles 54, Calif.
Walker, Wythe Finch (J'45) (AK), Engr., Douglas Aircraft Co., Inc., Long Beach 4; *for mail,* 3637—6th Ave., Los Angeles 16, Calif.
Wall, Albert George ('42) (DCMN), Ch. Engr., Gruendler Crusher & Pulverizer Co., 2915 N. Market St., St. Louis 6; *for mail,* 4412 N. Mathews, St. Louis 20, Mo.
Wall, Thomas Oliver (J'41) (CAH), Sr. Rep., Hawaiian Area, Sperry Gyroscope Co., Inc., 502 Strangenwald Bldg., 119 Merchant St., Honolulu 1, T.H.
Wall, William Clark ('20; '35) (O), Retired; Box 40, R.F.D. 1, Old Lyme, Conn.
Wallace, Archibald Lyle ('30) (YMLN), Pres., Works Mgr., Metal Hose & Tubing Co., Dover, N.J.
Wallace, Arthur Ronald (J'45) (KSEFZC), Design Engr., Advance Htg. & Air Conditioning Corp., 117 N. Desplaines St., Chicago 6; *for mail,* 4407 N. Sacramento Blvd., Chicago 25, Ill.
Wallace, Charles Huntley ('29; '35) (LTKJ), Sales Mgr., Tolhurst Centrifugal Div., Am. Mch. & Metals, Inc., East Moline, Ill.
Wallace, Edward E. ('39) (ABC), Designer, Warner & Swasey Co., 55th St. & Carnegie Ave., Cleveland; *for mail,* 3380 Berkeley Rd., Cleveland Heights, Ohio.
Wallace, Edward George (J'45), Pvt., U.S.A., C 227 Bn., 69th Regt., Camp Blanding, Fla.
Wallace, Francis Xavier, Jr. (J'43) (SE), Test Engr., Gen. Elec. Co., Schenectady 5; *for mail,* 55 Furman St., Schenectady, N.Y.
Wallace, G. W. ('44) (FJP), C. W. & F. Coal Co., Benton, Ill.
Wallace, George A. (J'38) (NRSB), Stress Engr., Edw. G. Budd Mfg. Co., Red Lion & Veree Rds., Bustleton, Philadelphia; *for mail,* 43 W. Johnson St., Philadelphia 44, Pa.
Wallace, George MacKenzie (J'40) (AES), Mgr., Denver Office, Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio; *for mail,* Box 365, Denver, Colo.
Wallace, Henry Bremen, Jr. (J'35) (SEYLKF), Asst. New England Dist. Mgr., Foster Wheeler Corp., 80 Federal St., Boston 10, Mass.
Wallace, James Albert (J'42) (EFS), 409 W. 7th St., Boone, Iowa.
Wallace, John H. G. (J'33) (AEBZKH), Mgr., Pat. Dept., Airesearch Mfg. Co., 9851 Sepulveda Blvd., Los Angeles 45; *for mail,* 366 Skyewiay Rd., Los Angeles 24, Calif.
Wallace, Lawrence Wilkerson ('12) (CMOR), *Vice-President,* '37-'39; V.P., Trundle Engrg. Co., 1501 Euclid Ave., Cleveland; *for mail,* 15615 S. Moreland Blvd., Shaker Heights 20, Ohio.
Wallace, Robert Adams ('34; '35) (CDJMOZ), Cons. Engr., Clark Bldg., Pittsburgh 22, Pa.
Wallace, Ross S. ('04), Chmn. of Bd., Cent. Ill. Light Co., 316 S. Jefferson Ave., Peoria 2, Ill.
Wallace, Wm. Anderson (J'35), 74 Glendale Ave., Toronto, Ont., Can.
Wallace, William Marlette (J'44) (LCEGWY), Asst. Research Engr., Carbide & Carbon Chemicals Corp., Box P; *for mail,* 213 Vermont Ave., Oak Ridge, Tenn.
Wallace, William V. (J'38), Box 381, San Anselmo, Marin Co., Calif.
Wallach, Bernard H. ('44) (ABHE), Tech. Specialist, Aerodynamics, Edo Aircraft Corp., 136-48 Roosevelt Ave., Flushing; *for mail,* 2911 Barnes Ave., New York 67, N.Y.
Wallene, Frank O. ('36) (OS), Engrg. Consultant, Wallene Engrg. Co., 3304 W. 162nd St., Cleveland 11, Ohio.
Waller, C. Richard ('13) (BHKMNS), V.P., Ch. Engr., De Laval Steam Turbine Co., Trenton 2, N.J.
Waller, Paul Frederick William, Jr. ('44) (CFS), Dist. Engr., Combustion Engrg. Co., 2330 Bankers Bldg., Chicago, Ill.
Wallin, Joseph Waldemar ('31; '33; '35) (BFKS), Ch. Engr., E. Keeler Co., 238 West St., Williamsport; *for mail,* 705 Lincoln Ave., Williamsport 31, Pa.
Wallin, Maynard Burdette (J'44) (ACHNYZ), Tech. Serv. Engr., Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16; *for mail,* 509 Crosby St., Akron 3, Ohio.
Wallis, Howard L. (J'37) (CSW), Lt. (j.g.), U.S.N.R., U.S. Naval Air Sta., c/o Armament Test., Patuxent River, Md.
Wallis, John S. ('42), Petro Chem. Devel. Co., 120 E. 41st St.; *for mail,* 130 E. 40th St., New York, N.Y.
Walmsley, George (J'22) (SFEKLH), Engr., Gen. Engrg. Div., Stand. Oil Devel. Co., Box 37, Elizabeth B, N.J.
Walmsley, Samuel Edmund ('44), Ch. Engr., Bowater-Lloyd Gp., Bowater's Paper Mills, Ltd., Northfleet; *for mail,* Loughrigg, 246 Singlewell Rd., Gravesend, Kent, England.
Walser, Robert Danforth (J'41) (MCJEYK), Indus. Engr., "N" Plant, Aluminum Co. of Am., 2210 Harvard Ave., Cleveland 5, Ohio.
Walsh, Benjamin Howard (J'42), Lt., Army Air Forces, Unit A, 409th Bomb Gp., Clovis, New Mex.
Walsh, Carl Z. (J'40), c/o Colombian Petroleum Co., Apt. 100, Cucuta, Colombia, S.A.
Walsh, Edwin Parsons (J'40) (BAKSH), Design Engr., Charge of Combustion Devel., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester P.O.; *for mail,* 73 Kingsman Rd., Boothwyn, Pa.
Walsh, Frank Ogden, Jr. ('41) (ER), Dist. Sales Mgr., Whitcomb Loco. Co., 401 Volunteer Bldg., Atlanta 3, Ga.
Walsh, Frederick Francis (J'40) (SCMYFH), Tech. Supvr., Steam & Power Plant, St. Clair Processing Corp. Ltd., Sarnia, Ont., Can.
Walsh, Gerald William, Jr. (J'44) (ZBK), Test Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1; *for mail,* 408 Wendell Terrace, Syracuse 3, N.Y.
Walsh, Harry Haughton, Jr. (J'38) (ACM), Mfg. Engr., Consltd. Vultee Aircraft Corp., Couchville Pike; *for mail,* 2946 Primrose Circle, Nashville, Tenn.
Walsh, James Lawrence ('35) (CDWRNA), Univ. Club, 1 W. 54th St., New York 19, N.Y.
Walsh, James Paul (J'43) (BNZ), Head, Devel. Sec., Shock & Vibration Div., Naval Research Lab.; *for mail,* 101 Joliet St., S.W., Washington 20, D.C.
Walsh, Joseph ('36) (CHJ), Pres., Walsh Constr. Ltd., Granville Island, Vancouver, B.C., Can.
Walsh, Robert Everett ('43) (SCKOF), Utilities Engr., Head, Utilities Sec., Office of Rubber Reserve, Reconstr. Finance Corp., 1626 K St., N.W., Washington, D.C.
Walsh, Thomas A. ('29) (CGY), Mgr., Safety Engrg. Serv. Bur., Am. Optical Co., 70 W. 40th St., New York; *for mail,* 6 Midland Gardens, Bronxville 8, N.Y.
Walsh, Walter John, Jr. (J'45), Ensign, U.S.N.R.; 4749 Lemona Ave., Sherman Oaks, Calif.
Walsh, William E. ('42) (NABM), Ch. Engr., Design Div., Cowles & Co., Inc., 612 N. Michigan Ave., Chicago 11; *for mail,* Box 1302, Chicago 90, Ill.
Walsh, William James ('15; '22) (CJKM), V.P., Factory Mgr., Delta File Works, Inc., 4837 James St., Philadelphia 37; *for mail,* 618 W. Sedgwick St., Philadelphia 19, Pa.
Walsh, Woodrow (J'37), Capt., Hdq., 7th Engr. Bn., A.P.O. 5, c/o Postmaster, New York, N.Y.
Walter, Delton Earl (J'45) (CDOLMS), Steel Engr. & Estimator, Bethlehem Pacific Coast Steel Co., 3800 Iowa St., Seattle; *for mail,* 3045—42nd St. W., Seattle 99, Wash.
Walter, Harold Edmund ('41) (SEHFKB), Prof. Mech. Engrg., Newark College of Engrg., 367 High St., Newark 2, N.J.
Walter, John M. ('40) (NMJCA), Ch. Engr., G. A. Gray Co., 3611 Woodburn Ave., Cincinnati 7; *for mail,* 6002 Grand Vista Ave., Cincinnati 13, Ohio.
Walter, Martin, Jr. (J'39) (TDCY), Mill Mgr., New Bedford Cordage Co., 131 Court St., New Bedford; *for mail,* 123 Elm St., South Dartmouth, Mass.
Walter, Maurice Albert (J'42) (ABZC), Sr. Flight Test Engr., Cornell Aero. Lab., Buffalo 5, N.Y.
Walter, Ralph Edwin ('28; '35) (C), Mech. Supvr., Frye & Co., 2203 Airport Way, Seattle 14; *for mail,* 3639 Magnolia Blvd., Seattle 99, Wash.
Walter, Stanley T. (J'38), 8310—35th Ave., Jackson Heights, L.I., N.Y.
Walters, Franklin Claire (J'42), Liaison Engr., Glenn L. Martin-Neb. Co., Ft. Crook, Omaha; *for mail,* Apt. 20, 2208 Capital Ave., Omaha 2, Neb.
Walters, Howard Robert (J'45), Test Engr., Oliver Corp., 300 Lawler St.; *for mail,* 204 Ferguson, Charles City, Iowa.
Walters, J. E. ('35) (CDJ), Pres., Alfred Univ., Alfred, N.Y.
Walters, John Calverley ('33; '43) (MCJGEA), 19 Winchester Rd., Lakewood, N.Y.
Walters, Russell J. ('45) (OSMNB), Owner, Mgr., Walters Engrg. Co., Fox Theater Bldg., Detroit 1, Mich.
Walther, Paul Hugo ('20) (FKS), Pres., Ch. Engr., Am. Chimney Corp., 147—4th Ave., New York 3, N.Y.; *for mail,* 1020 Abbott Blvd., Palisade, N.J.
Walton, Earle ('21; '35) (S), Order Interpreter, Westinghouse Elec. Corp., 1180 Raymond Blvd., Newark 2; *for mail,* 43 S. Walnut St., East Orange, N.J.
Walton, Edward Haviland ('44) (SCJFEK), Mech. Engr., United Illum. Co., 80 Temple St., New Haven 6, Conn.
Walton, Eugene Gordon, Jr. (A'44) (MCJDN), Prod. Engr., John E. Mitchell Co., 3800 Commerce St., Dallas 1, Tex.
Walton, Hiram Lessiter ('36) (CDEKMS), Pres., Smith, Hinchman & Grylls, Inc., 800 Marquette Bldg., Detroit 26, Mich.
Walton, James Scott (J'43) (AMR), Ensign, U.S.N.R., 5325 Ellsworth Ave., Pittsburgh, Pa.
Walton, Robert E. (J'37) (HLDNO), Dist. Mgr., Alexander-Walton & Co., Wilmington; *for mail,* 28 S. Pennewell Dr., Edgemoor Gardens, Wilmington 261, Del.
Walton, Sylvan B. (J'35) (BKS), Asst. Prof. Mech. Engrg., Univ. of Ky.; *for mail,* 719 Avalon Park, Lexington, Ky.
Walz, Rex Perry (J'40) (ACLM), 712—6th St., Brookings, S.D.
Wambaugh, Russell Harrison ('43) (RCDS), Mech. Engr., Dept. of Commerce, Washington 25, D.C.; *for mail,* 305 Ellsworth Dr., Silver Spring, Md.
Wan, Conrad Chang-Yen (J'42) (ABK), Struc. Research Engr., Chance Vought Aircraft Div., United Aircraft Corp., Sniffen Lane, Stratford; *for mail,* 126 Post Rd., Fairfield 1, Conn.
Wanamaker, George Knight, Jr. ('41), Babcock & Wilcox Co., 2511 Carew Tower, Cincinnati, Ohio.
Wander, Myron (J'45) (SKFCOE), Design Engr., Mar. Dept., Combustion Engrg. Co., Inc., 200 Madison Ave., New York; *for mail,* 110 Riverside Dr., New York 24, N.Y.
Wandrey, Erwin (J'33) (ABC), 6150 S. Greenwood, Chicago 37, Ill.
Waner, Norman S. (J'44) (NBSMKJ), Designing & Devel. Engr., Am. Engrg. Co., Kensington Sta., Philadelphia 25; *for mail,* 1324 Disston St., Philadelphia 11, Pa.
Wang, Shou Chin ('44) (CEJMNS), Chmn., Chinese Supply Comm., 2311 Massachusetts Ave., N.W., Washington 8, D.C.
Wang, Tzyh-Shin ('44) (JMR), Universal Trading Corp., 630—5th Ave., New York 20, N.Y.
Wang, Zu-Fong ('45) (CD), Managing Dir., Cent. Auto Parts Mfg. Works, 5 Loong Ying Rd., Hwa-Loong Chiao, Chungking, China.
Wangelin, Don J. (J'45) (JZCNO), Plant Engr., Pure Oil Co., Box 266, Winnetka; *for mail,* 1624 Mt. Pleasant Rd., Northfield, Ill.
Wangelin, Hugo (J'46), 73 Academy Rd., Bala-Cynwyd, Pa.
Wannall, George Lawrence (J'45) (AKB), 166 Edgewater Apts., Baltimore 21, Md.
Wanner, Ralph (J'35) (NGLYJ), Designing Draftsman, Sylvania Elec. Products, Inc., Mill Hall; *for mail,* Mackeyville, Pa.
Wannop, Leonard Graham (J'46), Box 717, c/o Lago Oil & Transport Co., Ltd., Aruba, N.W.I.
Wanski, Morris Harold (J'45) (BHGNOY), Sales Engr., Morse Instrument Co., Hudson; *for mail,* 627 Fernwood Dr., Akron 2, Ohio.
Wanzer, Arthur William ('46), Ch. Engr., Murray & Tregurtha, Inc. & Mathewson Mch. Works, 2 Hancock St.; *for mail,* 10 Holyoke St., North Quincy, Mass.
Ward, Charles Allen ('32) (GCMNY), Pres., Sales & Gen. Mgr., Brown & Bigelow, St. Paul 4, Minn.
Ward, Clyde Q. ('37) (SFKO), Supt., Heat Dept. & Engrg., St. Joseph Light & Power Co., 520 Francis St., St. Joseph 2, Mo.
Ward, Harry Cameron (J'41) (JLMN), Mech. Engr., Cube Steak Mch. Co., 2 Northhampton St., Boston; *for mail,* 167 Everett St., Wollaston 70, Mass.
Ward, J. Carlton, Jr. ('17; '25) (AC), Pres., Fairchild Eng. & Airplane Corp., 30 Rockefeller Plaza, New York 20, N.Y.
Ward, J. Kenneth (J'45) (A), 1235 Westgate Ave., West Los Angeles, Calif.
Ward, John Henry ('41), Prod. Engr., St. Joseph's Ry., Light, Heat & Power Co.; *for mail,* 2710 Douglas St., St. Joseph, Mo.
Ward, John W., Jr. (J'36) (BDM), 4224 Clairmont Ave., Birmingham, Ala.
Ward, Joseph Augustus, Jr. (J'45), Sunset Rd., Pompton Plains, N.J.
Ward, Nairne Forsyth ('21; '34; '35) (AJM), Matls. Engr., Naval Air Sta., Alameda; *for mail,* 2330 Rose Terrace, Berkeley 8, Calif.
Ward, Wallace Wesley (J'44) (BHM), Design Engr., Universal Camera Corp., 28 W. 23rd St., New York, N.Y.; *for mail,* 180 Summit Ave., Bloomfield, N.J.
Ward, William Edward (J'32) (BAYHJM), 2599 Lee Rd., Cleveland Heights 18, Ohio.
Ward, Winslow A., III (J'41) (A), Plastics Engr., Am. Cyanamid Co., 30 Rockefeller Plaza, New York; *for mail,* 80-24 Broadway, Elmhurst, L.I., N.Y.
Warden, Guy Lane ('16; '23; '29) (SZKF), Power Equip. Engr., 209 W. 12th St., Los Angeles 15; *for mail,* 2232 Moreno Dr., Los Angeles 26, Calif.
Wardle, Thomas Forster (J'44), 266—79th St., Brooklyn 9, N.Y.
Ware, David Herndom ('45), Gen. Elec. Co., 1 River Rd.; *for mail,* 2328 Barcelona Rd., Schenectady, N.Y.
Ware, John H., III (J'42) (CFOEJM), V.P., Gas-Oil Products, Inc., 45 S. 3rd St., Oxford, Pa.
Ware, John S. (J'27) (CKL), c/o Gen. Mills, Inc. do Brasil, Avenida Rio Branco, 277-sala 1508, Rio de Janeiro, Brazil, S.A.
Ware, Marsden (J'22) (BENJFC), Ch. Engr., Mar. Eng. Div., Packard Motor Car Co., 1580 E. Grand Blvd., Detroit 32; *for mail,* 8276 Huntington Rd., Huntington Woods, Mich.
Wareham, James K. ('38) (ACJL), Supt., Jobbing Div., Aluminum Cooking Utensil Co., New Kensington, Pa.
Warfel, J. Richard (J'41) (ACM), Aero. Serv. Engr., Sperry Gyroscope Co., Inc.; *for mail,* 452 N. Prince St., Lancaster, Pa.
Warfield, H. Ridgeley, Jr. ('40), Partner, Fonda & Warfield, 326 St. Paul Pl., Baltimore, Md.

Waring, George Houstoun ('23) (CEFOY), Cons. Engr., McInerney Spring & Wire Co., 655 Godfrey Ave., Grand Rapids 2; *for mail*, 1611 Pontiac Rd., Grand Rapids 6, Mich.

Waring, Robert W. ('30; '44) (ABCJMY), Ch. Engr., E. Main Plant, Bridgeport Brass Co., 30 Grand St., Bridgeport 2, Conn.

Warming, Troels (J'41) (EB), *Junior Award*, '43; Diesel Eng. Div., Harnischfeger Corp., Port Washington, Wis.

Warmington, Thomas J. ('20; '27; '35) (BDHMNS), Dir. Engrg., Wm. Bros. Boiler & Mfg. Co., 10th Ave., S.E., Minneapolis 14, Minn.

Warneke, Maynard J. ('45) (CYMNLJ), Trng. Supvr., Westinghouse Elec. Corp., East Pittsburgh, Pa.

Warner, Cecil F. (J'39) (HKS), Instr., Purdue Univ., West Lafayette, Ind.

Warner, Charles Marshall ('26) (BD), Supt., Wire Mill, Scovill Mfg. Co., 99 Mill St., Waterbury; *for mail*, 430 Frost Rd., Waterbury 54, Conn.

Warner, Donald Franklin ('26; '35) (ABKME), Asst. Designing Engr., Gen. Elec. Co., 920 Western Ave., Lynn; *for mail*, 196 Aspen Rd., Swampscott, Mass.

Warner, Edward ('17; '25; '38) (ABRG), Pres., Interim Council, Prov. Internatl. Civil Aviation Organization, 1212 Dominion Sq. Bldg., Montreal 2, Que., Can.

Warner, Jacob Latch ('17) (CDGO), Spec. Asst., Serv. Dept., E. I. du Pont de Nemours & Co., Du Pont Bldg., Wilmington 91; *for mail*, 2308 W. 11th St., Wilmington 56, Del.

Warner, John A. C. ('37) (AE), Secy., Gen. Mgr., Society of Automotive Engineers, Inc., 29 W. 39th St., New York 18; *for mail*, Apt. 3-G, The Princeton Apts., 675 N. Terrace Ave., Mt. Vernon, N.Y.

Warner, John E. A. ('41) (BFLNSW), Ch. Engr., Robert Gair Co. Inc., 155 E. 44th St., New York 17, N.Y.; *for mail*, 280 West Ave., Milford, Conn.

Warner, Leslie Thomas ('36) (SFCDH), Constr. Engr., Sydney County Council, Queen Victoria Bldg., George St., Sydney, N.S.W., Australia.

Warner, Oswin Vinus (J'36) (JNY), Fdy. Supvr., Malleable Iron Fittings Co., Branford; *for mail*, 2648 Whitney Ave., Hamden 14, Conn.

Warner, Richard F., Jr. (J'33) (CJ), Employee Relations Dept., Research & Statistical Div., Stand. Oil Co. of N.J., 26 Broadway, New York; *for mail*, 16 Wakefield Rd., Eltingville, S.I. 12, N.Y.

Warner, Russell Gillette ('43) (CFKS), Ch. Engr., United Illum. Co., 80 Temple St., New Haven 6, Conn.

Warner, Stan E. (J'41), 1448 Panhandle Blvd., Richmond, Calif.

Warner, Thomas Clark, Jr. (J'42) (AZ), Capt., Army Air Forces, 333-A Harvard, Cambridge, Mass.

Warner, William Bernard (J'40) (SE), Steam Serv. Engr., Westinghouse Elec. Corp., 40 Wall St., New York 5; *for mail*, 1040 E. 27th St., Brooklyn 10, N.Y.

Warner, William Kenneth (J'45) (AHY), Engr., Charge of Design, Appliance Div., Hydro-Aire, 626 N. Robertson, Los Angeles; *for mail*, 318 W. Elmwood Ave., Burbank, Calif.

Warren, A. Joel (J'39) (FBO), Capt., Inf., U.S.A., Box 1577, Ft. Benning, Ga.; *for mail*, c/o Henry Frink, South Coventry, Conn.

Warren, Aldred K. ('12) (CLM), Retired; Shore Acres, Mamaroneck, N.Y.

Warren, Charles William (J'43), Tech. Asst. to Gen. Foreman, Pan Am. Refining Corp., Box 401, Texas City; *for mail*, Box 364, LaMarque, Tex.

Warren, Edward J. (J'40) (CDJ), 1307 King St., Wilmington, Del.

Warren, Everett Milo (J'37) (MJE), Tool Engrg. Dept., Talon, Inc., 626 Arch St.; *for mail*, R.D. 7, Meadville, Pa.

Warren, Glenn Barton ('24; '30) (SRABCE), Designing Engr., Turbine-Generator Div., Gen. Elec. Co., Schenectady 5, N.Y.

Warren, J. P. ('22; '24) (BFS), 1306 W. 4th St., Freeport, Tex.

Warren, Mead, Jr. (J'41) (SHBZG), Elec. Engr., Aluminum Co. of Am., Alcoa; *for mail*, Anderson Ave., Maryville, Tenn.

Warren, Raymond W. (J'37) (NBMCJ), Ord. Engr., Bur. of Ord., Navy Dept., Constitution Ave., Washington, D.C.; *for mail*, 206 W. Cameron Rd., Falls Church, Va.

Warthman, Kenneth L. (J'39) (BM), 1st Lt., Ord. Dept., Chief, Gage Lab., N.Y. Ord. Dist., War Dept.; *for mail*, 445 Central Ave., Orange, N.J.

Warwick, Thomas Machell (J'42) (ACS), Designer, Fed. Telecommunications Lab., Inc., 500 Washington Ave., Nutley; *for mail*, 87 McKinley St., Nutley 10, N.J.

Washburn, Elliott Sorino (J'40), 211 Crandall St., Brunswick, Ga.

Washburn, Franklin Edmond ('34; '35) (FSK), Dist. Supt., Serv. & Erection Dept., Combustion Engrg. Co., Inc., 1616 Walnut St., Philadelphia 3; *for mail*, Box 73, Furlong, Bucks Co., Pa.

Washburn, John Merrow (J'21) (CMJTNB), Treas., Merrow Mch. Co., 28 Laurel St., Hartford, Conn.

Washburn, Morgan, Jr. ('23; '30) (EBL), Serv. Engr. & Dealer for C. Lee Cook Mfg. Co. of Louisville, Ky., 1341 S. Hope St., Los Angeles 15, Calif.

Wasisco, Bernard Eugene (J'46), Foreman, Chicago Screw Co., 1026 S. Homan Ave., Chicago 24; *for mail*, 1306 Albion Ave., Chicago 26, Ill.

Wasser, Reuben ('28; '34; '35) (BFS), Mech. Engr., Pub. Serv. Elec. & Gas Co. of N.J., 396 Clinton Ave.; *for mail*, 50 Elmwood Ave., Irvington, N.J.

Wasserlauf, Bernard Charles (J'43), Master Sgt., Trans. Corps, U.S.A., 1111th Port Mar. Maint. Co., A.P.O. 75, San Francisco, Calif.; *for mail*, 238 Wainwright St., Newark 8, N.J.

Wasson, John W. (J'25) (CESLFD), Mgr., Engr. & Mchy. Sales Div., Von Hamm-Young Co. Ltd., Box 2630, Honolulu 3, T.H.

Waterbury, Lewis Clement ('24) (RMC), V.P., Mgr., Ponce & Guayama R.R., Cent. Aguirre Associates, Guayama, P.R.

Waterfall, Harry William ('21) (NSKF), Prof., Head, Mech. Engrg. Dept., La. State Univ., Baton Rouge 3, La.

Waterman Edwin S. (CESFO), Dist. Mgr., Busch-Sulzer Bros.-Diesel Eng. Co., 3300 S. 2nd St., St. Louis 18; *for mail*, 3935 Fillmore St., St. Louis 16, Mo.

Waterous, Charles Alexander ('16), Retired; 69 Dufferin Ave., Brantford, Ont., Can.

Waters, Daniel Vaughn ('17; '21; '27) (MBJLO), Mech. Engr., West. Elec. Co., Inc., 100 Central Ave., Kearny; *for mail*, "Wa-Rid," R.F.D. 2, Flemington, N.J.

Waters, Everett O. ('14; '25; '35) (BNAHR), Prof. Mech. Engrg., Yale Univ., 15 Prospect St., New Haven 11, Conn.

Waters, Vincent Frederick ('44) (BDGLMW), Gen. Mgr., Ernest H. Abernethy Publ. Co., Inc., 75—3rd St., N.W., Atlanta, Ga.

Waters, William Laurence ('09) (ORCSMN), Partner, Bury & Waters, 150 Nassau St., New York 7; *for mail*, 83 N. Broadway, White Plains, N.Y.

Watkins, Arold ('45) (SCDFL), Draftsman, Kodak Park Works, Eastman Kodak Co., Rochester 4; *for mail*, 96 Mayfair Dr., Rochester 5, N.Y.

Watkins, Charles Albert (J'39) (MND), Engrg. Draftsman, Pangborn Corp., Pangborn Blvd.; *for mail*, 840 Mulberry Ave., Hagerstown, Md.

Watkinson, Roland Martin ('35; '43) (CSE), Estimator, Fed. Shipbldg. & Dry Dock Co., 744 Broad St., Newark, N.J.; *for mail*, 796 E. 37th St., Brooklyn 10, N.Y.

Watlington, E. Hugh ('31), Dept. Mgr., Pearman Watlington & Co., Hamilton, Bermuda, B.W.I.

Watson, Bruce Bastress ('44) (RFKJS), Res. Inspr., Test Dept., Pa. R.R.; *for mail*, 1300—20th Ave., Altoona, Pa.

Watson, David Milnes (J'45) (SKOEFC), Engrg. Design Dept., Wickes Boiler Co.; *for mail*, 1320 Owen St., Saginaw, Mich.

Watson, Douglas (J'43) (CMJD), Asst. to Ch. Indus. Engr., Columbia Steel Co., 235 Montgomery St., San Francisco 6; *for mail*, 37 Northampton Dr., Palo Alto, Calif.

Watson, Fred Burton (J'45), Westinghouse Elec. Corp.; *for mail*, Box 2, Paul, Idaho.

Watson, Harold Fielding ('43) (DBJ), Stands. Engr., Link-Belt Co., 2045 Hunting Park Ave., Philadelphia; *for mail*, 519 E. Tulpehocken St., Philadelphia 44, Pa.

Watson, Harold H. ('28) (CMS), Mgr., Elec. Operas., No. States Power Co., 15 S. 5th St., Minneapolis 2, Minn.

Watson, Harry Dexter ('38) (SEFAW), Prof., Head, Mech. Engrg. Dept., Univ. of Me., Orono, Me.

Watson, Henry Daniel ('35) (KSCDBL), Ch Tech. Engr., Mitchell Engrg. Ltd., Bedford Sq., London, W.C.1; *for mail*, 68 Bullhead Rd., Boreham Wood, Herts, England.

Watson, Herbert Lockridge ('07; '21) (CSJMNR), Pres., De Laval Steam Turbine Co., Trenton 2; *for mail*, 27 Whittier Ave., Trenton 8, N.J.

Watson, Max Powell ('39; '45) (EFMOY), Dist. Engr., United Gas Pipe Line Co. & Union Producing Co., Box 1628, New Orleans 11, La.

Watson, Ralph M. ('40; '44) (BNCHSE), Asst. to V.P. in Charge of Engrg., Worthington Pump & Mchy. Corp., 401 Worthington Ave., Harrison; *for mail*, 23 Hazelwood Rd., Bloomfield, N.J.

Watson, Raymond H. ('28; '34; '35) (BDJHSZ), Mech. Engr., U.S. Engr. Office, 10 E. 17th St., Kansas City 8; *for mail*, 5011 Park St., Kansas City 4, Mo.

Watson, Robert ('43) (RYFABZ), V.P., Waugh Equip. Co., 310 S. Michigan Ave., Chicago 4, Ill.

Watson, Robert Fulton (J'44), 1111 Union St., Alameda, Calif.

Watson, William ('20), Retired; 1464 S. 76th St., West Allis, Wis.

Watson, William Freeborn (J'37) (CMLYJ), 63 Longfellow, Detroit 2, Mich.

Watson, William Malcolm (J'41) (BCM), Mech. Engr., Charge Design, Hygrade Sylvania Corp., Danvers; *for mail*, Salem Country Club, Peabody, Mass.

Watt, John Reid (J'36) (MAEYO), Instr. in Mech. Engrg., Univ. of Tex., Austin; *for mail*, 1011 W. 31st St., Austin 21, Tex.

Watters, Ray Bradshaw, Jr. (J'43) (CLM), Aviation Electronics Technician, U.S.N., Box 15-17, Akron; *for mail*, 1328 Delia Ave., Akron 2, Ohio.

Watters, S. Edmund (J'42) (NJRLBS), Mem. Tech. Staff, Bell Tel. Labs., Inc., 463 West St., New York 14, N.Y.

Watts, George W. ('24; '35) (LBKJZS), Dir. Engrg., Stand. Oil Co. of Ind., 910 S. Michigan Ave., Chicago 80, Ill.

Watts, Robert Lucien ('18) (CELMOS), Mgr., Lubriplate Div., Fiske Bros. Refining Co., 129 Lockwood St., Newark 5, N.J.; *for mail*, 25 Central Park W., New York 23, N.Y.

Wauchope, George Andrew ('44) (HBNSZ), Dir., Gwynnes Pumps Ltd., Hammersmith, London, W. 6; *for mail*, 24 Hertford Ave., East Sheen, London, S.W. 14, England.

Waugaman, Arthur Rankin ('46), Dist. Mgr., Babcock & Wilcox Co., 1525 Ford Bldg., Detroit; *for mail*, 253 Merriweather Rd., Grosse Pointe Farms, Mich.

Way, David Spencer (J'44) (ASZK), Test Engr., Wright Aero. Corp., Paterson 3; *for mail*, 21 N. 13th St., Haledon, Paterson 2, N.J.

Way, Stewart (J'39) (BEAFSJ), Sec. Engr., Thermodynamics Research Labs., Westinghouse Elec. Corp., East Pittsburgh, Pa.

Wayman, Robert William (J'40) (CHNMAJ), Project Engr., Detroit Transmission Div., Gen. Motors Corp., 5140 Riopelle St., Detroit 11; *for mail*, 21650 Moross Rd., Detroit 24, Mich.

Waymouth, Geo. W. ('39), R.F.D. 10, Loudon Branch, Concord, N.H.

Wean, Raymond John ('28) (JBCLMY), Pres., Wean Engrg. Co., Inc., 347 N. Park Ave., Warren, Ohio.

Weatherhead, Charles Frank (J'43), 274 St. Clarens Ave., Toronto, Ont., Can.

Weatherly, Richard M. (J'44) (EAH), Route 4, Greensboro, N.C.

Weaver, Clarence Roberts (J'40) (RCDFS), Gang Foreman, Meadows Enginehouse, Pa. R.R. Co., Kearny, N.J.; *for mail*, 15 N. 2nd St., Sunbury, Pa.

Weaver, Earl Lewis, Jr. (J'43), Student Engr., Natl. Tube Co.; *for mail*, 225 Oxford Dr., McKeesport, Pa.

Weaver, Earl Marshall (J'39) (JMCB), Ch. Engr., Gray Tool Co., Box 1655, Houston 1, Tex.

Weaver, Edgar L. (J'42) (SRBH), 1809 Berkley Ave., S.W., Grand Rapids 9, Mich.

Weaver, Ernest Witwer ('20; '26) (FNKJCB), Asst. Ch. Engr., Surface Combustion Corp., Toledo 1; *for mail*, 1622 Potomac Dr., Toledo 7, Ohio.

Weaver, Frank Roderick ('23) (CJTWM), Pres., Hibben, Hollweg & Co., 110 S. Meridian St., V.P., E. C. Atkins & Co., 402 S. Illinois St.; *for mail*, 110 S. Meridian St., Indianapolis, Ind.

Weaver, James R. ('36) (CDM), Plant Mgr., Westinghouse Elec. Corp., 653 Page Blvd., Springfield 2, Mass.

Weaver, Leon H. A. ('21; '25) (CSL), 423 Birch Pl., Westfield, N.J.

Weaver, Richard Durbin (J'40) (NBMJAO), Mech. Engr., Hallicrafter Co., 2611 Indiana Ave., Chicago 11; *for mail*, 5741 Kenmore Ave., Chicago, Ill.

Weaver, William Edward (J'34) (YWKJA), Lt. Comdr., Bur. of Aero., Navy Dept.; *for mail*, 3207 Foxhall Rd., N.W., Washington, D.C.

Weaver, William Henry ('27; '32 '35) (CMDO), Ch. Indus. Engr., Crown Can Co., Erie Ave. & I St., Philadelphia 34; *for mail*, 326 Derwyn Rd., Lansdowne, Pa.

Webb, Albert Erickson (J'39) (HNBCZ), Mech. Engr., Stewart-Warner Corp., 1826 Diversey Pkwy., Chicago 14; *for mail*, 3757 N. Fremont St., Chicago 13, Ill.

Webb, Baxter Hill (J'36) (CADRBE), Assoc. Automotive Engr., Dept. of Street Rys., City of Detroit, 14129—2nd Blvd., Highland Park 3; *for mail*, 14912 Rosemary, Detroit 5, Mich.

Webb, Benjamin Wellington ('40) (CFS), Philadelphia Dist. Mgr., Combustion Engrg. Co., Inc., 1616 Walnut St., Philadelphia, Pa.

Webb, Charles Carrick ('41) (NBYDMZ), Mech. Engr., Mchy. Devel. Gp., Engrg. Dept., E. I. du Pont de Nemours & Co.; *for mail*, 1303 Rodney St., Wilmington, Del.

Webb, Clifford L. (M'43), Creamery Package Mfg. Co., 692 Ponce de Leon Ave., Atlanta, Ga.

Webb, Ernest Clarence ('44) (FKS), Engrg. Serv. Mgr., Iron Fireman Mfg. Co., 3170 W. 106th St., Cleveland; *for mail*, 24721 West Lake Rd., Bay Village, Ohio.

Webb, Frank Kline, Jr. (J'39) (ALC), Maj., 410-7 Kearny, Ft. Leavenworth, Kan.

Webb, George Henry (J'44) (D), Asst. Ch. Engr., Jervis B. Webb Co., 8951 Alpine Ave., Detroit 4, Mich.

Webb, Jervis Bennett ('27) (CD), Pres., Gen. Mgr., Jervis B. Webb Co., 8951 Alpine Ave., Detroit, Mich.

Webb, Jervis Campbell (J'37) (DMCB), Mech. Engr., Asst. to Presiding Officer, Jervis B. Webb Co., 8951 Alpine Ave., Detroit 4, Mich.

Webb, John Henry Edward (J'46), 163 Earlsdale Ave., Toronto 10, Ont., Can.

Webb, Ralph Duvivier ('42) (ZLSC), Head, Instrument Dept., Carbide & Carbon Chemicals Corp., South Charleston 3, W.Va.

Webb, Robert G. (J'39) (BHK), Lab. Technician, Chrysler Corp., 12800 Oakland Ave., Detroit; *for mail*, 1327 Troy-West, Ferndale, Mich.

Webb, Thomas Clifford (J'38) (BEKM), Sales Engr., Clark Bros. Co., Inc., 5715 Bickett, Huntington Park, Calif.

Webb, Thomas Lake Berry ('44) (OSHKZF), Staff Engr., Utility Consultant, So. Calif. Gas Co. Rubber Reserve, 553 Rameriz St., Los Angeles; *for mail,* 815 Northwood Ave., Compton, Calif.
Webber, Harold S. ('39) (CDGM), Plant Supt., Mech. Engr., Schorsch & Co., 120 Bruckner Blvd., Bronx 54, N.Y.
Webber, Laurance Edmund (J'40) (WSFKZ), Research Asst. Prof. Indus. Engrg., Engrg. Exper. Sta., Univ. of N.H., Durham, N.H.; *for mail,* 53 Rochester St., Berwick, Me.
Weber, Albert Manning ('45) (CMD), Gen. Mgr., Clark & Clark, Southwest Ave., Wenonah, N.J.
Weber, Andrew Raymond ('36) (KSFNA), Prof., Head, Mech. Engrg. Dept., Univ. of Dayton, Dayton 9, Ohio.
Weber, August, Jr. ('16) (CDEMN), Retired; 1155 Stratford Rd., Schenectady 8, N.Y.
Weber, Eugene ('23), 1608 Cook St., Denver 6, Colo.
Weber, Frank A. (J'42) (JYAM), Field Engr., Firestone Indus. Products Co., Akron; *for mail,* 1620 Thornapple St., Akron 1, Ohio.
Weber, Fred Alexander (J'43) (MJCLDB), Maint. Mechanic & Supvr., Lamp Dept., Hoboken Works, Gen. Elec. Co., 410—8th St., Hoboken; *for mail,* 277 Liberty Ave., Jersey City 7, N.J.
Weber, George Clyde ('45) (SECL), Foreman, Allison Div., Gen. Motors Corp., Indianapolis; *for mail,* 849 Gilbert Ave., Indianapolis 3, Ind.
Weber, Henry Francis (J'44) (NBAHZC), Mech. Engr., Natl. Advis. Com. for Aero., Langley Field; *for mail,* 260 Regent St., Hampton, Va.
Weber, Homer Squire ('24; '31; '35) (BAJN), Prof. Engrg. Drawing & Mechanics, Ga. Sch. of Tech., Atlanta, Ga.
Weber, James George (J'42) (JMC), Fdy. Prod. Engr., Ampco Metal, Inc., 1745 S. 38th St., Milwaukee 4; *for mail,* 8535 Hawthorn, Wauwatosa 13, Wis.
Weber, John T. (J'40) (BCJMN), Engr., Weber Trailer & Mfg. Co., 1100 E. 5th St., Los Angeles 13; *for mail,* 1643½ Carmona Ave., Los Angeles 35, Calif.
Weber, L. Joe (J'37) (FEKJD), Metallurgist, Gasoline Dept., Phillips Petroleum Co.; *for mail,* Box 143, Bartlesville, Okla.
Weber, Marion Huber (J'39) (YBRK), Mech. Engr., Design, B. F. Goodrich Co., 500 S. Main St., Akron; *for mail,* 146 N. Prospect St., Kent, Ohio.
Weber, Nicholas (J'33), Ch. Engr., C. H. Leach Co., 117 Liberty St., New York, N.Y.; *for mail,* 15 Tessen St., Teaneck, N.J.
Weber, Oscar Emile ('29; '35) (HSBC), Mgr., Contract Serv. Dept., S. Morgan Smith Co., 500 Lincoln St.; *for mail,* R.D. 1, York, Pa.
Weber, Paul R. (J'45) (LAYKCO), Engr., Tech. Sec., E. I. du Pont de Nemours & Co., Box 993, Charleston, W.Va.
Weber, Philip Francis (J'26) (ACMJN), Factory Mgr., Kollsman Instrument Div., Square D Co., 80-08—45th Ave., Elmhurst, N.Y.
Weber, Ralph Eugene, Jr. (J'46), Lt., U.S.N.R.; 69 Hobart Ave., Summit, N.J.
Weber, Robert Johnston ('44) (SRHKEC), Mgr., Cent. Sta. & Trans. Div., Westinghouse Elec. Corp., 306—4th Ave., Pittsburgh 30, Pa.
Weber, Theodore R. ('37) (BJR), Ch. Mech. Engr., Ry. Steel Spring Div., Am. Loco. Co., Latrobe, Pa.
Weber, William Reeb (J'40) (RACM), Gang Foreman, Office of Ch. of M.P., Pa. R.R. Co., Philadelphia, Pa.; *for mail,* 4728—29th St. S., Arlington, Va.
Webers, Harold Flory (J'46), Devel. Engrg., Titanium Div., Natl. Lead Co., Carondelet Sta., St. Louis 11; *for mail,* 5152-A Eichelberger, St. Louis 9, Mo.
Webre, Alfred L. ('16), Mech. Engr., Merion, Pa.
Webster, Daniel James (J'36) (C), 1st Lt., Ord. Dept., U.S.A., 414th Ord. Evac. Co., A.P.O. 124, c/o Postmaster, New York, N.Y.
Webster, Daniel Thomas, Jr. ('32; '37) (SKZHBO), Mech. Engr., Clyde R. Place, 420 Lexington Ave., New York 17, N.Y.
Webster, Earl Parker ('43) (CDGKMN), Mech. Engr., Natl. Biscuit Co., 449 W. 14th St., New York 11, N.Y.
Webster, Fred N. (J'40) (FKS), Instr. in Mech. Engrg., Engrg. Sch., Tufts College; *for mail,* 34 Jackson Rd., Medford 55, Mass.
Webster, George Conner (J'43) (ONBFZG), Mech. Engr., Spec. Rep., Engrg. Devel. Co., 627 F St., N.W., Washington 4, D.C.
Webster, Harry D. ('17) (REA), Retired; 392 S. Main St., Greenville, Pa.
Webster, John D. ('22; '35) (SFEHKB), Mech. Engr., Ebasco Internatl. Corp., 2 Rector St., New York 6, N.Y.; *for mail,* 208 Tenafly Rd., Englewood, N.J.
Webster, Joseph F. (J'28) (ACR), Draftsman, Bethlehem Steel Co., 3rd St.; *for mail,* 624 W. Union Blvd., Bethlehem, Pa.
Webster, Lawrence Burns ('10; '14; '18) (SJRG), 35 Boudinot St., Princeton, N.J.
Wechsberg, Otto ('21) (BEFJKN), Pres., Gen. Mgr., Coppus Engrg. Corp., 344 Park Ave., Worcester 2, Mass.
Weckerle, John Harry (J'44) (SKHCND), Asst. Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3; *for mail,* R.F.D. 1, Roscoe, N.Y.
Weckstein, Samson Morris ('24; '35) (JNEBYR), Ch. Engr., Indus. Div., Timken Roller Bearing Co., Dueber Ave., S.W., Canton 6; *for mail,* 204—25th St., N.W., Canton 3, Ohio.
Weddington, Jim Ray (J'42) (SHAEKF), Weight Analyst, Douglas Aircraft Co., Inc., 3000 Ocean Park Blvd., Santa Monica; *for mail,* 10469 Kinnard Ave., Los Angeles 24, Calif.
Weddle, Alfred Harvey, Jr. (J'43) (BSCAKN), Engrg. Asst. to Gen. Mgr., Vassar College, Poughkeepsie, N.Y.
Wedge, Donald Eugene (J'39) (CMJOWY), Departmental Foreman of Inspc., A.C. Spark Plug Div., Gen. Motors Corp., 1300 N. Dort Highway, Flint; *for mail,* 2521 Stevenson St., Flint 4, Mich.
Wedner, Benjamin Mayer ('43) (BNMEJL), Asst. Ch. Draftsman, Gulf Research & Devel. Co., P.O. Drawer 2038, Pittsburgh 30; *for mail,* 6520 Lilac St., Pittsburgh 17, Pa.
Weeks, Dan Eugene ('39; '45) (SKFW), Supvg. Engr., Robert L. Johnson Co., 344 Monadnock Bldg., San Francisco 5; *for mail,* 2344 Virginia St., Berkeley 4, Calif.
Weeks, DeWitt C. ('41) (FSZK), Performance Engr., Consltd. Edison Co. of N.Y., Inc., Room AU-14, 4 Irving Pl., New York 3, N.Y.
Weeks, John Harding (J'45) (MAJZ), Mech. Engr., Natl. Advis. Com. for Aero., Cleveland Airport, Cleveland; *for mail,* 1640 Wooster Rd., Rocky River, Ohio.
Wege, Ervin Carl (J'42) (HJWYAB), Norrie, Wis.
Wegener, Francis A. (J'15) (CEM), V.P., Gen. Mgr., Welsbach Co.; *for mail,* 105 N. Brown St., Gloucester City, N.J.
Wegg, David Spencer ('09; '14; '21) (EFLMOT), Engr., Jackson & Moreland, 31 St. James Ave., Boston 16; *for mail,* 25 Martin Rd., Wellesley 81, Mass.
Wegner, Harold Leroy (J'44) (JMD), Draftsman, Mich. Bumper Corp., 740 Ann St., Grand Rapids 2; *for mail,* 1216 Valley Ave., N.W., Grand Rapids 4, Mich.
Wehmeyer, Clarence William ('37) (SJBFKZ), Mech. Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16; *for mail,* 12 Herbert Ave., White Plains, N.Y.
Wehr, C. Fred'k ('18; '35), Pres., Treas., Wehr Steel Co., 5234 W. Mobile St., Milwaukee, Wis.
Wehrmann, Wilhelm (J'37) (CEMS), Supvr., N.Y. Naval Shipyard, Brooklyn; *for mail,* 4329 Forley St., Elmhurst, L.I., N.Y.
Wei, Y. F. ('45) (Prof.), Dir., Tech. Dept., Ministry of Communications, Chungking, China; *for mail,* c/o Prof. P. B. Eaton, Lehigh Univ., Easton, Pa.
Weibel, Emil Edwin ('36) (BNAZK), Assoc. Prof. Mech. Engrg., Univ. of Calif., Berkeley 4, Calif.
Weick, Fred Ernest ('44) (ABNE), V.P., Charge of Engrg., Engrg. & Research Corp., Riverdale, Md.
Weidenhammer, James A. (J'39) (BHM), Designer, Internatl. Business Mchs. Corp.; *for mail,* 2713 Watson Blvd., Endicott, N.Y.
Weidner, Carl Birch (J'40) (MEFOH), Lt. (j.g.), Mar. Inspc., U.S.C.G., Fed. Bldg., Mobile 9, Ala.; *for mail,* c/o Mrs. P. G. Weidner, 1232 E. 27th Pl., Tulsa 5, Okla.
Weierich, Andrew Charles ('44) (OACW), Owner, A. C. Weierich Co., Box 973, Ch. Engr., Davidson Enamel Products, Inc., Box 328; *for mail,* Western Ohio Ave., Lima, Ohio.
Weigel, Albert Charles ('14; '19; F'43) (SJEF), V.P., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Weigel, Albert R. ('27) (BCHKOR), Research Engr., Consltd. Steel Corp., Eastern & Slauson, Los Angeles 22; *for mail,* 535 S. Gramercy Pl., Los Angeles 5, Calif.
Weigel, Robert H. (J'40) (ACF), 401 W. Park, Urbana, Ill.
Weil, Richard Lee (J'42) (JBL), Project Engr., Gray Mfg. Co., 16 Arbor St., Hartford 5, Conn.
Weil, Robert S. (J'42) (NYJS), Design Engr., Radar Engrg. Div., Gilfillan Bros., Inc., 1815 Venice Blvd., Los Angeles; *for mail,* 5433½ Lexington Ave., Los Angeles 27, Calif.
Weiland, Edward (J'40) (SFD), Ch. Engr., Mun. Research & Testing Lab. of St. Louis, 55 Mun. Courts Bldg., St. Louis 3; *for mail,* 4961 Laclede Ave., St. Louis 8, Mo.
Weiland, Walter F. ('37), Assoc. Prof. Mech. Engrg., Univ. of Neb., Lincoln, Neb.
Weiler, George Henry (A'42) (ABCJMR), Mgr., R.R. Div., Vanadium Corp. of Am., 420 Lexington Ave., New York 17, N.Y.
Weiler, Robert Edgar (J'43) (ABS), Jr. Design Engr., Research & Devel. Sec., Steam Div., Westinghouse Elec. Corp., Lester; *for mail,* 822—14th Ave., Prospect Park, Pa.
Weill, Melville Kaiser ('21) (ACWOM), Pres., Skydyne, Inc., River Rd., Port Jervis, N.Y.
Weimann, Alfred Frederick ('31) (CLO), Divisional Pur. Agt., U.S. Rubber Co., 1230—6th Ave., New York; *for mail,* 22 Huntington Rd., Garden City, N.Y.
Weimar, Hans Carl ('27; '35; '35) (BNH), Design & Devel. Engr., Brown & Sharpe Mfg. Co., Promenade, Providence 1; *for mail,* 162 Lenox Ave., Providence 7, R.I.
Weinberg, Edwin Bernard (J'39) (KLSAM), Heat Transfer Engr., Therm & Changer, Inc., 625 Market St., San Francisco 5, Engrg. Instr., Modesto Jr. College; *for mail,* 527 N. Orange Ave., Modesto, Calif.
Weinberg, Herbert Lee ('24; '27; '35 (REC), c/o Am. Loco. Co., Schenectady 5, N.Y.
Weinberg, Irving (J'45) (CKS), Asst. Plant Engr., Vilter Mfg. Co., 2217 S. 1st St.; *for mail,* 2819 N. 52nd St., Milwaukee 10, Wis.
Weinberg, Philip H. ('32) (ESCLMF), Sales Engr., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Weinbrecht, John Frederick (J'38) (NBLG), Draftsman, Engrg. Dept., Monarch Mch. Tool Co.; *for mail,* 224 S. Miami Ave., Sidney, Ohio.
Weiner, Arnold Philip (J'45) (NBAHYG), Stress Analyst, Jackson & Moreland, Engrs., 31 St. James Ave., Boston 16; *for mail,* 2 Pama Gardens, Brighton 35, Mass.
Weiner, Lewis P. ('30; '36) (LCSMAF), V.P., Gen. Mgr., Scottish-Am. Distillers Ltd., Foot of Alexander St., Peoria 3; *for mail,* High Point Rd., Peoria 4, Ill.
Weiner, Samuel Zelick ('45) (SOKHBF), Asst. Engr., Gibbs & Hill, Inc., Pa. Sta., New York 1; *for mail,* 501 Ave. R, Brooklyn 23, N.Y.
Weingarten, Morris Robert (J'43) (HNJKBZ), Design Engr., RCA Victor Div., Radio Corp. of Am.; *for mail,* 49 S. Duke St., Lancaster, Pa.
Weinhold, Julius F. ('25; '31; '35) (SCD), Supt. Power & Maint., Winchester Repeating Arms Co., Div. of Olin Industries, Inc., New Haven; *for mail,* 115 Westwood Rd., New Haven 15, Conn.
Weinstein, Alexander ('30) (CO), 120 Broadway, New York 15, N.Y.; *for mail,* 1221 Hilldale Ave., Los Angeles 46, Calif.
Weinstein, Henry R. (J'38), 103 St. George St., St. Augustine, Fla.
Weinstein, Isaac (J'41) (AHJ), Lt., 159 Canal St., Ellenville, N.Y.
Weinthal, Stanley E. (J'42), Prod. Engr., Walters Mfg. Co., Pennsylvania Ave., Oakmont; *for mail,* 4366 Winterburn St., Pittsburgh 7, Pa.
Weintraub, Sidney S. (J'39) (ACK), 443 Allison Dr., Dallas, Tex.
Weir, The Rt. Hon. Viscount (H'20) (MKCHA), Chmn. of Dirs., G. & J. Weir Ltd., Holm Fdy., Cathcart, Glasgow, Lanark, Scotland.
Weir, A. D. (J'44), Weir Appliance & Equip. Co., 6323 Gaston Ave.; *for mail,* 7301 Lakewood Blvd., Dallas, Tex.
Weir, David MacDonald, Jr. ('40) (SLM), Sales Engr., Pittsburgh Piping & Equip. Co., 10—43rd St., Pittsburgh 1; *for mail,* 1109 Evergreen Ave., Millvale, Pittsburgh 9, Pa.
Weir, Frederick Utter ('45) (OSKEGC), Engr., Thomas B. Hunter, Rm. 710, 41 Sutter St.; *for mail,* 2528 Broadway, San Francisco 15, Calif.
Weir, George E. ('27; '37) (SOBKFL), Engr., Mech. Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7, Mass.; *for mail,* 8528—118th St., New York 16, N.Y.
Weir, Samuel Boone (J'44) (EJKRS), Boiler & Turbine Test Engr., Naval Boiler & Turbine Test Lab., S. Broad St.; *for mail,* 7019 Creshiem Rd., Philadelphia 19, Pa.
Weis, Arthur R. ('27; '35) (HCEJNM), Pres., Pacific Pumps, Inc., 5715 Bickett St., Huntington Park, Calif.
Weisbein, Jack (J'40) (HKS), 115 Grand, Kansas City, Mo.
Weisberg, Herman ('37) (SFJEDA), Mech. Engr., Elec. Engrg. Dept., Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Weisberger, Arthur Aldon ('37) (MDRC), Prod. Supt., Am. Steel Fdys., Hammond, Ind.
Weiser, Earle Philip (J'40) (CB), V.P., Charge of Opers., A & W, Inc., 3337 N. Williams; *for mail,* 7412 N. Chase Ave., Portland 12, Ore.
Weiser, Sidney (J'41) (NMCJ), Asst. Engr., Fed. Mfg. & Engrg. Corp., 211 Steuben St., Brooklyn 5; *for mail,* 1519—55th St., Brooklyn, N.Y.
Weishaar, Gerard Paul (J'43) (BCDLMN), Dist. Mgr., Am. Pulley Co., 526 Folsom St., San Francisco 5, Calif.
Weismann, Victor Paul ('35; '42) (OLCDSM), Sales Engr., Bethlehem Pacific Coast Steel Corp., Box 2057, Terminal Annex, Los Angeles, Calif.
Weismantle, Arthur Robert (J'36) (SFCKLH), Mech. Engr., Steam Div., Foster Wheeler Corp., 165 Broadway, New York, N.Y.
Weiss, Alexander ('25; '35), Mech. Engr., Babcock & Wilcox Tube Co.; *for mail,* 3900—4th Ave., Beaver Falls, Pa.
Weiss, Alexander ('39) (ZHBDSL), Commercial Trust Bldg., Philadelphia 2, Pa.
Weiss, Arthur J. ('22; '26; '35) (CFLS), Plant Engr., Lowe Paper Co., River St., Ridgefield; *for mail,* 14 Kenwood Rd., Tenafly, N.J.
Weiss, Charles Robert ('40) (ABDJNS), Ch. Engr., Ewart Works, Link-Belt Co., Box 346, Indianapolis 6, Ind.
Weiss, Herbert Albert ('41) (NYB), Prof. Mech. Engrg., Clarkson College of Tech.; *for mail,* 34 Main St., Potsdam, N.Y.
Weiss, Herbert Klemm (J'42) (BAZH), Box 189, Aberdeen, Md.
Weiss, Joseph R. ('27; '36) (EOSLKZ), Cons. Engr., 51 E. 42nd St., New York 17, N.Y.
Weiss, Louis Theodore ('13; '25) (CJM), 772 Pacific St., Brooklyn 17, N.Y.
Weiss, Orin Andrew ('39) (LMSHWD), Cons. Engr., Box 19, Sta. C, New York 14, N.Y.

Weiss, Paul Albert Henry ('18; '25; '35) (CEF HKS), Plant Engr., Cent. Hudson Gas & Elec. Corp., South Rd., Poughkeepsie, N.Y.
Weissbach, Edward A. ('41) (CDLMZ), Supt. Equip., Campbell Soup Co., 100 Market St., Camden 1; *for mail*, 44 W. Cedar Ave., Merchantville, N.J.
Weisselberg, Arnold ('26; '35) (LKBSON), Cons. Engr., 277 Broadway, New York 7, N.Y.
Weithofer, Frank Warren (J'41) (NBEHAS), Design Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 2320-A N. 58th St., Milwaukee 10, Wis.
Weitzel, William F. ('40; '43) (DSKBHC), Matl. Handling Engr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore 24; *for mail*, 2907 Westfield Ave., Baltimore 14, Md.
Weitzmann, Earle Joseph ('32; '35) (BCJKNZ), Maint. Engr., Niacet Chemicals Div., U.S. Vanadium Corp., Pine Ave. & 47th St.; *for mail*, 631 Vanderbilt Ave., Niagara Falls, N.Y.
Welanetz, Ludolf F. (J'36) (ABH), Asst. Prof., Postgraduate Sch., Naval Acad., Annapolis, Md.
Welch, Albert E. ('20; '25), Mech. Engr., Charge of Maint. & Power, Am. Thread Co., Main St.; *for mail*, Box 122, Willimantic, Conn.
Welch, Mrs. Betty Slocombe ('36; '45) (ABK FHO), Firm of Mrs. Nicholas A. Welch, Aero. Engr., 21 Arapahoe Rd., West Hartford 7, Conn.
Welch, Charles Franklin (J'42) (YLCAMD), Technician, 4th Grade, Hdq. Co., 2502nd S.C.U., South Post, Ft. Myer, Va.
Welch, Chester William ('34) (LSDZJK), Indus. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail*, 23 Sevinor Rd., Marblehead, Mass.
Welch, Floyd Edward, Jr. (J'44) (MCNKJH), 3344½ Imperial High, Lynwood, Calif.
Welch, Leon C. ('13), Asst. Gen. Mgr., Stand. Oil Co. of Ind., 910 S. Michigan Ave., Chicago, Ill.
Welch, Lloyd B. (J'41) (MKS), Elec. & Mech. Engr., Bldg. 23, Eastman Kodak Co., Kodak Park, Rochester 13, N.Y.
Welch, Nicholas Anthony ('36; '45) (SOL), Mech. Engr., Am. Hardware Corp., 102 Washington St., New Britain; *for mail*, 21 Arapahoe Rd., West Hartford, Conn.
Welch, Philip John (J'38) (MCDLJB), Asst. to Indus. Engr., Am. Pulley Co., 4200 Wissahickon Ave., Philadelphia, Pa.
Welch, William, Jr. ('26; '37) (SFEK), Steam Power Engr., Long Island Ltg. Co., 250 Old Country Rd., Mineola, L.I., N.Y.
Welch, William Edward (J'44) (JRS), 4609 N. Camac St., Philadelphia 40, Pa.
Welch, William Pressley (J'36) (BN), Research Engr., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Welcker, William A., Jr. ('30; '36) (BHJ), Asst. Business Mgr., Battelle Memorial Inst., 505 King Ave., Columbus, Ohio.
Weld, Alfred Ollis ('13) (AKHE), Treas., Mech. Engr., George A. Weld & Co., 120 Milk St., Boston 9; *for mail*, 48 Winthrop St., Winchester, Mass.
Weld, Miss Lydia Gould ('15; '35) (Y), Retired; Box 51, Route 1, Carmel, Calif.
Weld, Philip Brownlie (J'43) (CMA), Sales Engr., Gen. Elec. Supply Corp., 127 E. Main St., Waterbury; *for mail*, Taft Sch., Watertown, Conn.
Weldon, Richard Laurence ('20; '25; '27) (AC), Pres., Bathurst Power & Paper Co., Ltd., Sun Life Bldg., Montreal, Que., Can.
Weldy, George Henry (J'43) (CDS), c/o Armour Leather Co., Sheboygan, Wis.
Weldy, Robert K. (J'37) (EACJYN), Head, Engrg. Dept., Fuel Injection Div., Ex-Cell-O Corp., 1200 Oakman Blvd., Detroit 6; *for mail*, 5092 Ivanhoe, Detroit 4, Mich.
Welford, P. G. ('14; '25; '35) (BCD), Sales Engr., Link-Belt Ltd., 791 Eastern Ave., Toronto, Ont., Can.
Welge, Harold B. ('41) (SHKFLC), Mech. Engr., Heat & Power Dept., Engrg. Div., Procter & Gamble Co., 311 M.A. & R. Bldg., Ivorydale 17; *for mail*, 4703 N. Edgewood Ave., Cincinnati 32, Ohio.
Welhart, Charles ('40) (CJLMN), Equip. Engr., Gen. Engrg. Dept., Owens-Ill. Glass Co.; *for mail*, 851 Washington Ave., Alton, Ill.
Welinder, Roy Anders (J'42), 560—78th St., Brooklyn, N.Y.
Wellauer, Edward Joseph ('35; '44) (NBJ), Supt. Research & Metal., Falk Corp., 3001 W. Canal St., Milwaukee 8; *for mail*, 8122 Warren Ave., Wauwatosa 13, Wis.
Wellech, Edmund Henry ('44) (NBCHO), Mgr., Mech. Research Dept., Corning Glass Works, Walnut St.; *for mail*, 75 W. 5th St., Corning, N.Y.
Wellenkamp, Paul ('39) (ACMJBO), Mgr., Exper. Mfg., Ranger Aircraft Engs. Div., Fairchild Eng. & Airplane Corp., Farmingdale, L.I., N.Y.; *for mail*, 752 Mountain Ave., Bound Brook, N.J.
Weller, Arthur Clarence (J'37) (SEFK), Engr., Tex. Co., 205 E. 42nd St., New York 17; *for mail*, Eton Lodge, Scarsdale, N.Y.
Weller, Arthur Stuart (J'45) (SEKBNH), Serv. Engr., Steam Div., Westinghouse Elec. Corp., 1 Montgomery St., San Francisco 4; *for mail*, 632 Jean St., Oakland 10, Calif.
Weller, LeRoy Ashton, Jr. (J'42) (MSHCK), 3rd Asst. Engr., Moore McCormack Lines, Inc., 5 Broadway, New York 4, N.Y.; *for mail*, 721 Euclid Ave., Upland, Calif.
Wellhofer, Ernest S. ('40) (DMJA), Ch. Rope Engr., Am. Chain & Cable Co., Inc., 81 E. Ross St., Wilkes-Barre, Pa.
Welling, Lindsay Howe ('16; '24; '35) (MCN JY), Gen. Mgr., Horma Mfg. Corp., 29-05—40th Rd., Long Island City; *for mail*, 3 Norwood Rd., Scarsdale, N.Y.
Wellington, C. Oliver (A'21) (CMO), Sr. Partner, Scovell, Wellington & Co., 111 Broadway, New York 6, N.Y.
Wellington, Q. W. ('40) (FKS), 208 N. Maplewood, Peoria 5, Ill.
Wellman, George A. (J'41) (BDL), Lt., U.S.N.R., Box 619A, Route 2, Bremerton, Wash.
Wellman, Samuel Knowlton ('41) (ACJ), Pres., Ch. Engr., S. K. Wellman Co., 1374 E. 51st St., Cleveland 3, Ohio.
Wellner, Eric Fred ('38; '42) (BKS), Field Engr., Westinghouse Elec. Corp., 10 High St., Boston 10; *for mail*, 6 Hobson St., Roslindale 31, Mass.
Wellons, Frank W. (J'40) (AJ), Aircraft Engr., SKF Industries Inc., Front St. & Erie Ave.; *for mail*, Hamilton Court, 39th & Chestnut, Philadelphia, Pa.
Wells, Albert William ('18; '32; '35) (R), Retired; 1093 Avoca Ave., Pasadena 2, Calif.
Wells, Arthur S. ('23) (SLFZKD), Plant Engr., Mead Corp., Main St.; *for mail*, 710 Yadkin St., Kingsport, Tenn.
Wells, Burling D. ('19; '27; '35) (BCM), Indus. Engr., Mallory Hat Co.; *for mail*, 89 Lake Ave., Danbury, Conn.
Wells, Cecil Gray ('30; '35) (CM), V.P., Natl. Tank Co., Box 1710, Tulsa 1, Okla.
Wells, Edward H. (J'33) (RKA), Ch. Engr., Trans. Dept., Johns-Manville Corp., 22 E. 40th St., New York 16, N.Y.
Wells, Edward Lathrop (J'43) (ABEOMY), Pvt., Army Air Forces, 343rd Fighter Sqd., 55th Fighter Gp., A.P.O. 374, c/o Postmaster, New York, N.Y.; *for mail*, 15 Union St., Randolph, Mass.
Wells, Edwin S., Jr. ('42) (SKER), Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Wells, Gorden Samuel (J'44) (BNCM), Devel. Engr., Stow Mfg. Co., 445 State St.; *for mail*, 78 Conklin Ave., Binghamton, N.Y.
Wells, Henry Stuart, Jr. (J'37) (ZNRTSA), Engrg. Lab. Asst., Friez Instrument Div., Bendix Aviation Corp., Baltimore; *for mail*, 1926 Mt. Royal Terrace, Baltimore 17, Md.
Wells, Herbert ('27; '38) (SFKE), Sales Engr., Coon DeVisser Co., 2051 W. Lafayette Blvd., Detroit 16; *for mail*, 5068 Kensington Rd., Detroit 24, Mich.
Wells, James Earl (J'43) (ACYZ), Devel. Engr., Goodyear Aircraft Corp., Massillon Rd., Akron; *for mail*, 479 E. Buchtel Ave., Akron 4, Ohio.
Wells, James Milton ('26; '45) (CHM), Sales Engr., Motch & Merryweather Mchy. Co., 2842 W. Grand Blvd., Detroit 2, Mich.
Wells, John Arthur (J'44) (SEBHRL), Sales & Application Engr., Westinghouse Elec. Corp., 1 Montgomery St., San Francisco, Calif.
Wells, Richard Carter (J'41) (MCA), Planner & Scheduler, Planning & Matl. Dept., Glenn L. Martin Co., Baltimore 3; *for mail*, 3301 W. Strathmore Ave., Baltimore 15, Md.
Wells, Robert F. (J'40) (BCM), c/o Engrg. Drawing Office, Canadian Vickers Ltd., Maisonneuve, Montreal, Que., Can.
Wells, Robert Henri ('42), Pres., Corrigan, Osburne & Wells, 60 E. 42nd St., New York, N.Y.
Wells, Robert Lynn (J'40) (BCA), Asst. Project Engr., Engrg. Dept., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester, Pa.
Wells, Walter Farrington ('14), Retired; 458 Washington Ave., Brooklyn 5, N.Y.
Wells, William Daemish ('42) (HMRC), Asst. Plant Supt., Tedlee Chem. Co., 115 Dobbin St., Brooklyn; *for mail*, 189 Weaver St., Larchmont, N.Y.
Welly, Robert Burton (J'41) (AEK), Ensign, U.S.N.R.; 13415 Lauder Ave., Detroit, Mich.
Welp, Robert Russell (J'41) (ACMYO), Ensign, U.S.N.R., Carrier Aircraft Serv. Unit 1, F.P.O., San Francisco, Calif.
Welsch, Arthur Harry (J'42) (CMA), 2nd Lt., Army Air Forces, Maint. Engrg. Officer, Army Air Field, Box 517, Kingman, Ariz.; *for mail*, 814 S. Springfield Ave., Chicago 24, Ill.
Welsh, Ernest J. (J'40) (CLJSZ), Plant Engr., Louisville Div., Natl. Distillers Products Corp., 7th & Bernheim Lane, Louisville, Ky.
Welsh, Harvey William (J'45), Specification Engr., Wright Aero. Corp., 132 Beckwith Ave., Paterson, N.J.
Welsh, Jack Reed (J'40) (ACM), Lt., U.S.N.R., Bur. of Aero., Navy Dept., c/o Pratt & Whitney Div., Niles-Bement-Pond Co.; *for mail*, 4801 Roanoke Pkwy., Kansas City, Mo.
Welsh, John Vernon ('45) (KCHLY), Sales Engr., Owens-Corning Fiberglas Corp., 3445 W. 8th St., Los Angeles 5; *for mail*, 3882 Olympiad Dr., Los Angeles 43, Calif.
Welsh, Milford Gordon ('45) (CHSW), Mech. Engr., Newport News Waterworks Comm., City Hall, Newport News; *for mail*, R.F.D., Denbigh, Va.
Welsh, Robert W. (J'35) (OSBLKC), Design Engr., Monsanto Chem. Co., 1700 S. 2nd St.; *for mail*, 2329-A Michigan Ave., St. Louis 4, Mo.
Welsh, William Howard ('42) (BEFKS), M.M., Power Plant, N.Y. Naval Shipyard, Brooklyn; *for mail*, 122 Willoughby Ave., Brooklyn 5, N.Y.
Welshman, Hugh ('44) (SLBJON), Mgr., Indus. Piping Div., Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Welter, Gustave ('22; '35) (SFKCJE), V.P., Ch. Engr., Bigelow Co., Box 706, New Haven 3; *for mail*, 670 Winthrop Ave., New Haven 11, Conn.
Welti, Robert Walter (J'43) (ZLFA), Radio Technician, 2nd Class, Naval Sta. Box 24, Navy 3149, F.P.O., San Francisco, Calif.
Weltman, Albert Bernard (J'45) (COMJNY), Engr. Trainee, Wagner Elec. Corp., 6400 Plymouth St., St. Louis, Mo.; *for mail*, 101 N. 10th St., East St. Louis, Ill.
Wempe, Robert Herman (J'44) (OCA), Engr., Carrier Corp., 20 N. Wacker Dr., Chicago 6, Ill.
Wemple, Merritt Clark (J'43) (CHMNY), Pres., Gen. Mgr., Pump Engrg. Co. Inc., Rm. 519, 1331—3rd Ave. Bldg., Seattle 1, Wash.
Wendel, D. P. ('34; '35) (A), Ch., Boiler Opera., Buffalo Gen. Elec. Co., C. R. Huntley Sta. 2, Elec. Bldg.; *for mail*, 3442 Main St., Buffalo, N.Y.
Wendell, Dwight Marston (J'45) (CMYL), Technician, 5th Grade, Corps of Engrs., U.S.A., (assigned to Carbide & Carbon Chemicals Corp. as Tech. Engr.), Spec. Engrg. Det., Bks. Area, Oak Ridge, Tenn.; *for mail*, Axtell, Neb.
Wendes, John C. H. ('28) (BELNY), Ch. Engr., Naugatuck Chem. & Synthetic Rubber Divs., U.S. Rubber Co., 1230 Ave. of the Americas, New York 20, N.Y.; *for mail*, 338 Marlboro Rd., Englewood, N.J.
Wendschuh, O. H. (J'46) (FHS), Mech. Engr., Ft. Howard Paper Co.; *for mail*, 1321 S. Broadway, Green Bay, Wis.
Wendt, Edgar F. ('21), Pres., Buffalo Forge Co., 490 Broadway, Buffalo, N.Y.
Wendt, Leland Arthur ('35; '45) (ESRF), Sr. Field Test Engr., Shell Oil Co., Inc., Wood River, Ill.; *for mail*, 8555 Church Rd., St. Louis 15, Mo.
Wendt, William Rudolph, Jr. (J'45) (OKRS), Instr. in Mech. Engrg., Univ. of Wis., 1513 University Ave., Madison 6, Wis.
Wene, Alvin William (J'42) (SKLBOY), Draftsman, Engrg. Dept., Rohm & Haas Co., Inc., Box 219, Bristol; *for mail*, 5024 C St., Philadelphia 20, Pa.
Wener, Nathan Leonard (J'41) (ABM), Lt., U.S.N.R., YDG-6, F.P.O., San Francisco, Calif.; *for mail*, 125 W. Lee St., Louisville 8, Ky.
Wenig, Harold George (J'45) (ANJBMZ), Pvt., Army Air Forces, 4000th Base Unit, Wright Field, Dayton, Ohio; *for mail*, 165 Broome St., New York 2, N.Y.
Wenner, James John ('45) (OSCKMN), 4228—81st St., Cincinnati, Ohio.
Wentworth, E. Francis ('22; '30), Sales Engr., N.Y. Air Brake Co., 420 Lexington Ave., New York, N.Y.; *for mail*, 36 Washington Terrace, East Orange, N.J.
Wentworth, Harry Thomas ('41) (CMOY), Partner, Harworth Engrg., 360 Valley; *for mail*, 391 Turrell Ave., South Orange, N.J.
Wentworth, Reginald Andrew ('11; '13) (CDM), Mgmt. Counsel, 165 Crescent Ave., Plainfield, N.J.
Wentz, Heidel Howett ('26) (CDEFMR), Expediter, Worthington Pump & Mchy. Corp., Roberts Ave., Buffalo 6, N.Y.
Wenzel, Alfred C. ('27; '40) (FSZHCO), Ch. Contract Engr., Republic Flow Meters Co., 2240 Diversey Pkwy., Chicago 47; *for mail*, 6619 N. Rockwell St., Chicago 45, Ill.
Werft, August Rudolph ('45) (DFHJNO), Mech. Engr., H. C. Frick Coke Co., Fayette Title & Trust Bldg.; *for mail*, 166 S. Mt. Vernon Ave., Uniontown, Pa.
Werger, John Richard (J'42) (ABCGME), Prod. Liaison Engr., Wright Aero. Corp., Paterson 3; *for mail*, 284 Bluff Rd., Palisade, N.J.
Werley, Harlan R. (J'45) (DLBONH), Plant Layout & Design, Campbell Soup Co., 101 Market St., Camden; *for mail*, 263 Kings Highway W., Haddonfield, N.J.
Werner, Frederick William (J'33) (NMDBCA), Finishing Mill Engr., Scott Paper Co., Front & Market Sts., Chester; *for mail*, Mt. Alverno Rd., R.D. 2, Media, Pa.
Werner, George Henry (J'41) (BAHM), Engr., Gen. Elec. Co., Thompson Rd., Syracuse; *for mail*, S. Main St., North Syracuse, N.Y.
Werner, Herbert Blake (J'45) (ES), Mech. Engr., Design, Ord., Army Serv. Forces, Aberdeen Proving Ground; *for mail*, 18 Aberdeen Ave., Aberdeen, Md.
Werner Philip ('32; '40) (NMCBJ), Mech. Engr., Plant Engrg. Dept., Babcock & Wilcox Co., Barberton; *for mail*, 566 Roslyn Ave., Akron 2, Ohio.
Werner, Richard ('25; '35) (EFS), Cons. Engr., W. T. Waggoner Bldg., Ft. Worth, Tex.
Werner, Roy Walter (J'44) (JWF), 21-12—31st St., Long Island City 5, N.Y.
Werngren, Martin Stuart (J'37) (SBKCHJ), Maj., Ord. Dept., U.S.A., Military Trng. Div., Hdq., Army Serv. Forces, Washington, D.C.

Wernick, Nathaniel Kenneth (J'43) (OCMNJ), Assoc. Ch. Engr., Associated Electronics Co., 132 Nassau St., New York 7; *for mail*, 438 Amboy St., Brooklyn 12, N.Y.
Werst, Harry Kenneth (J'26) (CMTDLS), Factory Mgr., Williamson-Dickie Mfg. Co., 509 Vickery Blvd., Ft. Worth 1; *for mail*, 3105 Westcliff Rd. W., Ft. Worth, Tex.
Wert, Edwin Arthur ('31), Gen. Mgr., S.W. Region, Kennedy-Van Saun Mfg. & Eng. Corp., 1021—9th & Hill Bldg., 315 W. 9th St., Los Angeles 15, Calif.
Wertman, David (J'36) (HBNMZ), Sr. Engr., Bur. of Ord., Navy Dept., Washington, D.C.; *for mail*, 2703 Webster St., Mt. Rainer, Md.
Wery, Albert George ('41) (BCHLN), Project Engr., Whipple & Rattray Co., 114 Liberty St., New York, N.Y.; *for mail*, 722 Larch Ave., Teaneck, N.J.
Weschler, George A. ('42), Broadview Farm, Warrenton, Va.
Weschler, Maurice Elmo ('21; '31) (OSEKFN), Head, Dept. of Mech. Engrg., Catholic Univ. of Am., Washingtou 17, D.C.; *for mail*, 6510 Summit Ave., Chevy Chase 15, Md.
Weschler, William Sampson (J'43) (BJMNYA), Mech. Design & Devel. Engr., RCA Victor Div., Radio Corp. of Am., Front & Cooper Sts.; *for mail*, 1530 Baird Ave., Camden, N.J.
Wescott, Blaine B. ('29) (JBH), Ch., Matls. & Prod. Chemistry Div., Gulf Research & Devel. Co., P.O. Drawer 2038, Pittsburgh 30, Pa.
Weselik, Leslie Joseph ('45) (LNOCZK), Devel. Engr., White Cap Co., 1819 N. Major Ave., Chicago; *for mail*, 844 E. 87th Pl., Chicago 19, Ill.
Wesemann, Edwin J. (J'41) (SCMO), Mech. Design Engr., Ebasco Services, Inc., 2 Rector St., New York 6; *for mail*, 59 Union St., Valley Stream, N.Y.
Weske, John Robert ('34; '35) (BASHKE), Prof. Aero. Engrg., Rensselaer Poly. Inst., Troy, N.Y.
Wesson, Charles Macon (H'41), Maj. Gen., 3114 "O" St., N.W., Washington, D.C.
Wesson, Paul Bancroft ('12) (BDKL), Mech. Engr., Eastman Kodak Co., Kodak Park; *for mail*, 121 Gorsline St., Rochester 13, N.Y.
Wesstrom, David Boman (J'28) (BJS), Process Engr., E. I. du Pont de Nemours & Co., Wilmington; *for mail*, 311 W. 34th St., Wilmington 218, Del.
West, Arthur ('02; '18; F'37) (EFSN), *Vice-President*, '07-'09; Retired; The Huntington Hotel, Pasadena 15, Calif.
West, Charles Ernest (J'40) (AZCBMO), Ensign, U.S.N.R., Naval Research Lab.; *for mail*, 3817—2nd St., S.E., Washington 20, D.C.
West, Donald Parker (J'39) (JMNBCO), Asst. Plant Engr., Simonds Arbasive Co., Tacony & Fraley Sts., Philadelphia 37; *for mail*, 22 N. Old York Rd., Willow Grove, Pa.
West, Frank Russell ('40) (BJM), V.P., Charge of Engrg., Excel Fdy. & Mch. Co., Inc., Fall River, Mass.; *for mail*, Box 164, San Andreas, Calif.
West, Henry Ivey ('41; '43) (SFC), Ch. Power Plant Engr., Univ. of N.C.; *for mail*, 503 Cameron Ave., Chapel Hill, N.C.
West, Howard F. ('38) (CEOJSF), Ch. Mech. Engr., Mfg. Dept., Stand. Oil Co. of Ohio, Midland Bldg., Cleveland 15, Ohio.
West, James D. (J'41) (JKM), 4734 Dorchester Ave., Chicago, Ill.
West, James Edward, Jr. (J'44), Mech. Engr., Bearing Design, SKF Industries, Inc., Front & Erie Aves., Philadelphia; *for mail*, 3307 Longshore Ave., Philadelphia 24, Pa.
West, John P. (J'43) (SBKMEN), Lt. (j.g.), U.S.N.R., Engrg. Exper. Sta., Annapolis, Md.; *for mail*, 955 Lexington Ave., New York 21, N.Y.
West, John Tunnell, Jr. ('41) (SOTELF), Mech. Engr., Lehigh Engrg. Co., 1150 Mauch Chunk Rd., Bethlehem; *for mail*, R.D. 1, Barto, Pa.
West, John W., Jr. (J'19), Secy., Commercial Sec., American Gas Association, 420 Lexington Ave., New York, N.Y.
West, Louis Lester ('36) (CNMJB), Ch. Engr., Gries Reproducer Corp., 780 E. 133rd St., New York 54; *for mail*, 270 Ocean Ave., Seaford, L.I., N.Y.
West, Robert Ferguson (J'43) (MNOWGB), Mech. Engr., Design & Devel., T. C. Wheaton Co., 212 G St., Millville, N.J.
West, Sidney Smith (J'42) (BCHLMN), Designing Engr., Hauselmann Engrg. Corp., 4757 Broadway; *for mail*, 4755 Raleigh St., Denver 12, Colo.
West, Woodrow Wilson (J'43) (DNBH), Sales & Serv. Engr., Pa. Crusher Co., Liberty Trust Bldg., Philadelphia; *for mail*, 1502 Hickory Lane, Rosemont, Pa.
Westaway, Clarence Royal ('44) (HSLKEA), Centrifugal Pump Specialist, Ingersoll-Rand Co., 285 Columbus Ave.; *for mail*, 179 Malboro St., Boston 16, Mass.
Westcott, Harry R. ('16; F'36) (CSDLMO), *Manager*, '31-'34, *Vice-President*, '34-'36 and '37; Pres., Westcott & Mapes, Inc., 109 Church St., New Haven 5, Conn.
Westendorf, Charles Leslie ('39; '45) (CFLS), Power Design Engr., Engrg. Dept., E. I. du Pont de Nemours & Co., Wilmington, Del.
Westerberg, C. Frederick ('22), Ch. Draftsman, Stanley Works, Seaview Ave.; *for mail*, 360 Gurdon St., Bridgeport 6, Conn.
Westerdahl, Axel ('27) (LWSD), Engr., W.Va. Pulp & Paper Co., 230 Park Ave., New York 17, N.Y.; *for mail*, 57 Woodland Rd., Bloomfield, N.J.
Westergaard, Harald Malcolm ('39) (B), Dean, Graduate Sch. of Engrg., Gordon McKay Prof. Civ. Engrg., Harvard Univ., 212 Pierce Hall, Cambridge 38, Mass.
Westergaard, Viggo ('45) (EHJ), Dept. of Pub. Works, 125 Worth St., New York 13, N.Y.
Westerlund, George Erhard ('37; '44) (OSDCEF), Mech. Supvr., Mech. Engrg. & Power Plant Design, Burns & Roe Inc., 233 Broadway, New York 7; *for mail*, 4313—9th Ave., Brooklyn 32, N.Y.
Westermaier, Francis Victor ('31) (CFLMO), V.P., Charge of Engrg., Lloyd Associated Inc., 1530 Lewis Tower, Philadelphia 2, Pa.; *for mail*, 400 King's Highway E., Haddonfield, N.J.
Westermann, John Leonard, Jr. (J'43) (SJKF), Turbine Engr., Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Westervelt, W. I. ('15) (AFJ), Brig. Gen., U.S.A., Retired, Ross & Co., Engrs. & Consultants, 441 Lexington Ave., New York 17; *for mail*, Vanderbilt Hotel, Park Ave. at 34th St., New York 16, N.Y.
Westin, Lloyd Judson (J'43) (SOEFKN), Engr., Power Plant Work, Day & Zimmermann Inc., Packard Bldg., Philadelphia 2; *for mail*, 728 E. Dorset St., Philadelphia 19, Pa.
Westin, Warren Charles (J'42) (MCD), Asst. Indus. Engr., Fed. Tel. & Radio Corp., 591 Broad St., Newark 2, N.J.
Westlake, William Gladden ('45) (MLDK), Asst. Mech. Engr., Solvay Process Co., Hopewell, Va.
Weston, Everett Hawes (J'39) (RNJYFB), Leading Loco. & Car Draftsman, Chicago & Northwest. Ry., 4200 W. Kinzie St., Chicago 24; *for mail*, 2060 Ridge Ave., Evanston, Ill.
Weston, Leonard Arthur ('44), Adviser to Am Consulate Gen. at Batavia, Java, Dept. of State, Washington, D.C.; *for mail*, 59 Colton, Springfield, Mass.
Wetherbee, Arthur Everett (J'40) (ABNKZC), Flight Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., 400 Main St., East Hartford 8; *for mail*, 27 Basswood St., Newington, Conn.
Wetherill, Fred V. ('19; '30) (JKL), Box 43, Rosemont, Pa.
Wetherill, Robert, Jr. ('19) (FJ), Retired; Cleveland Athletic Club, Cleveland 15, Ohio.
Wetter, Pierce Trowbridge ('27; '30; '35) (MAN), Pres., Wetter & Williams Corp., 416—88th St., Brooklyn 9; *for mail*, 24 Washington Sq., New York 11, N.Y.
Wetterstrom, Edwin (J'44) (ABHJNS), Mechanics Engr., 5518 W. Huron St., Chicago 44, Ill.
Wetz, Leonard R. (J'40) (CEK), R.R. 2, Morrow, Ohio.
Wetzel, Irwin Theodore (J'40) (KEFBZ), Lecturer, Dept. of Mech. Engrg., Tech. Inst., Northwest. Univ., Evanston, Ill.
Wetzel, John J. (J'36) (CBEFAH), Gear Engr., Dodge Div., Chrysler Corp., 7900 Jos. Campau St., Hamtramck; *for mail*, 4660 Bedford Rd., Detroit 24, Mich.
Wetzel, Theodore Alfred ('41) (BNZLJC), Devel. Research Engr., Kearney & Trecker Corp., 6784 W. National Ave., Milwaukee 14; *for mail*, 2027 N. 47th St., Milwaukee 8, Wis.
Wexler, Meyer (J'30) (HKS), Sales Engr., Foster Wheeler Corp., 165 Broadway, New York, N.Y.; *for mail*, 559 W. Inman Ave., Rahway, N.J.
Weyers, Curtis Robert (J'41) (EKS), Field Serv. Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6; *for mail*, 190 Beach 43rd St., Far Rockaway, L.I., N.Y.
Weygandt, Arthur Scott ('45), Tech. Dir., Industrias Quimicas Argentinas "Duperial," Paseo Colon 285, Buenos Aires, Argentina, S.A.
Weygant, Robert M. (J'39) (CDM), Sales Roadman, Engr., Am. Can Co., 14th St. & Sheridan Rd., North Chicago; *for mail*, 1018 N. Sheridan Rd., Waukegan, Ill.
Weyher, Theodore A. ('40) (C), Col., Am. Legation, Berne, Switzerland.
Weymouth, Thomas Rote ('10; F'36) (EFBGOH), *Vice-President*, '30-'32; Cons. Engr., 930 Park Ave., New York 28, N.Y.
Whalen, Francis (J'39) (CJODAM), Engr., Humble Oil & Refining Co., Baytown; *for mail*, 3114 University Blvd., Houston 5, Tex.
Whaley, Frank Halsey (J'38) (MDC), Engr., Am. Sugar Refining Co., 49 S. 2nd St., Brooklyn; *for mail*, Main St., East Moriches, N.Y.
Whaley, Fred C. (J'41), Ensign, 151 E. 53rd St., Seattle, Wash.
Whallon, James Elvin ('27; '34; '35) (S), Gen. Supt., Mech. Constr. Bur., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 3, N.Y.
Wharton, H. Jerome (J'38) (EFCHOS), Engr., Gulf Oil Corp., 17 Battery Pl., New York; *for mail*, 82 N. Chatsworth Ave., Larchmont, N.Y.
Wheat, James Clements ('44) Ch. Crane Engr., Indus. Brownhoist Corp., Bay City, Mich.
Wheat, Oscar G. ('30; '39) (CDMAFS), Mgr., 27th St. Plant, Armstrong Cork Co., Beaver Falls, Pa.
Wheatley, John G. ('29), Supt., Engrg. Dept., Eagle, Globe & Royal Indemnity Co., 150 William St., New York, N.Y.
Wheaton, Henry Clay ('45), Mgr., N.Y. Office, Bailey Meter Co., 30 Church St., New York 7, N.Y.
Wheaton, William E. ('21; '35) (CGM), Works Mgr., Walter Scott & Co., South Ave.; *for mail*, 818 Webster Pl., Plainfield, N.J.
Wheeler, Albert Henry ('44) (SR), Steam Serv. Supvr., Westinghouse Elec. Corp., 3001 Walnut St., Philadelphia 4; *for mail*, 646—15th Ave., Prospect Park, Pa.
Wheeler, Brian ('43) (R), Asst. Mech. Engr., Lima Loco. Works, Inc., Lima, Ohio.
Wheeler, Burr ('18) (SRJEC), Exec. V.P., Chile Exploration Co., 25 Broadway, New York 4, N.Y.
Wheeler, Chester C. (J'40) (ANS), Mech. Engr. P-3, Naval Ord. Testing Sta., 1080 E. Green St., Pasadena 1; *for mail*, 2083 Pepper Dr., Altadena, Calif.
Wheeler, Frank Gideon ('41) (BJM), Miehle Ptg. Press & Mfg. Co., 14th St. & S. Damen Ave., Chicago; *for mail*, R.R. 1, Hinsdale, Ill.
Wheeler, Fred Aston (A'45) (CKEFNB), 5000 Finley Ave., Los Angeles 27, Calif.
Wheeler, Gardner Edmund, Jr. (J'36) (DJMCLN), Sales Engr., G. E. Wheeler Co., 453 Chapel St., New Haven; *for mail*, 64 Marvel Rd., New Haven 15, Conn.
Wheeler, Hobart W. R. ('30) (K), Sales & Engrg. Consultant, Griscom-Russell Co., 285 Madison Ave., New York 17, N.Y.
Wheeler, James Walker (J'45) (ZACMBO), Product Engr., Sperry Gyroscope Co., Great Neck; *for mail*, 8 Highland Pl., Sea Cliff, N.Y.
Wheeler, John Edwin (J'42) (EASFZ), Devel. Engrg., Superior Eng. Div., Natl. Supply Co., Springfield, Ohio.
Wheeler, Joseph B. ('38) (BFS), Asst. Ch. Engr., Steam Power Plants Dept., Cahokia Power Plant, Union Elec. Co. of Ill., Monsanto, Ill.
Wheeler, Louis James ('28) (CDF), Gen. Supt., Lone Star Cement Corp., 1217—1st Natl. Bank Bldg., Dallas 1, Tex.
Wheeler, Richard Carlton (J'44) (CSLOFR), Maint. Supt., Calvert Distilling Co., Relay; *for mail*, 5619 Oakland Rd., Baltimore 27, Md.
Wheeler, Walter George (J'37) (NABJ), 1437 Midvale Ave., West Los Angeles 24, Calif.
Wheeler, William Alfred, Jr. (J'45) (NBM), Mch. Designer, Bullard Co., 286 Canfield Ave., Bridgeport 2; *for mail*, 90 Aldine Ave., Bridgeport 4, Conn.
Wheelock, Bennett Robinson, Jr. ('38; '45) (SKFDLO), Mech. Engr., Gilbert Associates, Inc., 412 Washington St., Reading; *for mail*, 204 Harvard Blvd., Lincoln Park, Pa.
Whelan, Paul R. ('39) (EFS), 708 W. Bethune St., Detroit 2, Mich.
Whelan, Roderick Joseph ('20; '35) (CJM), Pres., Ohio Nut & Bolt Co., 600 Front St., Berea, Ohio.
Whelchel, Cornelius Cooper ('39) (SCKFDB), Ch. Mech. Engr., Buffalo Niagara Elec. Corp., 535 Washington St., Buffalo 3, N.Y.
Wherritt, Charles Reed ('46) (BCT), Mech. Engr., Celanese Corp. of Am.; *for mail*, 532 Cumberland St., Cumberland, Md.
Whetzel, J. C. ('44), 1311 Murdoch Rd., Pittsburgh, Pa.
Whipp, Wendell E. ('30) (BCM), Pres., Monarch Mch. Tool Co., Sidney, Ohio.
Whipple, Geo. F. (A'17), Educational Dir., Whipple Tech. Libraries, Boston, & Thomas Publ. Co., 461 E. 8th Ave., New York, N.Y.; *for mail*, 50 Beacon St., Boston 8, Mass.
Whipple, James Halley, Jr. ('46) (AER), Supt. Diesel Equip., Denver & Rio Grande West. R.R. Co., 1531 Stout St., Denver, Colo.
Whipple, Richard Starbird (J'41) (ADM), 1st Lt., Army Air Forces, Meteorology Br., 19th Weather Sqd., A.P.O. 622, c/o Postmaster, Miami, Fla.; *for mail*, 30 Madison Ave., Newtonville 60, Mass.
Whipple, Thomas T. ('29; '41) (EKMB), Project Engr., Lummus Co., 420 Lexington Ave., New York; *for mail*, 830 Bronx River Rd., Bronxville 8, N.Y.
Whipple, William ('05; '16) (ESFKH), Prof. Steam Engrg., La. State Univ., University Sta., Baton Rouge 3, La.
Whisler, Forbes Devon (J'35) (ESRAFO), Exper. Engr., Baldwin Loco. Works, Paschall P.O.; *for mail*, 4527 Walnut St., Philadelphia 42, Pa.
Whistler, Arthur McLeod ('44) (EFHKLZ), Process Engr., C. F. Braun & Co., 1000 S. Fremont Ave., Alhambra, Calif.
Whistler, Charles Clayton, Jr. (J'45) (CROAMY), Layout Engr., Frigidaire Div., Gen. Motors Corp., Moraine City; *for mail*, 206 W. Hudson Ave., Dayton 5, Ohio.
Whitacre, Verne L. ('43) (KLYF), Insulation Mgr., N.Y. Power Products Dept., Johns-Manville Sales Corp., 22 E. 40th St., New York 16, N.Y.; *for mail*, 275 Engle St., Englewood, N.J.
Whitaker, Charles H. (J'23) (AGKZ), Oven Engr., Crown Cork & Seal Co., Box 1837, 4425 Eastern Ave., Baltimore 3, Md.
Whitaker, Ebenezer ('23) (CES), Ch. Engr., N.Y. Athletic Club, 180 W. 59th St., New York, N.Y.; *for mail*, 389 Lincoln Ave., Cliffside Park 10, N.J.
Whitaker, Harry Emerson ('18; '35; F'43) (AEF), V.P., Ch. Engr., Dir., Ford, Bacon & Davis, Inc., 39 Broadway, New York 6, N.Y.

Whitaker, John Albert ('19; '24; '35) (AB), Designing Engr., Douglas Aircraft Co., Santa Monica; *for mail*, 5430 Auckland Ave., North Hollywood, Calif.
Whitaker, Randall Judson ('30; '45) (CYZNJM), Supervisory Inspr., Quality Control, Naval Torpedo Sta.; *for mail*, 13 Pell St., Newport, R.I.
Whitaker, U. A. ('24; '36) (ACJMNZ), Pres., Aircraft-Mar. Products Inc., 1523 N. 4th St., Harrisburg, Pa.
Whitaker, Will Alton ('33; '35) (LBJD), Asst. Head, Engrg. Dept., La. Div., Stand. Oil Co. of N.J., Box 551, Baton Rouge 1; *for mail*, 1167 Main St., Baton Rouge 8, La.
Whitcomb, Adrian H. (J'39) (EFR), 84 F. George Legare Homes, Navy Yard 54, S.C.
Whitcomb, Charles F., Jr. (J'34) (ES), 10713 N. Libya St., Portland 3, Ore.
White, A. O., Jr. (J'46), Lecturer, Ga. Sch. of Tech.; *for mail*, 310 Ponce de Leon Ave., N.E., Atlanta, Ga.
White, Albert Easton ('23) (JSO), *Manager*, '42-'45; Prof. Metal. Engrg., Dir. of Engrg. Research, Univ. of Mich., Ann Arbor, Mich.
White, Albert Francis ('18) (DFO), Sales Engr., John Inglis Co., 14 Strachan Ave.; *for mail*, 130 Glendale Ave., Toronto, Ont., Can.
White, Albert Outram ('31; '43) (SERN), Design Engr., Turbine-Generator Eng. Div., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail*, 302 Root Ave., Schenectady 2, N.Y.
White, Alden D. ('34; '40) (CES), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail*, Box 305, Hanover, Mass.
White, Bradford C. (J'38) (SBKH), Engr., Process Div., Stone & Webster Engrg. Corp., 49 Federal St., Boston 7; *for mail*, 3 Hancock Ave., Lexington 73, Mass.
White, Britton ('25; '35; '35) (CM), Indus. Engr., 519 Security Bldg., Denver 2, Colo.
White, Clarence John (J'45) (EFJYWL), Specialties Div. Rep., Natl. Tube Co., 820 Travis, Houston 2, Tex.
White, Edward W. (J'41) (JKP), Gen. Del., Archer City, Tex.
White, Ellis Edmond ('41; '42) (ACM), Tech. Dir., Trans-Pac. Corp., 1026 Pacific Mutual Bldg., Los Angeles 14, Calif.
White, Francis Everett (J'42) (SM), 127 Exchange St., Emporia, Kan.
White, Frank O. ('20) (FLW), Fraser Cos. Ltd., Edmundston, N.B., Can.
White, Harold E. ('25; '35) (FKS), Engr., Cons. Sec., Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail*, 172 High St., Reading, Mass.
White, Harrison Gates ('38) (OSTHEF), Cons. Engr., 9 Andrew St., Springfield 9; *for mail*, 182 Sumner Ave., Springfield 8, Mass.
White, Henry Packard (J'34) (ABNOZ), Ch. Engr., H. P. White Co., Box 1852, 1958 E. 90th St., Cleveland 6, Ohio.
White, Herbert Judson ('09) (BCHJMN), Mgr., White & Son, 25 Stuart St., Lynbrook, L.I., N.Y.
White, Ira Morgan ('41) (HNCB), Ch. Engr., Pelton Water Wheel Co., 2929—19th St., San Francisco, Calif.
White, J. H. (J'19) (EFS), Gen. Supt., Fla. Power & Light Co., Miami, Fla.
White, J. W. Huyler ('42) (SFCLMD), Property Inspr., Prudential Ins. Co. of Am., 763 Broad St., Newark; *for mail*, 32 Peck Ave., Newark 7, N.J.
White, James A. ('00; A'18) (NDMTCO), Cons. Engr., 234 Locust St., Lockport, N.Y.
White, James C. ('25; '37) (CTYD), Pres., Gen. Mgr., Tenn. Eastman Corp.; *for mail*, 1214 Linville St., Kingsport, Tenn.
White, James Joseph, Jr. ('41), V.P., McArdle & Cooney, Inc., 519 Arch St., Philadelphia; *for mail*, 3 Green Hill Lane, Overbrook, Pa.
White, John Culbertson ('06; '11) (EFKNOW), Steam & Mech. Engr., 550 State St., Madison 3, Wis.
White, John Ernest (J'45) (JN), Design Draftsman, De Laval Steam Turbine Co., Nottingham Way, Trenton 2; *for mail*, 247 Atlantic Ave., Trenton 9, N.J.
White, John Rigsby ('31; '41) (CLEFS), V.P., Dir., Imperial Oil Ltd., 56 Church St., Toronto 1, Ont., Can.
White, Jud E. ('34; '43) (ZLMC), Indus. Engr., Taylor Instrument Co., 95 Ames St.; *for mail*, 80 Meadow Dr., Rochester 10, N.Y.
White, Karl Karey, Jr. (J'35) (CJKLZB), Zone Maint. Supvr., Philadelphia Refinery, Gulf Oil Corp., Box 7408, Philadelphia; *for mail*, 59 Price Ave., Lansdowne, Pa.
White, Kenneth Hastings (J'40) (MJDCSH), Asst. Prof. Mech. Engrg., Rensselaer Poly. Inst., Troy, N.Y.
White, Merton G. ('06; '21), Field Engr., Marlin Rockwell Corp., 132 E. Court St., Cincinnati 2; *for mail*, 2324 Park Ave., Cincinnati 6, Ohio.
White, Paul A. ('31; '41) (EM), Indus. Sales Engr., Socony-Vacuum Oil Co., 907 S. 1st St.; *for mail*, 1314 E. Kensington Blvd., Milwaukee, Wis.
White, Philip Stuart ('40) (CMWDJY), Dir. of Installation, George S. May Co., 122 E. 42nd St., New York 17, N.Y.
White, Raymond E. ('20) (ACW), R.F.D. 1, Box 198, Encinitas, Calif.
White, Richard E. (J'38) (CMP), 112 Crestwood Dr., Houston 7, Tex.
White, Robert H. ('18) (RSMOFE), Ch. Engr., Am. Loco. Co., Schenectady 5; *for mail*, 1444 Rugby Rd., Schenectady 8, N.Y.
White, Robert Selim (J'46), U.S. Navy, 356 Albion St., Wakefield, Mass.
White, Roger Hartley (J'42) (NZAS), Staff Mem., Engr., Los Alamos Scien. Labs., Box 1663, Santa Fe, New Mex.
White, Severn Andrew ('44) (C), Ch. Safety Engr., Sun Indemnity Co. of N.Y., 55—5th Ave., New York 3; *for mail*, 3027—93rd St., Jackson Heights, N.Y.
White, Thomas ('30) (BHMNW), Managing Dir., Laighpark Engrg. Works, Thomas White & Sons, Ltd.; *for mail*, Hillside, Thornly Park Ave., Paisley, Scotland.
White, Thomas Lester ('44) (MNJBC), Ch. Engr., Commercial Shearing & Stamping Co., 1775 Logan Ave., Youngstown; *for mail*, 602 Willis Ave., Youngstown 7, Ohio.
White, Trescott Siemons ('43) (BENR), Design Engr., Enterprise Eng. & Fdy. Co., 18th & Florida Sts.; *for mail*, 51 Dolores Terrace, San Francisco 10, Calif.
White, Vincent McKim (J'39) (COA), Sales Engr., Export, Westinghouse Elec. Internatl. Co., 657 Wardman Park Hotel, Washington 8, D.C.
White, W. Frank (J'38) (FCEY), Staff Engr., So. Calif. Gas Co., Box 3249, Terminal Annex, Los Angeles 54; *for mail*, 364 W. Broadway, Glendale 4, Calif.
White, W. M. ('06; F'43) (BCH), Cons. Engr., Suite 1115, 735 N. Water St., Milwaukee, Wis.
White, Walter J. ('36), Box 44, Balboa Heights, C.Z.
White, Weston B. (J'45), Asst. Engr., Mech. Engrg. Dept., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York; *for mail*, 141-34—73rd Terrace, Flushing, L.I., N.Y.
White, Wilbert Leonard (J'45) (SOCHDN), Power Plant Prod. Foreman, Tex. Elec. Serv. Co., Elec. Bldg., Ft. Worth 3; *for mail*, 3921 Pershing St., Ft. Worth 7, Tex.
White, William McKean, Jr. (J'41) (MC), V.P., White Mfg. Co., Elkhart, Ind.
White, William Robt. (J'37) (RBSD), Draftsman, Paducah Shop, Ill. Cent. System; *for mail*, 206 Harahan Blvd., Paducah, Ky.
Whiteford, Alexander W. ('21), Engr., Charge R.R. Sales, Haynes-Stellite Co., 30 E. 42nd St., New York, N.Y.
Whiteford, Jas. F. ('08), Gen. Mgr., Cons. Engr., 87 Regent St., London, England.
Whitehead, Charles Powell ('46), Pres., Gen. Steel Castings Corp., Eddystone, Pa.
Whitehead, Robert Charles, Jr. (J'42) (ESJN), Instr. in Mech. Engrg., Dept. of Mech. Engrg., Lehigh Univ., Bethlehem, Pa.
Whitehouse, Irving ('37) (ABCJ), Mgr., Product Devel. Div., Republic Steel Corp., Cleveland; *for mail*, 4409 Renwood Rd., South Euclid, Ohio.
Whitehouse, John Charles (J'45) (BCM), Box 431, Gen. P.O., Adelaide St., Toronto, Ont., Can.
Whitehurst, John C. (J'32), Engr., Taylor-Colquitt Co., Box 1491, Spartanburg, S.C.
Whitener, Ernest Karl ('42) (TCNG), Gen. Supt., Cocker Mch. & Fdy. Co., Box 431, Gastonia, N.C.
Whitescarver, Leland Davenport ('44), Asst. to Mgr., Turb. Div., Gen. Elec. Co., Lynn, Mass.
Whitesel, Harry Alfred ('44) (OKNZEA), 16 Lorraine St., Glen Ridge, N.J.
Whiteside, Donald William (J'46), Sales Engr., Howry-Berg Steel & Iron Works, 2949 N. Speer, Denver 11, Colo.
Whiteside, Jim Tate (J'41) (NDMC), Sr. Engr., Supvr. Mech. Design Sec., Clinton Engr. Works, Tenn. Eastman Corp., Box 590, Knoxville 5; *for mail*, 107 E. Pasadena Rd., Oak Ridge, Tenn.
Whiteside, Saml. P. ('16) (CFS), Test Engr., Swift & Co., Union Stock Yards, Chicago; *for mail*, 640 Hinman Ave., Evanston, Ill.
Whiteside, Victor ('40) (SOFEL), Engr., Burns & McDonnell Engrg. Co., 107 W. Linwood, Kansas City 2; *for mail*, 5318 Harrison, Kansas City 4, Mo.
Whitfield, Mathew Paul (J'40) (JMS), Ensign, U.S.N.R.; Box 52, 354 Central Ave., Centerville, Calif.
Whitford, Robert Henry ('32; A'43) (BGM), Physics-Chemistry Librarian, College of the City of N.Y., Convent Ave. & 139th St.; *for mail*, 100 Hamilton Pl., New York 31, N.Y.
Whitham, Jay Manuel ('83) (HS), Retired; 134 Mill St., Cambridge, Md.
Whiting, Edward M. ('27; '35) (ACDJMZ), V.P., Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Whiting, Harold Willett ('45) (EKNSBH), Ch. Engr., Compressor Div., Worthington Pump & Mchy. Corp., Clinton & Roberts Sts., Buffalo; *for mail*, 521 S. Legion Dr., Buffalo 20, N.Y.
Whiting, John G. ('42) (KCO), 1432 N. Clark St., Chicago 10, Ill.
Whiting, R. A. ('09; '18) (RSEJKF), Power Engr., Pub. Serv. Co. of No. Ill., 72 W. Adams St., Chicago; *for mail*, 160 Olmstead Rd., Riverside, Ill.
Whitley, Frederic Nelson ('14) (FKOSB), Pres., Treas., Frederic N. Whitley, Inc., 173 Pacific St., Brooklyn 2, N.Y.
Whitlock, Elliott H. ('01; F'36) (F), *Manager*, '13-'16, *Vice-President*, '33-'35; Owner, Whitlock Mfg. Co., 11224 Locust Ave., Cleveland 2, Ohio.
Whitlock, Stanley James ('42) (EJS), Ch. Engr., Besler Systems, Emeryville; *for mail*, 118 Greenbank Ave., Piedmont, Calif.
Whitman, Ezra Bailey ('24) (OSHE), Partner, Whitman, Requardt & Associates, 1304 St. Paul St., Baltimore 2, Md.
Whitmarsh, Harry Charles (J'40) (CMADJ), Prod. Control Supvr., A. B. Dick Co., 720 W. Jackson Blvd., Chicago; *for mail*, 1000 Grove St., Evanston, Ill.
Whitmore, Marvin (J'41 (MCSFOD), Lt., U.S.N.R., Coal Inspc. & Utilization, Office of Dir. of Supply, Naval Sta., Norfolk, Va.
Whitmore, Ralph Delano ('45) (NMCLDY), Mech. Engr., 20th Century-Fox Film Corp., 444 W. 56th St., New York 19, N.Y.
Whitmore, William Wallace, III (J'46) (EMO), 283 High St., Lockport, N.Y.
Whitmyre, George Robert (J'37) (FS), Cent. Y.M.C.A., 2200 Prospect Ave., Cleveland 15, Ohio.
Whitney, H. LeRoy ('22; F'38), Rm. 400, 41 E. 42nd St., New York 17, N.Y.
Whitney, Maurice Page ('30) (JAH), Acting Gen. Mgr., Eclipse Mch. Div., Bendix Aviation Corp., Elmira, N.Y.
Whitney, Morgan M. (J'23) (EK), Sales Engr., Griscom-Russell Co., 285 Madison Ave., New York, N.Y.
Whitney, Wm. M. (A'86), Pres., Baxter D. Whitney & Son, Inc., Winchendon, Mass.
Whiton, Herbert Starkes ('15) (SEBFKD), Ch. Mech. Engr., Pub. Utility Engrg. & Serv. Corp., 231 S. La Salle St., Chicago 4, Ill.
Whiton, Lucius E. ('05), Pres., Treas., D. E. Whiton Mch. Co., 190 Howard St.; *for mail*, 836 Pequot Ave., New London, Conn.
Whitsit, Lyle Antrim ('22) (HOSCJF), Cons. Hyd. Engr., Ebasco Internatl. Corp., 2 Rector St., New York 6, N.Y.
Whitsitt, William Baker ('18) (BEFRS), Retired; 2310 Poplar Dr., Larchmont, Baltimore 7, Md.
Whitson, Lee Seymour ('39; '46) (CMDLNY), Ch. Indus. Engr., Minn. Min. & Mfg. Co., 900 Fauquier Ave., St. Paul; *for mail*, 4841 Harriet Ave., Minneapolis 9, Minn.
Whitt, Sidney A. (J'33) (CKL), Ch. Engr., Cordley & Hayes, 443—4th Ave., New York 16; *for mail*, 630 Palisade Ave., Yonkers, N.Y.
Whittaker, Harry ('45) (JLNTYD), Cons. Engr., 211 Central Ave., Cranford, N.J.
Whittaker, Harry Leroy (J'44) (FKS), Supt., Steam Dept., Dow Chem. Styrene Div., Dow Chem. Co., Los Angeles; *for mail*, 303 Glendora Ave., Long Beach 3, Calif.
Whittelsey, Charles Chauncey ('42) (CE), V.P., Ford, Bacon & Davis Constr. Corp., 39 Broadway, New York 6, N.Y.
Whittemore, Herbert Lucius ('03; '10; F'36) (B), *Manager*, '30-'33; Chief, Engrg. Mechanics Sec., Natl. Bur. of Stands., Connecticut Ave. & Van Ness St., Washington 25; *for mail*, 3906 McKinley St., N.W., Washington 15, D.C.
Whitting, Henry Colmer (J'39) (CMJB), Capt., Ord. Dept., War Dept., Office of Chief of Ord., Watertown Arsenal, Watertown, Mass.
Whittingham, David John (J'42) (HNC), Ch. Project Engr., Hyd. Div., N.Y. Air Brake Co.; *for mail*, 407 Cross St., Watertown, N.Y.
Whittlesey, David Walter (J'40) (ABC), 3404 N.E. 17th Ave., Portland 12, Ore.
Whittlesey, F. E. ('19) (BEJ), V.P., Raymond Mfg. Co., Div. of Associated Spring Corp., 226 S. Center St., Corry, Pa.
Whittlesey, James Wright (J'40) (ACE), Lt., U.S.N.R., Matl. Officer, Scout Observation Serv. Unit, F.P.O., San Francisco, Calif.; *for mail*, 3404 N.E. 17th Ave., Portland 12, Ore.
Whyte, Carl Barzellous ('46) (COP), Ch. Refinery Engr., Tutwiler Refinery, Cities Serv. Refining Corp., Lake Charles, La.; *for mail*, Box 423, Chicago 90, Ill.
Wiberg, Oscar A. ('31), Svenska Turbinfabriks, Akttiebolaget, Ljungstrom, Finspång, Sweden.
Wibling, Seth E. (J'34), H. Wibling Tool & Mfg. Co., 94 Taylor St., Danbury, Conn.
Wicher, William Edward (J'45), 193 Lynn Shore Dr., Lynn, Mass.
Wichum, Victor ('16; '21) (LYTZ), Maj., Industrial Engr., National Defense Research Comm., Franklin Inst., Philadelphia, Pa.; *for mail*, 1291 Dean St., Brooklyn 16, N.Y.
Wick, G. Rodney (J'32) (FKL), Process Engr., Jefferson Chem. Co., Inc., 30 Rockefeller Plaza, New York 20, N.Y.; *for mail*, 766 Fairacres Ave., Westfield, N.J.
Wick, Jas. L., Jr. ('32), Pres., Gen. Mgr., Falcon Bronze Co., 218 S. Phelps St., Youngstown, Ohio.
Wickenden, Thomas Howard ('28) (JC), Mgr., Devel. & Research Div., Internatl. Nickel Co., Inc., 67 Wall St., New York 5, N.Y.
Wickenden, William Elgin ('29) (C), Pres., Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6, Ohio.
Wickersham, John H. ('21) (CDEH), 14 S. Duke St., Lancaster, Pa.
Wickersham, Nathan Rowland ('07) (CEJM), Retired; 106 E. 5th St., Corning, N.Y.
Wicklatz, Edward G. ('42) (BJM), 5154 Montrose Ave., Chicago, Ill.
Wickman, Lester Frederick ('44) (NMJLOD), Ch. Engr., Mech. Equip. Engrg. Co., Inc., 170 Alta Rd., Oakland 11, Calif.

Wicks, Clarence James (J'42) (CSTFDJ), Asst. Plant Engr., Ludlow Mfg. & Sales Co.; *for mail*, 45 Park Pl., Ludlow, Mass.
Wickstrom, Harold Ward ('45) (OEFKH), Cons. Engr., 507 S. Fuller Ave., Los Angeles 36, Calif.
Widdowson, James Alfred (J'42) (KBZ), Seaman, 1st Class, 114-434 Navy Pier, Chicago, Ill.; *for mail*, 200 Front St., York, Pa.
Wideman, Milton Alan (J'42) (A), 101 E. 13th Ave., Columbus 1, Ohio.
Widmer, Arthur J. ('38) (MDCOL), Pres., Widmer Engrg. Co., Suite 918, 122 N. 7th St., St. Louis 1; *for mail*, 212 S. Elm Ave., Webster Groves 19, Mo.
Wieber, George A. ('15; '20; '35) (SELDT), Cent. Sta. Divs., Gen. Elec. Co., Schenectady; *for mail*, 15 Wallace St., Scotia 2, N.Y.
Wied, Julius Paul ('43) (BMNY), Project Engr., Apex Elec. Mfg. Co., Sandusky, Ohio.
Wiedmer, Jack Bernard (J'42) (CEHMS), Lt. (j.g.), U.S.N.R., Ord. Stock Office, Naval Gun Factory, Bur. of Ord., Navy Dept., Washington 25; *for mail*, 711 E. Capitol St., Washington, D.C.
Wieland, Chas. F. ('01) (ACM), 199 Hillcrest Rd., Berkeley, Calif.
Wienecke, Herman A. ('37), 1516 S. Florence Ave., Tulsa, Okla.
Wiener, Max (J'42) (AEM), 275 Seymour Ave., Newark, N.J.
Wiener, Philippe Jules (J'44), 2nd Lt., Inf., A.U.S., 3556th QM. Truck Co., A.P.O. 162, c/o Postmaster, New York; *for mail*, 12 W. 72nd St., New York 23, N.Y.
Wiener, Robert Paul ('42) (NMAH), Seaman, 2nd Class, Ship's Co. Gunnery Bks. 113, A.S. Post Office, Camp Peary, Va.
Wier, Joseph Patterson (J'45) (MAJCN), Design Engr., Geophysical Equip. Div., Magnolia Petroleum Co., Box 900, Dallas; *for mail*, 4625 Belmont, Dallas 4, Tex.
Wier, Robert Louis (J'46), Prod. Engr., Plant 23, Fisher Body Div., Gen. Motors Corp., Detroit; *for mail*, 18473 Whitcomb, Detroit 19, Mich.
Wier, William Parker, Jr. (J'42) (BJLZ), Process Engr., Bausch & Lomb Optical Co., 635 St. Paul St., Rochester 2; *for mail*, 474 East Ave., Rochester 7, N.Y.
Wiese, Oscar Henry ('16; '20) (EFCKJS), Ch., Inspc. & Survey Sec., War Shipping Admin., 45 Broadway, New York 6; *for mail*, 8448—85th Ave., Woodhaven 21, N.Y.
Wiesley, Donald Bruce (J'40) (CMN), Asst. Plant Mgr., Am. Can Co., 37th & E. 8th St., Oakland 1; *for mail*, 3601 Calafia Ave., Oakland 3, Calif.
Wiggenhorn, Roman Hubert (J'43) (BNHMAO), Lt., Army Air Forces, Engrg. Officer, A.P.O. 19905-A, c/o Postmaster, San Francisco, Calif.; *for mail*, 337 Clover St., Dayton 10, Ohio.
Wiggin, Frederick Alexander (J'37) (ABR), Wahiawa, T.H.
Wiggin, Rinaldo Edward ('45), Engr., Gen. Elec. Co., 40 Federal St., Lynn; *for mail*, 41 Ballard St., Saugus, Mass.
Wiggins, Carlyle Aubrey, Jr. (J'44) (RSEHBN), Spec. Apprentice, M.P. Dept., Norfolk & West. Ry., Box 197, Roanoke 2, Va.
Wiggins, Maurice Edwin (J'37) (RCKFS), Gang Foreman, Supvr. Pool Equip., Wilmington Shops, Pa. R.R., Wilmington, Del.; *for mail*, 300 Main St., Riverton, N.J.
Wiggins, William David, Jr. (J'41) (ACKS), Ensign, U.S.N.R.; *for mail*, 701 Hazelhurst Rd., Merion, Pa.
Wiggs, Gordon Lorne ('41) (OCM), Prin., G. Lorne Wiggs & Co., 1411 Crescent St., Montreal 25, Que., Can.
Wight, Collins ('17; '19) (CHMOS), Cons. Engr., 129 S. Ludlow St., Dayton 2, Ohio.
Wightman, Frank Allen ('24) (RSE), Asst. Mech. Engr., Mass. Dept. of Pub. Utilities, 167 State House, Boston 33; *for mail*, 14 Calvin Rd., Newtonville 60, Mass.
Wigle, Roy A. (J'26) (YCBETR), Owner, Imperial Rug Co., 630 S. Fulton Ave., Mt. Vernon; *for mail*, 15 Minerva Dr., Yonkers, N.Y.
Wigren, Clarence F. ('43) (CMS), Estimating Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Wiitanen, Wilho William ('36; '46) (KLS), Stacey-Dresser Engrg. Co., 2063 E. 4th St., Cleveland 15, Ohio.
Wikander, Oscar Ragnar ('19) (BCJMNR), *Melville Medallist*, '35; Mech. Engr., Ring Spring Dept., Edgewater Steel Co., Box 478, Pittsburgh 30, Pa.
Wilberding, Marion X. ('28) (OSHCF), Pres., Wilberding Co. Inc., 1822 Eye St., N.W., Washington 6, D.C.
Wilbur, Ralph S. ('12; '16; '34) (SHEF), Prof. Mech. Engrg., Chmn., Mech. Engrg. Dept., Duke Univ., Box 265, Durham, N.C.
Wilbur, Richard Snow (J'41) (CJSF), Bethlehem Steel Co., Sparrows Point; *for mail*, 3014 Guilford Ave., Baltimore 18, Md.
Wilcox, Abbott David (J'37) (SLNBDO), Honolulu Iron Works Co., 165 Broadway, New York 6, N.Y.; *for mail*, 34 Shelter Cove Rd., Milford, Conn.
Wilcox, Allan Chester (J'42) (KJSEO), 1st Lt., Army Air Forces, Engrg. Officer, Route 1, East Lansing, Mich.
Wilcox, Carl Clifford ('05; '15) (SFEKHJ), Head, Mech. Engrg. Dept., Univ. of Notre Dame, Notre Dame; *for mail*, 1507 E. Colfax Ave., South Bend 17, Ind.
Wilcox, Clyde E. ('34; '35), Pilot, Air Force Ferrying Comd., Municipal Airport, Nashville, Tenn.
Wilcox, Donald Brooks (J'41) (MO), Prof., Dept. of Indus. Engrg., Ga. Sch. of Tech., Atlanta, Ga.
Wilcox, Edward Milton ('44) (CFS), Dist. Mgr., Foster Wheeler Corp., 1304 Union Cent. Bldg., Cincinnati 2, Ohio.
Wilcox, Harold C. ('24; '35) (EMR), Assoc. Mech. Editor, Simmons-Boardman Publ. Corp., 30 Church St., New York 7, N.Y.
Wilcox, Joseph S. ('42), 623 James St., Pelham Manor, N.Y.
Wilcox, Perley S. ('99; '14), Chmn. of Bd., Eastman Kodak Co., 343 State St., Rochester 4, N.Y.
Wilcox, Ralph Stanley (J'45) (AK), Lt., Army Air Forces; 5446 Shasta Ave., Richmond, Calif.
Wilcox, Walter M. (J'46) (CJO), Keeney Mfg. Co., Newington, Conn.
Wilcox, Ward William (J'42) (ACM), 6291 Dawning Dr., Cleveland 9, Ohio.
Wilcoxon, Erving M. (J'41) (AJM), Office of Chief of Ord., Washington, D.C.
Wilcoxson, Leslie Swales (26; '36) (FKSRJL), V.P., Research & Devel., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Wild, Albert Franklin (J'37) (AZC), Mech. & Elec. Engr., Gen. Elec. Co., Schenectady; *for mail*, 33 E. North St., Ballston Spa, N.Y.
Wilde, Frederick George (J'39) (BN), Design Engr., Paragon-Revolute Corp., 77 South Ave., Rochester 4; *for mail*, 1075 Park Ave., Rochester 10, N.Y.
Wilde, Ray S. M., Jr. (J'44) (AES), Lt., Army Air Forces, Hdq., Air Tech. Serv. Comd., Box 1721, Area "A", Wright Field, Dayton, Ohio.
Wilde, Thomas B. (J'32) (HJCML), Owner, Prod. Mold & Mch. Co., 8507 Park Lane, Los Angeles 1, Calif.
Wilder, Cecil Lyman ('33; '42) (ZML), Devel. Engr., Taylor Instrument Cos., 95 Ames St., Rochester; *for mail*, 177 East Pkwy., Rochester 5, N.Y.
Wilder, Forrest Gay ('44) (BDFJNO), Mar. Surveyor, Appraiser & Cons. Engr., 131 State St., Boston; *for mail*, 38 Circuit Rd., Winthrop, Mass.
Wilder, Henry C. (J'37) (CLM), Methods Engr., West Lynn Works, Gen. Elec. Co., 40 Federal St., West Lynn; *for mail*, 304 Lowell St., Lawrence, Mass.
Wilder, Herbert Pearce ('42) (SK), 1st Asst. Engr. Officer, U.S. Maritime Serv., 37 Broadway, New York; *for mail*, 27 Central Ave., Hartsdale, N.Y.
Wildhaber, Ernest ('23; '35) (MBJA), Mech. Engr., Gleason Works, Rochester 3; *for mail*, 124 Summit Dr., Rochester 10, N.Y.
Wilds, Horace William ('39) (CMS), Bldg. Mgr., N.Y. Life Ins. Co. Bldg., 51 Madison Ave., New York 10, N.Y.
Wilenzick, Bernie (J'37) (EFP), Ch. Clerk, Measurement Dept., Interstate Natural Gas Co., Inc., P.O. Box 1482, Monroe, La.
Wiley, Fred Edwin (J'40) (YBCNZM), Ch. Engr., Great Am. Plastics Co., 55 North St., Fitchburg; *for mail*, 465 Merriam Ave., Leominster, Mass.
Wiley, Joseph Norris (J'41) (MNBZH), Asst. Product Engr., Sperry Gyroscope Co., Inc.; *for mail*, 151 Schenck Ave., Great Neck, L.I., N.Y.
Wiley, Paul Rogers (J'44), W.Va. Pulp & Paper Co., Charleston, S.C.
Wiley, Russell C. (J'37) (CLS), Mech. Engr., Jenks & Ballou, 2600 Indus. Trust Bldg., Providence, R.I.
Wiley, Wm. O. (A'25), Chmn. of Bd., John Wiley & Sons, Inc., 440—4th Ave., New York, N.Y.
Wilferth, Lawrence Edwin, Jr. (J'46) (CEM), U.S. Navy, 2804 W. Alhambra Rd., Alhambra, Calif.
Wilhelm, Dean Morlan ('40) (CLMN), Works Mgr., Patterson Fdy. & Mch. Co., 1250 St. George; *for mail*, R.D. 2, East Liverpool, Ohio.
Wilhelm, Jack Elmore ('41) (SFZALC), Mech. Engr., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio; *for mail*, White Oak & Sycamore Drs., White Oak Heights, Pittsburgh 9, Pa.
Wilhelm, Wilbur Forrest, Jr. (J'43) (ABHKN), Design Engr., Aviation Gas Turbine Div., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia; *for mail*, 411 Nassau Blvd., Prospect Park, Pa.
Wilhoit, Lowell Maynard ('36; '45) (MJFBCP), Ch. Engr., Otis Engrg. Corp. & Otis Pressure Controls, Inc., Box 7206, 6612 Denton Dr., Dallas 9, Tex.
Wilke, Carl William ('40) (DCF), Asst. Supvr. of Works, Stand. Tool Co., 6918 Central Ave., Cleveland 4, Ohio.
Wilke, William Peter (J'34) (JCO), Mech. Engr., Hammond Lead Products Inc., 1st Trust Bldg.; *for mail*, 21 Glendale Pk., Hammond, Ind.
Wilkenfeldt, John William ('29; '42) (LBDK), Designing Engr., Hoffman-La Roche, Inc., Roche Park, Nutley; *for mail*, 11 Aldon Terrace, Bloomfield, N.J.
Wilkens, John A. (A'33) (GCATY), V.P., Treas., Sweeney Lithograph Co., 69 Academy St., Belleville, N.J.; *for mail*, 358 Crown St., Brooklyn, N.Y.
Wilkerson, Edgar Howard (J'40) (KESH), Assoc. Engr., U.S. Engrg. Dept., 7515 Figueroa St.; *for mail*, 727 W. 78th St., Los Angeles 44, Calif.
Wilkins, Charles Norman ('46) (CO), Bldg. Serv. Engr., Day & Zimmermann, Inc., Packard Bldg., Philadelphia 2; *for mail*, 426 Roberts Ave., Glenside, Pa.
Wilkins, Roger Felch (J'41) (ES), Fire Protection Inspr., Factory Ins. Assn., 555 Asylum St., Hartford, Conn.; *for mail*, 10 Forrest Pl., North Attleboro, Mass.
Wilkins, Roy ('31), Retired; Hamburg, Calif.
Wilkins, Vernon Bates (J'45), 1011 Walnut Ave., Syracuse 10, N.Y.
Wilkins, William Burdette ('44) (MNWY), 245 E. Ridgewood Ave., Ridgewood, N.J.
Wilkinson, Archibald Stewart (J'33) (SBKH), Chargeman, Engrg. Design Sec., Fed. Shipbldg. & Dry Dock Co., Lincoln Highway, Kearny; *for mail*, 12 N. 21st St., East Orange, N.J.
Wilkinson, Ford Lee, Jr. ('26; '29) (KSF), *Vice-President*, '43-'45; Prof., Dean, Mech. Engrg., Speed Scien. Sch., Univ. of Louisville, Louisville 8, Ky.
Wilkinson, George David, Jr. ('33; '45) (CDM), *Charles T. Main Award*, '33; Capt., Admin. Mgmt. Sect., Control Div., Ord. Dept., The Pentagon, Washington 25, D.C.; *for mail*, 2626 S. Lynn St., Arlington, Va.
Wilkinson, Thomas Lee ('94; '05; F'36) (CEFHS), *Vice-President*, '24-'26; 2506 Iowa St., Davenport, Iowa.
Wilkinson, Thomas Morris (J'45), 147 W. Gleen, Auburn, Ala.
Wilks, August Charles ('25; '35) (CMOJDL), A. C. Wilks Co., R.F.D. 2, Vail Rd., Bethel, Conn.
Will, Donald S. (J'41) (ABH), 200 N.W. 21st St., Oklahoma City, Okla.
Willard, Arthur Cutts ('19) (KFS), Pres., Univ. of Ill., Urbana, Ill.
Willard, Donald E. ('45), Pres., Allith-Prouty, Inc., Box 606; *for mail*, Danville, Ill.
Willard, John Artemas ('17; '21) (CEMOT), Partner, Bigelow, Kent, Willard & Co., 500—5th Ave., New York 18, N.Y.
Willard, Leigh ('07; '10) (FYE), Pres., Interlake Iron Corp., 1900 Union Commerce Bldg., Cleveland 14, Ohio.
Willard, Warren Howard (J'44) (CMSZLO), Time Study Engr., Camera Works, Eastman Kodak Co., 333 State St., Rochester 4; *for mail*, 83 Ave. C, Point Pleasant, N.Y.
Willcox, Frederick Preston ('44; '44) (GOZ), Cons. Engr., Ansco Div., Gen. Aniline & Film Corp., Binghamton; *for mail*, 216 Vestal Rd., Vestal, N.Y.
Willcox, George Bingham ('95; '08) (LMDOZY), Pat. Atty., 900 S. Warren Ave., Saginaw, Mich.
Willcox, John Frederic ('44) (CDKSML), Asst. Mgr., Eatmore Ice Cream Div., Beatrice Creamery Co., Los Gatos; *for mail*, 701 Menlo Oaks Dr., Menlo Park, Calif.
Willemin, Robert B. (J'41) (CMJDYH), Matls. Engr., Naval Shipyard, Philadelphia; *for mail*, 1500 Devereaux Ave., Philadelphia 24, Pa.
Willerton, Gustav E. ('27; '35) (ABDMN), Mch. Designer, Internatl. Business Mchs. Corp., 310—5th Ave., New York 1; *for mail*, 300 W. 12th St., New York 14, N.Y.
Willett, Francis Merrill ('16; '22; '35) (SFOK), Sr. Engr., Picatinny Arsenal, Dover; *for mail*, 253 S. Main St., Wharton, N.J.
Willey, Calvin McAuley (J'45) (FKS), Mech. Engrg. Instr., Carnegie Inst. of Tech., Pittsburgh 13, Pa.
Willey, Frank William ('19) (C), Owner, Willey-Wray Elec. Co., 1523-7 Central Pkwy., Cincinnati 14, Ohio.
Willhelm, Oscar F. ('31; '33) (MAJ), V.P., Durham-Enders Razor Corp., Mystic; *for mail*, 629 Ocean Ave., Bingham Park, New London, Conn.
Willi, Albert Bond ('36) (ACEM), Exec. V.P., Continental Aviation & Engrg. Corp., Muskegon; *for mail*, 441 McKinley Rd., Grosse Pointe 30, Mich.
Willi, Albert Bond, Jr. (J'40) (EAFBJN), Design Engr., Fed.-Mogul Corp., 11031 Shoemaker St., Detroit 13; *for mail*, 15797 Deerfield Ave., East Detroit, Mich.
Williamowsky, David Joseph (J'43) (CHST), Tech. Editor, Ord. Publications, Naval Gun Factory, Washington; *for mail*, 1425 G St., N.E., Washington 2, D.C.
Williams, Allan James, Jr. (J'44) (ACMWY), Asst. Engr., Dorez Plastics & Chemicals Co., Walck Rd., North Tonawanda; *for mail*, 107 Minerva St., Tonawanda, N.Y.
Williams, Alpheus Fuller (A'00) (MJORTD), Chmn., Alpheus Williams & E. G. Dowse, Ltd., Box 399, Germiston, Transvaal, S. Africa.
Williams, Arthur ('36) (RSKFBE), Ch. Engr., Superheater Co., East Chicago, Ind.
Williams, Charles Ellsworth (J'43) (HSKBDG), 3rd Engr., Merchant Marine; *for mail*, 6 Liberty Court, Hempstead, N.Y.
Williams, Charles W. (J'38) (EMBAC), Engr., Liaison between Mfg. & Cent. Engrg., Dept. 428, Highland Park Plant, Chrysler Corp., Box 1919, Detroit 31, Mich.

Williams, Chester Arthur (J'42) (CM), Asst. to Factory Supvr., Radio Tube Div., Tung-Sol Lamp Works, 370 Orange St., Newark, N.J.
Williams, Chester E. ('41), V.P., Treas., Gen. Mgr., Johnson & Bassett, Inc., 114 Foster St.; *for mail*, 41 Laconia Rd., Worcester, Mass.
Williams, Chester Monroe ('45) (NOD), Design Engr., Stand. Steel Corp., 5001 S. Boyle Ave., Los Angeles 11; *for mail*, 1049 N. Hayworth Ave., Los Angeles 46, Calif.
Williams, Clayton H. (J'36) (BMW), Bell Tel. Labs., Inc., Whippany Radio Lab., Whippany; *for mail*, 74 Belmont Ave., Dover, N.J.
Williams, Clyde E. ('45) (CJLARY), Dir., Battelle Memorial Inst., 505 King Ave., Columbus 1, Ohio.
Williams, David Gordain ('18; '24) (NKDCMS), Ch. Engr., Trojan Powder Co., 17 N. 7th St.; *for mail*, 113 S. 16th St., Allentown, Pa.
Williams, David R. (J'41) (BCJ), Capt., 566 Green St., Havre de Grace, Md.
Williams, David T. ('20) (ERC), Asst. to V.P., Am. Loco. Co., 30 Church St., New York 8, N.Y.
Williams, Douglas Ward ('37) (SFE), Sales Mgr., Babcock & Wilcox Co., 1515 Guardian Bldg., Cleveland 14, Ohio.
Williams, Edward Eugene ('24) (SFCK), *Manager, '45; Vice-President, '45-'46*; Gen. Supt., Steam Plants, Duke Power Co., Charlotte 1, N.C.
Williams, Edward Everett (J'43) (R), Capt., 763rd Ry. Shop Bn., Camp Millard, Bucyrus, Ohio.
Williams, Ernest J. (J'40), 155 S. Colonial Ave., Richmond, Va.
Williams, Ervin Maurice (J'38) (FRS), Ch. Engr., Clinchfield Div., Pattison & Bowns, Inc., [illegible]
Williams, [illegible] ('43) (JCM), Engr., Rogers Mch. Works, Inc., 61 N. Main St., Alfred, N.Y.
Williams, Frank S. G. ('34; '35) (BHJP), Mgr., East. Sales, Taylor Forge & Pipe Works, 50 Church St., New York, N.Y.; *for mail*, 540 Elm St., Westfield, N.J.
Williams, George David (J'45) (LSBRK), Draftsman, Process Piping Div., E. B. Badger & Sons Co., 75 Pitts St., Boston, Mass.
Williams, George Leslie ('42) (DLJEYC), Gen. Engr., Armour & Co., Union Stock Yards, Chicago 9; *for mail*, 321 Bell Ave., La Grange, Ill.
Williams, George M. ('21; '35) (ACD), Pres., Russell Mfg. Co., Middletown, Conn.
Williams, [illegible] Sales Mgr., Allen Billmyre Co., 431 Fayette Ave., Mamaroneck; *for mail*, Park Plaza Apts., Larchmont, N.Y.
Williams, Guy Franklin ('44) (SKLCEY), Devel. Engr., Dowell Inc., Kennedy Bldg., Tulsa 3; *for mail*, 1903 E. 37th St., Tulsa, Okla.
Williams, Harry D. (J'46) (ACM), Lt., U.S.N.R., Engrg. Div., Assembly & Repair Dept., Naval [illegible], Norfolk 11, Va.
Williams, Herman B. (J'32) (CMNOHG), Re[illegible], Devel. & Design Engr., Jones & Lamson Mch. Co.; *for mail*, 8 Fairview, Springfield, Vt.
Williams, Howard Edward ('25; '04) (BNR), Ch. Mech. Draftsman, Calumet & Hecla Consltd. Copper Co.; *for mail*, 1144 Calumet Ave., Calumet, Mich.
Williams, Howard O. ('30; '33) (CM), Gen. Mgr., R. E. Howell Co., Minneapolis 14, Minn.
Williams, J. Howard ('24) (DJMNOS), Mech. Engr., Grinnell Corp., 260 W. Exchange St., Providence 1, R.I.
Williams, Jack Harry ('44) (SFKOZE), Cons. Engr., Gilbert Associates, 412 Washington St.; *for mail*, Box 73, Reading, Pa.
Williams, James Lawrence (J'45) Apprentice Seaman, V-12, U.S.N.R.; 115 Dollner St., Visalia, Calif.
Williams, Jarvis, Jr. ('45) (CMOAEG), Cons. Indus. Engr., 8 E. 45th St., New York 17, N.Y.
Williams, John Davis (J'33) (TEF), Plant Engr., Kendall Co., Walpole; *for mail*, Medfield, Mass.
Williams, [illegible] War Assets Admin.; *for mail*, 3811 Newark St. N.W., Washington, D.C.
Williams, John Robert (J'37) (RCAEM), Spec. Machinist, Ill. Cent. R.R., Vicksburg; *for mail*, 309—15th St., N., Columbus, Miss.
Williams, Leonard Karl (J'43) (HMS), 302 N. 2nd St., Minersville, Schuylkill Co., Pa.
Williams, Llewellyn ('07), V.P., Charge of Engrg. & Mfg., York Corp., Roosevelt Ave., York, Pa.
Williams, Louis William ('12; '25) (JCMNOD), Cons. Engr., 149 Broadway, New York; *for mail*, 216 Clinton Ave., Brooklyn 5, N.Y.
Williams, Morris Franklin (J'29) (BCDJMO), Supt. Maint. & Constr., Universal Cyclops Steel Co., Bridgeville; *for mail*, 442 Fairway Dr., Pittsburgh 27, Pa.
Williams, Paul ('17; '25) (CAEMO), Works Mgr., Timken Steel Corp., 1201 Camden Ave., S.W., Canton 6, Ohio.
Williams, R. Arthur ('45) (R), V.P., Am. Car & Fdy. Co., 30 Church St., New York, N.Y.
Williams, Ralph L. ('24; '27) (EFKSZ), Owner, Firm of Ralph L. Williams, Engrg. & Sales, 49 Federal St., Boston 10, Mass.
Williams, Ray Shields (J'43) (HJMVSZ), Prod. Engr., Plant 1, Hayes Corp., Orrville, Ohio.
Williams, Richard Elbert (J'42) (CSBDML), R.R. 2, Hayworth, Ill.
Williams, Richard Franklin (J'43) (CKS), [illegible] U.S.N.R., Engr. Officer, U.S.S. *Steinaker*, DD-863, F.P.O., New York, N.Y.

Williams, Robert Eubank, Jr. (J'43) (SKCYAO), Application Engr., Westinghouse Elec. Corp., 1407 Comer Bldg., Birmingham 3; *for mail*, 6 Norman Dr., Birmingham 9, Ala.
Williams, Robert K. (J'39) (CM), 514 Pickwick Bldg., 903 McGee St., Kansas City, Mo.
Williams, Robert Lawrence (J'42), Timken Roller Bearing Co., 1832 Union Commerce Bldg., Cleveland, Ohio.
Williams, Robert Maxwell ('45) (ACEKNJ), Ch. Design Engr., Aircraft Eng. Div., Packard Motor Car Co., 1330 Laskey Rd., Toledo 12; *for mail*, 1361 Wildwood Rd., Toledo 9, Ohio.
Williams, Samuel Crane ('17; '26) (AY), Dir. of Research, Scudder, Stevens & Clark, 1 Wall St., New York 5, N.Y.
Williams, Samuel Longfellow (J'24) (RCBZJY), East. Mgr., Westinghouse Air Brake Co., 3400 Empire State Bldg., New York 1, N.Y.
Williams, Samuel S. ('42) (SFKDOH), Combustion & Power Engr., 816 Martin Bldg., Birmingham 3, Ala.
Williams, Sayre Stuart (J'43) (ACE), Lt. (j.g.), U.S.N.R.; 3727 Pershing Ave., San Diego 4, Calif.
Williams, Theodore Cortlandt ('39) (CSL), Constr. Mgr., Stone & Webster Engrg. Corp., 90 Broad St., New York 4, N.Y.; *for mail*, 6 Columbia Ave., Vineland, N.J.
Williams, Thomas Stanley (J'42) (CGNLKD), Pres., Superior Decals, Inc., 2803 Ft. Worth Ave., Dallas 11, Tex.
Williams, William Albert (J'39) (BCDL), Ch. Engr., Am. Pulley Co., 4200 Wissahickon Ave., Philadelphia 29, Pa.
Williams, William Francis (J'40) (CAML), Ensign, U.S.N.R., Engr., Power Plants Design, Bur. of Aero., Navy Dept., Washington 25, D.C.; *for mail*, 9401 Flower Ave., Silver Spring, Md.
Williamson, Charles W. ('20; '35) (NMDBAJ), Designer & Checker, Morgan Constr. Co., 15 Belmont St., Worcester; *for mail*, Box 341, 228 Worcester St., North Grafton, Mass.
Williamson, Frank A. ('37) (BEP), Engr., Designer, So. Counties Gas Co., Rm. 716, 810 S. Flower St., Los Angeles; *for mail*, 426 Pioneer Dr., Glendale, Calif.
Williamson, Gerald V. ('26; '37) (SFKJCE), System Engr. & Steam Power Oper., Union Elec. Co. of Mo., 315 N. 12th Blvd., St. Louis 1, Mo.
Williamson, Harry Walter ('27), Supvg. Engr., Gen. Elec. Co., 106 W. 14th St., Kansas City 6, Mo.; *for mail*, 4134 Eaton St., Kansas City 3, Kan.
Williamson, John Louis (J'42), 68-30 Burns St., Forest Hills, N.Y.
Williamson, Robert Charles (J'41) (JBMK), Engrg. Rep., Gen. Elec. Co., Ft. Edward Plant, Ft. Edward; *for mail*, 125 Elmer Ave., Schenectady 8, N.Y.
Williamson, William R. ('21; '26), Cons. Engr., 208 S. La Salle St., Chicago, Ill.
Williamson, William Rodger (J'43) (KEBSHR), Devel. Engr., Maxim Silencer Co., 85 Homestead Ave., Hartford; *for mail*, 37 Hughes St., Hartford 6, Conn.
Williford, David P. (J'41) (ABG), 1510 S. 91st St., Seattle 8, Wash.
Williges, John Anthony (J'43) (ONDLSH), Engr., Hamilton & Williges, 718 Bank of Commerce Bldg., Oakland 12; *for mail*, 15 Crescent Ave., San Francisco 10, Calif.
Willis, Alvin H. (J'41) (BHK), Cons. Engr., 3628 Hilmar Rd., Baltimore 7, Md.
Willis, Burton Clifford, Jr. (J'40) (ACOFBJ), Mech. Engr., Pa. Flexible Metallic Tubing Co., 72nd & Powers Lane, Philadelphia 42; *for mail*, Creek Rd., Westtown, Pa.
Willis, Cecil Custer ('42) (EFS), Supt. of Generation, Okla. Gas & Elec. Co., 3rd & Harvey Sts., Oklahoma City 2, Okla.
Willis, Charles C. ('20) (CGM), Mgr., Field Engrg. Shop, Spool Cotton Co., 745—5th Ave., New York 22, N.Y.; *for mail*, Cedar Crest, Bound Brook, N.J.
Willis, Cullie Bosacki, Jr. (J'43) (SEFL), 606 St. Charles Ave., Montgomery, Ala.
Willis, David Ellery (J'43), Box 1950, Route 4, Tampa, Fla.
Willis, John Brewster (J'40) (EHY), Sr. Partner, Ch. Engr., Willis Engrg. Co., Burton, Ohio.
Willis, Ogden Edwards (J'39) (AELSY), Jr. Mech. Engr., E. I. du Pont de Nemours & Co., Belle; *for mail*, Coalburg, W.Va.
Willis, Peyton Tyler ('45) (BEFHKS), Engr., Leland S. Rosener, 233 Sansome St., San Francisco 4; *for mail*, 2110—26th Ave., San Francisco 16, Calif.
Willis, Philip Anton ('29; '35) (SENKMA), Asst. Examiner-in-Charge, Engrg. Unit, Examining Div., Civ. Serv. Comm., 8th & F Sts., N.W., Washington 25; *for mail*, 2912 Ft. Baker Dr., S.E., Washington 20, D.C.
Willis, Richard Lewis ('33; '44) (FCS), Supt. of Combustion, Bethlehem Steel Co., E. 3rd St.; *for mail*, 634—8th Ave., Bethlehem, Pa.
Williston, Arthur L. ('96; '99) (CMS), Retired; 936 High St., Dedham, Mass.
Willits, Charles William ('36) (CYWM), Asst. Gen. Mgr., Mitts & Merrill, 1009 S. Water St.; *for mail*, 251 Lockwood St., Saginaw, Mich.
Willits, Victor W. (J'36) (CHLZ), M.M., Procter & Gamble Mfg. Co., 1226 Loomis St., Dallas 2; *for mail*, 2820 Hedge Row, Dallas 9, Tex.
Willoughby, Victor Roy ('21) (R), Retired; R.F.D. 1, Concord, N.H.

Wills, Christian A. ('31), V.P., Gen. Mgr., Wm. B. Pollock Co.; *for mail*, 60 W. Princeton Ave., Youngstown, Ohio.
Wills, Howard Andrew ('43) (BEFJKN), Field Rep., Gibbs & Cox Inc., 21 West St., New York 6, N.Y.; *for mail*, Box 696, Tacoma 1, Wash.
Wills, John Adrian (J'43) (BLWY), Pres., Wills & Roberts Plastics Mfg. Corp., 33 S. Lake Ave., Pasadena 1, Calif.
Wills, John Gordon, Jr. ('34; '45) (CJMBDL), Engr., Blaw-Knox Co., Box 1198, Pittsburgh; *for mail*, 701 Delaware Ave., Oakmont, Allegheny Co., Pa.
Wills, John Robert (J'43) (ACNMSO), Asst. Lab. Equip. Engr., Boeing Aircraft Co., 7755 E. Marginal Way; *for mail*, 4018—12th Ave., N.E., Seattle 5, Wash.
Wills, Saul D. (J'38) (EHM), 542 E. 50th St., Savannah, Ga.
Willsea, Jasper ('43) (CNG), Pres., Gen. Mgr., Willsea Works, 371 St. Paul St., Rochester 5, N.Y.
Willsey, John Crampton (J'41) (AMCK), Design Engr., Gen. Elec. Co., 1635 Broadway, Ft. Wayne 2; *for mail*, R.R. 1, Ashley, Ind.
Willson, David S. ('36) (JMN), Research Engr., John Wood Mfg. Co., Inc., Muskegon; *for mail*, 921 Maffett, Muskegon Heights, Mich.
Willson, Edwin Laurence ('41) (FS), Mgr., Indus. Sales, Lehigh Navigation Coal Co., Inc., 123 S. Broad St., Philadelphia 9; *for mail*, 4537 Spruce St., Philadelphia 39, Pa.
Willson, T. Edgar ('40) (CMDWKS), Ch. Engr., Gen. Mgr., Mundet Cork Corp., 501 Bloy St., Hillside 5; *for mail*, Box 7, Demarest, N.J.
Wilmer, John Whittingham ('39; '46) (ACEHRS), Employment Rep., Consltd. Gas, Elec. Light & Power Co. of Baltimore, Baltimore; *for mail*, Club & Ruxton Rds., Ruxton 4, Md.
Wilmot, Russell Cartwright ('17; '35) (CD), Gen. Elec. Co., Bridgeport 2; *for mail*, 55 Dora Circle, Bridgeport 4, Conn.
Wilms, Hermann ('14; '21; '35), Mech. Engr., Midwest Piping & Supply Co.; *for mail*, 527 Clara Ave., St. Louis, Mo.
Wilson, Alexander, III ('21) (FHS), Mgr., Transmission & Distribution Dept., Philadelphia Elec. Co., 1000 Chestnut St., Philadelphia 5, Pa.
Wilson, Alexander Hardie ('18; '35) (BCD), Ch. Engr., Thomas Hoist Co., 20 S. Hoyne Ave., Chicago 12; *for mail*, 1118 S. Home Ave., Oak Park, Ill.
Wilson, Benjamin J. ('21; '25; '29) (ZBEFHS), Ch., Mech. Engrg. Div., Research Dept., Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia 44; *for mail*, 224 Lafayette Ave., Oreland, Pa.
Wilson, Charles D. ('29; '41) (SNBKMJ), Engr., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 1509 S. 73rd St., West Allis 14, Wis.
Wilson, Chas. E. ('40), 7 Hampton Rd., Scarsdale, N.Y.
Wilson, Charles R. (J'32), 51 Mentelle Pk., Lexington, Ky.
Wilson, Chester W. ('18) (SEHO), Mech. Engr., Am. Smelting & Refining Co., 120 Broadway, New York 5, N.Y.
Wilson, Christian, Jr. ('19; '28) (CKLMS), Dist. Mgr., Yarnall-Waring Co., 506 Wabash Bldg., Pittsburgh 22; *for mail*, 513 Willow Dr., Pittsburgh 16, Pa.
Wilson, Clifford Marcell (A'45) (BCOJSM), Dist. Supvr., Otis Elev. Co., 2301 Locust St., St. Louis 3; *for mail*, 7449 Amherst Ave., University City 5, Mo.
Wilson, Davis Douglas ('39; '46) (MCLD), Ch. Prod. Engr., Hobbs Mfg. Co., 609 N. Main St., Ft. Worth; *for mail*, 3633 Hazeline Rd., Ft. Worth 3, Tex.
Wilson, Donald James (J'45), Jr. Engr., Bechtel McCone Corp., 220 Bush St., San Francisco 4; *for mail*, 449 Orange St., Oakland 10, Calif.
Wilson, Donald R. (J'37) (FKS), Babcock & Wilcox Co., Marquette Bldg., 140 S. Dearborn St., Chicago, Ill.
Wilson, Earl Lynn (J'44) (SBAEFN), Research Engr., Research Labs., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Wilson, Edmund Powel ('39) (JYMLDC), Matls. Engr., West. Elec. Co., Inc., 2500 Broening Highway, Baltimore; *for mail*, 115 Manor Ave., Overlea, Baltimore 6, Md.
Wilson, Edward Ewell ('45) (FSKZ), Ch. Engr., St. Joseph Light & Power Co., 520 Francis St., St. Joseph 2; *for mail*, 2929 Renick St., St. Joseph 52, Mo.
Wilson, Ernest Browning (J'42) (EFSCBK), Field Engr., Robison Mchy. Co., Salt Lake City; *for mail*, 343 S. 9th E. Rear, Salt Lake City 2, Utah.
Wilson, Frank William ('43) (NBLK), Asst. Prof. Mech. Engrg., Univ. of Mich.; *for mail*, 1217 Baldwin Ave., Ann Arbor, Mich.
Wilson, Frederick George ('24; '26; '35) (CMYJ), Mech. Engr., Waterbury Cos. Inc., 835 S. Main St., Waterbury 90, Conn.
Wilson, George P. ('27; '31; '35) (BER), 85 Maltbie Ave., Suffern, N.Y.
Wilson, George Samuel ('07; '14) (EFS), Prof. Mech. Engrg., Univ. of Wash., Seattle 5, Wash.
Wilson, Grover C. ('25) (EASFKB), Asst. Dir., Automotive & Aircraft Dept., Universal Oil Products Co., Riverside; *for mail*, 500 Sunset Ave., La Grange, Ill.
Wilson, Hamilton M. ('30) (FS), Pres., H. M. Wilson Co., 18th & Brandywine Sts., Philadelphia 30, Pa.

Wilson, Harold Andrew (J'39) (KH), Assoc. Prof. Mech. Engrg., Rensselaer Poly. Inst., Troy, N.Y.
Wilson, Harold Manning ('42) (ZMCJY), Engr., Gen. Radio Co., 275 Massachusetts Ave., Cambridge 39; *for mail*, 27 Taft Ave., Lexington 73, Mass.
Wilson, Harry Paul (J'39) (FS), Engr., United Light & Power Serv. Co., United Light Bldg., Davenport, Iowa.
Wilson, Harry R. (J'39) (NBJC), Engr., Frederick M. Conran Mch. Co., 107 Colden St., Newark 4; *for mail*, R.F.D. 1, Scotch Plains, N.J.
Wilson, Henry Dalzell ('03; '12) (A), Dir. & Mem. of Export Com., Aviation Corp., Consltd. Vultee Aircraft Corp., 821—1st Trust Bldg., Pasadena 1, Calif.
Wilson, Henry Martsolf ('41) (AFNOSY), Sales Engr., James H. Herron Co., 1360 W. 3rd St., Cleveland 13; *for mail*, 1826 Grasmere Ave., Cleveland 12, Ohio.
Wilson, Hubert Neville (J'43) (EHJ), Jr. Mech. Engr., Clinton Engr. Works, Tenn. Eastman Corp., Oak Ridge; *for mail*, 470 Scenic Dr., Knoxville, Tenn.
Wilson, Jacob Diller, Jr. ('42) (DCBHJ), Secy., Am. Elev. & Mch. Corp., 17 Vandewater St., New York 7; *for mail*, 1197 E. 19th St., Brooklyn 30, N.Y.
Wilson, Jacob Diller, Sr. ('42) (DBCOEJ), Pres., Treas., Am. Elev. & Mch. Corp., 17-27 Vandewater St., New York 7; *for mail*, Box 285, Greenport, L.I., N.Y.
Wilson, James ('13), Mgr., Steam Turbine Dept., Field Engrg. Div., Allis-Chalmers Mfg. Co., Milwaukee; *for mail*, 1512 S. 77th St., West Allis 14, Wis.
Wilson, James Alan (J'41) (SLEFH), Field Engr., Elliott Co., Inc., 616 McBirney Bldg., Tulsa 3, Okla.
Wilson, James Edwards (J'38) (SEMKWZ), Engr., Dallas Air Conditioning Co., 2809 Canton, Dallas 11, Tex.
Wilson, James Raymond, Jr. (J'46), 90 Tower Hill Rd., Mountain Lakes, N.J.
Wilson, John A., Jr. ('23; '33; '35) (HLW), Lt. Col., Highwood Ave., Tenafly, N.J.
Wilson, John Alexander ('45) (BCM), Factory Mgr., New England Butt Co., 304 Pearl St., Providence; *for mail*, 20 Griswold Ave., Cranston, R.I.
Wilson, John Crosier ('17) (SDY), 3060 N. Marietta Ave., Milwaukee 11, Wis.
Wilson, John Emerson ('20) (DFLMS), Gen. Supt., Swift & Co., Union Stock Yards; *for mail*, 6210 Ingleside Ave., Chicago, Ill.
Wilson, John Grier (J'44), 1129 Pennsylvania, Denver, Colo.
Wilson, John Page (J'44), 229 Herbert Ave., Fanwood, N.J.
Wilson, John Woodrow (J'45) (CADM), Indus. Engr., Ernst & Ernst, 50 Congress St., Boston 9; *for mail*, 234 Norfolk St., Wollaston 70, Mass.
Wilson, Joseph George (J'44) (HLP), Mech. Engr., Shell Devel. Co., 100 Bush St., San Francisco 6, Calif.
Wilson, Joseph Maple ('43) (ECARS), Mgr., Products Application Dept., Shell Oil Co., Inc., Box 262, Wood River, Ill.
Wilson, L. B. (J'40), Chem. Engr., Dowell Inc., Rm. 2020, 30 Rockefeller Plaza, New York 20, N.Y.
Wilson, Lawrence Orville (J'45), Ensign (E)L, U.S.N.R., 119—88th St., Brooklyn 9, N.Y.
Wilson, Leroy Alonzo ('17; '22) (ESAFK), Prof. Mech. Engrg., Chmn., Dept. of Mech. Engrg., Univ. of Wis., 1513 University Ave.; *for mail*, 3210 Oxford Rd., Madison 5, Wis.
Wilson, Lester Godfrey ('42) (AM), Lt. Col., Retired; 20 Magnolia Pkwy., Chevy Chase 15, Md.
Wilson, Lyndon Francis ('41) (KRS), Pres., Gen. Mgr., Wilson Engrg. Corp., 122 S. Michigan Ave., Chicago 3, Ill.
Wilson, Marshall A. ('30; '41) (SFK), Tech. Asst., Engrg. Dept., B. F. Goodrich Co., 500 S. Main St., Akron 18; *for mail*, Box 14, R.R. 3, 134 Park Dr., Cuyahoga Falls, Ohio.
Wilson, Ralph Lawrence ('38) (EJS), Ch. Metal. Engr., Timken Roller Bearing Co., Canton 6, Ohio.
Wilson, Richard Christian (J'43) (CM), Operas. Supvr., Carbide & Carbon Chemicals Co.; *for mail*, 244 Boone Hall, Oak Ridge, Tenn.
Wilson, Robert A. (J'40) (CM), Spec. Program Co-ordinator, Office of Inspc. of Naval Material, U.S.N., 744 N. 44th St., Milwaukee 8; *for mail*, 3238 N. 44th St., Milwaukee 10, Wis.
Wilson, Robert Alexander (J'42) (SER), Ensign, U.S.N.R.; Alexander Hamilton Hotel, San Francisco, Calif.
Wilson, Robert Clifford (J'41) (KCMSJ), Pur. Agt., Griscom-Russell Co.; *for mail*, 710 Milburn Rd., N.E., Massillon, Ohio.
Wilson, Rosser L. (J'37) (RAJNB), Asst. Ch. Engr., Am. Brake Shoe Co., Mahwah, N.J.
Wilson, Rushen Arnold (J'35) (SKEFL), Engr., Eng. & Condenser Dept., Allis-Chalmers Mfg. Co., Milwaukee 1; *for mail*, 3233 S. 21st St., Milwaukee 7, Wis.
Wilson, Russel Sidney (J'43) (ABE), Filer, Idaho.
Wilson, Thomas ('17) (SFNC), Mgr., Chicago Office, Engrg. Societies Personnel Serv., 211 W. Wacker Dr., Chicago 6; *for mail*, 6132 Greenwood Ave., Chicago 37, Ill.
Wilson, Thomas F. (J'38) (SFKELC), Combustion Engr., Babcock & Wilcox Co., 140 S. Dearborn St., Chicago 3, Ill.
Wilson, Thomas Woodrow (J'43) (NTA), Ch. Engr., Terrell Mch. Co., 1200 N. Church; *for mail*, 3018 Carol Ave., Charlotte 2, N.C.
Wilson, Warden F. (J'29) (JKELHR), Gen. Sales Mgr., Lebanon Steel Fdy., Lebanon, Pa.
Wilson, Warren Elvin ('45) (BHO), Chmn., Heat & Fluids Research, Armour Research Foundation, 35 W. 33rd St., Chicago; *for mail*, 1452 Scott Ave., Winnetka, Ill.
Wilson, Warren Marlin (J'44) (LRSTYO), Owner, Wilson Engrg. & Sales Co., 407 The Arcade, Cleveland 14, Ohio.
Wilson, William Alan ('43) (BENSKH), Mech. Div. Engr., Research Dept., Elliott Co., Jeannette, Pa.
Wilson, William Broyles ('40; '45) (SELFRK), Design Engr., Turbine Div., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, 8 Townsend Terrace, East Lynn, Mass.
Wilson, William Henry ('41) (HNS), 352 Derby Circle, Chattanooga 4, Tenn.
Wilson, William Horn (J'42) (ABEKJ), Engr., Charge of Design, Master Mfg. Co., 1300 E. Ave. A, Hutchinson, Kan.
Wilson, William R. (J'99), 61 Bennett Pl., Amityville, L.I., N.Y.
Wilson, William S. (J'37) (ERSAH), Engr., Tech. Serv. Div., Socony-Vacuum Oil Co., Inc., 412 Greenpoint Ave., Brooklyn 22; *for mail*, 3 Catherine Pl., Bellmore, N.Y.
Wilson, Wylie G. ('24) (BLNOY), Cons. Engr., 275 Morris Ave.; *for mail*, 27 Stiles St., Elizabeth 3, N.J.
Wilt, Homer Edison (J'42) (AYEJBS), Design & Constr. Engr., B. F. Goodrich Co., 500 S. Main St., Akron 14; *for mail*, 1345½ Manchester Rd., Akron 7, Ohio.
Wiltrout, Robert E., Jr. (J'40) (BCL), 913 Hamlin St., Akron, Ohio.
Wiman, Charles Deere ('43), Pres., Deere & Co., 1325—3rd Ave., Moline, Ill.
Wimborough, James R. (J'36) (CMAJEZ), Process Engr., Perfect Circle Co., S. Washington St., Hagerstown, Ind.
Winchester, Henry F. ('33; '35) (ELJ), Chem. Engr., De Laval Separator Co., Pine St.; *for mail*, 19 Cannon St., Poughkeepsie, N.Y.
Winchester, Marshall Hayden ('41) (SRFE), Asst. Supvg. Engr., Boiler & Mch. Div., Travelers Ins. Co., 700 Main St., Hartford; *for mail*, 592 Poquonock Ave., Windsor, Conn.
Windle, Arthur Elmer ('18; '22; '27) (CSDLAF), Rm. 248, U.S. Veterans Admin. Facility, Kecoughtan, Va.
Windstrup, Robert Franklin (J'46), 6340 S. Loomis Blvd., Chicago 36, Ill.
Winemiller, Dorse Edward (J'43) (BFKS), Supt. Power, Pittsburgh Plate Glass Co., Barberton, Ohio.
Wines, Harry Thomas (J'32) (NCMJDB), Ch. Engr., Waldes Koh-I-Noor Inc., 4752—27th St., Long Island City 1; *for mail*, 7527—184th St., Flushing, L.I., N.Y.
Wingate, Francis Baldwin (J'42) (CMNORS), Gen. Serv. Engr., 220—9th Court W., Birmingham 4, Ala.
Wingo, William Brown (J'32) (OWCD), Lt. Comdr., Civ. Engr. Corps, U.S.N.R., Pub. Works Dept., Naval Air Sta., Norfolk; *for mail*, Apt. 2, 512 Graydon Park, Norfolk 7, Va.
Wingren, Roy Mathew ('28; '38) (BKY), Prof., Mech. Engrg. Dept., Agric. & Mech. College of Tex., College Station, Tex.
Winholt, Einar A. ('28) (EFS), Supvg. Power Engr., Deere & Co.; *for mail*, 1815—25th Ave., Moline, Ill.
Winkler, Marshall Harry (J'42) (MLJCDH), 260 Jefferson Dr., Pittsburgh 16, Pa.
Winkler, Theodore Edward ('38; '46) (CFS), Sr. Assoc. Mech. Engr., Dept. of Pub. Works, 1550 Erskine St., Detroit; *for mail*, 12181 Cloverlawn, Detroit 4, Mich.
Winkler, William John (J'45), Test Engr., Graduate Trng. Program, Allis-Chalmers Mfg. Co., Milwaukee; *for mail*, 2725 N. Carlton Pl., Milwaukee 10, Wis.
Winne, Harry Alonzo ('37) (C), V.P., Charge Engrg. Policy, Gen. Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Winship, John Trimble (J'43) (AFEY), Aviation Electronics Technician's Mate, 3rd Class, U.S.N.R.; *for mail*, 3308 Beechwood Ave., Cleveland Heights 18, Ohio.
Winship, Walter Edwin ('08) (EFKRS), Pres., Winship Oils, Inc., 344 Camp St., New Orleans 12, La.
Winslow, Arthur M. ('21), Prof. Mech. Engrg., Univ. of Wash., Seattle, Wash.
Winslow, Pearson ('21; '35), The Beekman, 575 Park Ave., New York, N.Y.
Winslow, Richard Gilbert (J'43) (LFZK), Assoc. Process Engr., Borger Refinery, Phillips Petroleum Co., Borger; *for mail*, Box 285, Phillips, Tex.
Winslow, Seth L. ('43) (CODLMA), Sr. Engr., Mgmt. Consultant, McKinsey Kearney & Co., 135 S. La Salle St., Chicago 3, Ill.
Winslow, William Howard, Jr. (J'40) (KZSCHL), Engr., Stand. Oil Co. of N.J., Box 551; *for mail*, 1748 Stanford Ave., Baton Rouge 15, La.
Winsor, Allen P., Jr. (J'41) (BJS), Co-Partner, New Bedford Boiler & Mch. Co., 42 Front St., Box 650, New Bedford; *for mail*, North Ave., North Rochester, Mass.
Winston, Paul Raymond ('44) (SZKLFC), Ch., High Pressure Boiler Sub-Sec., 8th Serv. Comd., War Dept., Santa Fe Bldg., Dallas 2; *for mail*, Box 915, Route 9, Dallas 10, Tex.
Winston, Stanton E. ('39) (EFS), Assoc. Prof. Mech. Engrg., Ill. Inst. of Tech., 3300 Federal St., Chicago; *for mail*, 401 S. Quincy St., Hinsdale, Ill.
Winter, Frederick Charles (J'35) (CMLDJA), Maj., Ord. Dept.; *for mail*, c/o Albert Winterhalter, 1784 Westchester, New York 60, N.Y.
Winter, Otto William ('42) (MNJCOD), V.P., Acme Pattern & Mch. Co., 1559 Niagara, Buffalo 13; *for mail*, Winter Rd., Sandy Beach, Grand Island, N.Y.
Winter, Robert G. (J'42), 3816 Aldrich Ave. S., Minneapolis, Minn.
Winters, Francis Gray (J'44) (KRE), Student & Engrg. Research Asst., Univ. of Tex.; *for mail*, Box 235A, Prather Hall, Austin, Tex.
Winters, Ray ('46), Results Engr., Kan. City Power & Light Co., 1330 Baltimore, Kansas City 10, Mo.
Winters, Robert Keith (J'45) (AEP), Box 1792, Odessa, Tex.
Winther, Geo. S. ('23), Lone Star Cement Corp., 342 Madison Ave., New York, N.Y.
Winthorpe, James ('40) (SMCD), Prod. Supvr., Scheduling, Combustion Engrg. Co., W. Main St., Chattanooga 2; *for mail*, 7 S. Brooks Ave., Chattanooga 4, Tenn.
Winton, Lewis Bennett ('22; '35) (ONZSFB), Cons. Engr., R.F.D. 3, Greenwich, Conn.
Wintriss, George (J'40) (BCMNOY), Pres., Zipon, Inc.; *for mail*, 68 Hobart Ave., Summit, N.J.
Wintzer, Herman Charles ('35; '43) (BSL), Asst. Engr., Nitrogen Div., Solvay Process Co., Hopewell, Va.
Wintzer, Rudolph Collings (J'42) (KHSBEF), Staff Engr., Babcock & Wilcox Co., 85 Liberty St., New York 6, N.Y.
Wippermann, Erwin Henry, Jr. (J'43), Mech. Engr., Murray Mfg. Co., 1250 Atlantic Ave., Brooklyn; *for mail*, 109-18—111th St., Ozone Park, N.Y.
Wiren, Robert Charles ('26; '30; '35) (SKEFA), Assoc. Prof. Mech. Engrg., Univ. of Toronto, Toronto, Ont., Can.
Wirshing, Armando O. (J'34), Managing Partner, Wirshing & Co., S. en C., Mercedita; *for mail*, 25 Hostos Ave., Ponce, P.R.
Wirsing, Charles Edward (J'42) (BJM), 211 Belair Ave., Aberdeen, Md.
Wirtsen, Ernst ('23; '27) (ABDEKR), Engrg. Dept., Sun Shipbldg. & Dry Dock Co.; *for mail*, 2405 Chestnut St., Chester, Pa.
Wischmeyer, Carl ('13; '22) (SKFEA), V.P., Prof. Mech. Engrg., Rose Poly. Inst., Terre Haute, Ind.
Wise, Alfred S. ('19; '24; '35) (REHYSJ), Sales Engr., Baldwin Loco. Works, 120 Broadway, New York 5, N.Y.; *for mail*, 264 Park St., Upper Montclair, N.J.
Wise, Daniel Emmer (J'36) (SFK), Asst. Ch. Engr., Clairton Works, Carnegie-Ill. Steel Corp., State St., Clairton, Pa.
Wise, Eugene Paul (J'42) (AEFSZ), Test Engr., Chrysler Corp., 12800 Oakland, Highland Park 3; *for mail*, 650 W. Boston Blvd., Detroit 2, Mich.
Wise, Herbert Augustus, Jr. (J'42) (ENFAB), Test Engr., Superior Eng. Div., Natl. Supply Co.; *for mail*, 426 Sheridan Ave., Springfield, Ohio.
Wise, Irwin Lewis ('23) (SLFEOH), Cons. Engr., 105 W. Adams St., Chicago 3, Ill.
Wise, Karl Marshall ('43), Dir. of Engrg., Bendix Products Div., Bendix Aviation Corp., South Bend 20, Ind.
Wise, Max R. (J'38), Maj., 44th Armd. Regt., 12th Armd. Div., A.P.O. 262, c/o Postmaster, New York, N.Y.
Wise, Richard Tyler (J'39) (NOCY), Sr. Draftsman, Plant Design & Constr. Dept., B. F. Goodrich Co., 500 S. Main St., Akron; *for mail*, 2667—9th St., Cuyahoga Falls, Ohio.
Wise, Roy T. ('36) (ABM), Cons. Engr., Union Twist Drill Co., Athol, Mass.; *for mail*, Box 208, Newtown, Conn.
Wiseman, Elton J. ('27) (SHKBO), Mech. Engr., Stone & Webster Engrg. Corp., 49 Federal St., Boston, Mass.
Wiseman, John T. ('19; '22), Gen. Mgr., Bethlehem Steel Co., Howard St., Quincy 69, Mass.
Wislicenus, George F. ('30; '38) (AHEKBZ), Research Engr., Toledo Div., Packard Motor Car Co., Toledo 12; *for mail*, 3144 Cheltenham Rd., Toledo 6, Ohio.
Wisner, Henry G. ('22; '35) (SKFOH), Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Wisniewski, Chester (J'46), 3335 E. Milwaukee, Detroit 11, Mich.
Wisniewski, Edmund Joseph (J'45) (AEJ), 1320 Lincoln Ave., Prospect Park, Pa.
Wisniewski, Edward (J'37) (AJMCHL), 3335 E. Milwaukee, Detroit 11, Mich.
Wisnik, Jacob (J'45) (NJE), U.S. Navy, 72 Broad St., Keyport, N.J.

Wist, Edward B. ('35; '42) (BKLC), Engr., Shell Devel. Co., 4560 Horton St., Emeryville 8; *for mail,* 1216 Bay St., Alameda, Calif.
Withers, Cleeman ('26; '35) (ACDY), Box 535, R.F.D. 6, Tucson, Ariz.
Witherspoon, David Lane (J'39) (OAJLC), Mech. Engr., Reynolds Metals Co., Inc., 2000 S. 9th St., Louisville; *for mail,* Anchorage, Ky.
Withington, Sidney ('37) (RSFE), Elec. Engr., N.Y., New Haven, & Hartford R.R., New Haven 6, Conn.
Witmer, Frank P., Jr. (J'36) (SBNH), Mech. Designer, H. A. Kuljian & Co., 1518 Walnut St., Philadelphia 2; *for mail,* 738 Pennsylvania Ave., Prospect Park, Pa.
Witmer, Guy Willard (J'44) (BKSEZN), Asst. Ch. Engr., Savannah Elec. & Power Co., Riverside Sta.; *for mail,* 627 E. 40th St., Savannah, Ga.
Witmer, Stanley Gore (J'44) (MRC), Student Engr., Electro-Motive Div., Gen. Motors Corp., La Grange; *for mail,* 5303 Victor St., Downers Grove, Ill.
Witt, Joshua Chitwood ('22; F'44) (FSELBD), Tech. Dir., Marquette Cement Mfg. Co., 140 S. Dearborn St., Chicago 3, Ill.
Witt, Ted Rose (J'43) (NTB), Maint. Supvr., Tenn. Eastman Corp., Kingsport, Tenn.
Wittenberger, Wallace Wayne (J'42) (OSKCF), Rm. 367-W, Northwestern Tech. Inst., Evanston, Ill.
Witters, Ambrose Gloshen (J'43) (LESKO), Asst. Plant Engr., Lever Bros. Co., Indianapolis Blvd., Hammond, Ind.
Wittig, Carl O. G. ('39) (BSRHO), Assoc. Prof. Mech. Engrg., Catholic Univ. of Am., Washington, D.C.; *for mail,* 8705 Sundale Dr., Silver Spring, Md.
Wittig, Frederick Emory (J'36) (OWANMC), Ch. Engr., Air Conditioning & Refrigeration Installation Design, Automatic Cooling & Htg. Co., Inc., 289 Post Ave., Westbury; *for mail,* Box 145, Glenwood Landing, N.Y.
Wittman, Lawrence (J'42) (YMWCJO), Tool Devel. Engr., Republic Aviation Corp., Farmingdale; *for mail,* 76-66 Austin St., Forest Hills, L.I., N.Y.
Witty, James Cecil, II (J'46), 313 S. Lafayette Blvd., South Bend 10, Ind.
Wobensmith, Zachary Taylor, II (J'27) (ZSTD), Pat. Atty., 726 Land Title Bldg., Philadelphia 10, Pa.
Wochos, Wenzel M., Jr. (J'39) (BFS), Research Engr., Elgin Natl. Watch Co.; *for mail,* 628 St. John St., Elgin, Ill.
Wockenfuss, William ('40) (NBYZGH), Design Engr., Control Instrument Co., Inc., 67—35th St., Brooklyn 32, N.Y.; *for mail,* 1987 Oakwood Pkwy., Union, N.J.
Woehl, Frank ('44), 719 City View Ave., Akron 2, Ohio.
Woehl, Frank Martz (J'44) (BNCMKE), 719 City View Ave., Akron 2, Ohio.
Woelbing, George H. ('34) (BCJMN), Mgr., Mfg. Div., Silver Engrg. Works, Inc., 3309 Blake St., Denver 5, Colo.
Woersching, Thomas Beaumont (J'38) (AKBEFZ), Sr. Aerodynamicist, Glenn L. Martin Co., Middle River; *for mail,* 2709 Goodwood Rd., Baltimore 19, Md.
Woerwag, Carl A. ('20; '35) (DCLSEF), Export Mgr., Link-Belt Co., 233 Broadway, New York 7, N.Y.
Wognum, James N. (J'41) (BCM), 10157 Lowe Ave., Chicago, Ill.
Wohlenberg, Walter J. ('17; '25; F'42) (SKFBDH), *Vice-President,* '42-'44; Prof. Mech. Engrg., Yale Univ., 400 Temple St., New Haven 11, Conn.
Wohnus, Miss Hope Margaret (J'44) (AC), 320 Tolland Turnpike, Manchester, Conn.
Wohrley, Jay Raymond ('37) (CEFKJL), Engr., Gas Advisers Inc., 70 Pine St., New York, N.Y.
Wojtaszek, Victor Bernard (J'44), 96 E. 7th St., New York 3, N.Y.
Wokurka, Elmer Oliver ('25; '41) (BCFKMS), Maint. Engr., Combustion Engrg. Co., Inc., 5319 Shreve Ave., St. Louis 15, Mo.
Wolcott, Charles Gordon (J'42) (ACMBK), Aero. Engr. A, Consltd. Vultee Aircraft Corp., Lindbergh Field, San Diego; *for mail,* 4276 Hawk St., San Diego 3, Calif.
Wolf, Arnold M. (J'34) (CJMNOD), Works Mgr., Lewyt Corp., 60 Broadway, Brooklyn 11; *for mail,* 1411—45th St., Brooklyn 19, N.Y.
Wolf, C. Earl (J'43) (SLFDCH), Plant Engr., Pittsburgh Plate Glass Co.; *for mail,* 828—4th St., New Martinsville, W.Va.
Wolf, Clarence Franklin (J'19) (BDLNOW), Engr., Jacob Wolf Co., Bowers Bldg., Mansfield, Ohio.
Wolf, Edward, Jr. (J'42) (CAEMNO), Seaman, 2nd Class, U.S.N.R., 1095 Mt. Vernon Ave., Akron 10, Ohio.
Wolf, George Walter, Jr. (J'44) (ASDFKM), 1432 N. 3rd St., Logansport, Ind.
Wolf, Irwin (J'22) (MCDY), Prod. Mgr., Noma Elec. Corp., 55 W. 13th St., New York 11; *for mail,* 1663—52nd St., Brooklyn 4, N.Y.
Wolf, Julius ('20; '25; '35) (EFS), Coal Sales Mgr., Matl. Serv. Corp., 33 N. La Salle St., Chicago 2; *for mail,* 8238 Eberhart Ave., Chicago 19, Ill.
Wolf, Robt. B. ('13; F'36), *Vice-President,* '20-'22; Mgr., Pulp Div., Weyerhaeuser Timber Co., Longview, Wash.
Wolfe, Bernard J. ('25; '33; '35) (CZBMWT), Designing Engr., Exper. Engrg. Dept. 77, Camera Works, Eastman Kodak Co., Rochester 4; *for mail,* 69 Mayfair Dr., Rochester 5, N.Y.
Wolfe, Merritt Wendell (J'43) (SNKAOJ), Design Engr., Babcock & Wilcox Co.; *for mail,* 642 Wooster Rd., W., Barberton, Ohio.
Wolfe, Oscar ('31) (BEGH), Cons. Engr., Foreign Operas., Texas Co., Chrysler Bldg., 135 E. 42nd St., New York 17, N.Y.
Wolfe, Richard Claude ('40) (JMC), Cons. Engr., 1277 St. Ives Pl., Los Angeles 46, Calif.
Wolfe, Robert F. (J'40) (BMS), 118 Beaumont Ave., Catonsville 28, Md.
Wolfe, Thomas Francis ('26) (EHJ), Ch. Engr., Cast Iron Pipe Research Assn., 122 S. Michigan Ave., Chicago 3, Ill.
Wolfe, Townley Robey, III (J'46), Revere Copper & Brass Co., Baylis & Danville Sts., Baltimore; *for mail,* 842 N. Patterson Park Ave., Baltimore 5, Md.
Wolff, John F., Jr. (J'18) (OSLDEH), Cons. Engr., Cloverly Ave. & Leedam St., Jenkintown, Pa.
Wolff, Richard Alfred ('43) (LMFCW), Secy., Wolff & Munier, Inc., 222 E. 41st St., New York 17, N.Y.
Wolff, Robert Edward (J'37) (CMNZJ), Asst. to Sales Mgr., Ill. Tool Works, 2501 N. Keeler Ave., Chicago 39; *for mail,* 2315 Commonwealth Ave., Chicago 14, Ill.
Wolff, William Alvord, Jr. (J'40) (CMD), Indus. Engr., Columbia Steel Co.; *for mail,* 647 Central Ave., Pittsburg, Calif.
Wolff, Wirt McCoy (J'45) (ABO), 166 Colorado, Highland Park 3, Mich.
Wolin, Robert H. (J'37) (SFKBE), Proposition Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Woll, I. Edward (J'35), Lt., Army Air Forces, 4408 Eichelberger Ave., Dayton 5, Ohio.
Wollheim, Walter E. ('17; '20) (ECRO), Mgr., Indus. Div., Nathan Mfg. Co., 416 E. 106th St., New York 29, N.Y.
Wollin, Ernest ('32; '35), Glademore Court, 48th & Locust Sts., Philadelphia, Pa.
Wolosin, David (J'42) (BCM), Lt. (j.g.), U.S.N.R., Boston Gp., 16th Fleet, Bldg. 21, U.S. Naval Drydock, South Boston 10, Mass.
Wolowicz, Chester Henry (J'37) (ABNMCG), Asst. Prof. Mech. Engrg., Northeast. Univ., 360 Huntington Ave.; *for mail,* 210 Beacon St., Boston, Mass.
Wolpers, Henry M. (J'41) (AJH), Ensign, U.S.N.R.; *for mail,* c/o *Am. Republic,* Poplar Bluff, Mo.
Wolpert, Philip J. (J'37) (SFK), Asst. Results Engr., Ohio Power Co., Philo, Ohio.
Wolsdorf, Henry Adolph ('19; '25; 35) (DCWLBY), Dir., Sales Research, Package Research Lab.; *for mail,* 78 Ogden Ave., Rockaway, N.J.
Wolverton, William B. (J'39) (BCR), 819 S. Washington St., Hinsdale, Ill.
Womack, Herschal Sidney (J'45) (BCS), Millwright Gen. Foreman, U.S. Cartridge Co.; *for mail,* 5241 Greer Ave., St. Louis, Mo.
Wommack, Kenneth Lamar (J'42) (KZBCJA), D.C. Motor Engrg., Westinghouse Elec. Corp., East Pittsburgh, Pa.
Wondisford, Floyd Edward (J'42), 163 Simmons Ave., Cohoes, N.Y.
Wong, Ging Ching (J'44), 956 S. San Pedro St., Los Angeles 15, Calif.
Wong, Gumm Duck (J'43) (EKS), 246 S. Main St., Greensburg, Pa.
Wong, Park Ying (J'43) (ASKMB), Mech. Engr., Ames Aero. Lab., Natl. Advis. Com. for Aero., Moffett Field; *for mail,* 1346 Mason St., San Francisco 11, Calif.
Wong, Richard (J'44) (AES), 137 E. 30th St., Los Angeles, Calif.
Wong Wen-hao (H'45), V.P., Exec. Yuan, Ministry of Economic Affairs, Chungking, China.
Wong, Yut Wee (J'45) (JMH), Spec. Student, Cincinnati Milling Mch. Co., Cincinnati; *for mail,* 2368 Victory Blvd., Cincinnati 6, Ohio.
Woo, Ji Yah (J'44) (EKM), 608 E. Madison St., Ann Arbor, Mich.
Wood, Alan Arthur ('14; '26) (SKHLCF), Partner, Luther & Wood, 1427 Eye St., Washington 5, D.C.; *for mail,* 4829 Drummond Ave., West Chevy Chase 15, Md.
Wood, Albert Carroll ('94; '00; '04; F'43) (OSFKD), Cons. Engr., Prin., Albert C. Wood & Associates, 1411 Walnut St., Philadelphia 2, Pa.
Wood, Andrew Gordon (J'43), 181 Cunningham St., Vallejo, Calif.
Wood, Arthur Leslie (J'45) (BLM), Ch. Tool Designer, Fuze Factory, Ferranti Ltd., Moston, Manchester; *for mail,* 41 Gainsborough Ave., Oldham, Lancashire, England.
Wood, Augustus ('01) (JM), Retired; 19 N. D St., Hamilton, Ohio.
Wood, Benjamin Franklin ('97; '07) (CORS), Pres., Stevens & Wood, Inc., 30 Broad St., New York 4, N.Y.
Wood, Carl Olen (J'39) (KNHJ), Engr., Devel. Dept., Carrier Corp., 300 S. Geddes St., Syracuse; *for mail,* 200 Shaver Ave., North Syracuse, N.Y.
Wood, Charles Estes ('16; '23) (CSAJEM), Asst. to Pres., Lunkenheimer Co., Beekman St., Cincinnati 14; *for mail,* 3636 Dickens Ave., Cincinnati 13, Ohio.
Wood, Charles Rowland (J'43) (AMCD), Jr. Engr., Curtiss Propeller Div., Curtiss-Wright Corp., Caldwell; *for mail,* 96 Elm St., Montclair, N.J.
Wood, Daniel Byrd, Jr. (J'41) (NOKBE), Olney, Tex.
Wood, David S. (J'41) (BJN), Research Asst., Calif. Inst. of Tech., 1201 E. California St., Pasadena 4, Calif.
Wood, Ernest P. (J'38) (EHS), Stevens & Wood, Inc., 30 Broad St., New York, N.Y.
Wood, Erwin Elmer ('00) (M), Pres., Treas., E. E. Wood Mchy. Co., 2832 E. Grand Blvd., Detroit 11, Mich.
Wood, Frank William (J'42) (ACEFHN), Engr., Oil Field, Gen. Petroleum Corp., 108 W. 2nd St., Los Angeles; *for mail,* 5937 Templeton St., Huntington Park, Calif.
Wood, Fredric Elmer ('44), Devel. Engr., Automatic Elec. Co., 1033 W. Van Buren St., Chicago, Ill.
Wood, Fremont E. ('41) (CHM), Factory Mgr., Eleam & Wood Mfg. Co., 3584 Cedar Ave., Long Beach 7, Calif.
Wood, Harlan Valentine ('40) (SEB), Turbine Engr., Fidelity & Casualty Co. of N.Y., 1822—1st Natl. Bldg., Oklahoma City 2, Okla.
Wood, Henry Hale ('28) (JBP), Dist. Mgr., Sales Engrg., Morgan Constr. Co., 2815 Koppers Bldg., Pittsburgh 19; *for mail,* 208 Hoodridge Dr., Pittsburgh 16, Pa.
Wood, Herbert Binns ('21; '35) (SEFK), Power Engr., E. I. du Pont de Nemours & Co., Market St., Wilmington 98; *for mail,* 199 Brandywine Blvd., Wilmington 274, Del.
Wood, Howard Ogden, III (J'43) (C), Ensign, U.S.N.R., N.Y. Naval Shipyard, Brooklyn; *for mail,* 11 Burton Ave., Woodmere, L.I., N.Y.
Wood, Iver C. ('23; '34) (CDJFZB), Factory Supt., Denver Fire Clay Co., Box 5510, Denver 17, Colo.
Wood, John Bruce (J'44) (ABE), Project Engr., Pilotless Aircraft, Air Tech. Serv. Comd., Wright Field, Dayton; *for mail,* 620½ Shroyer Rd., Dayton 9, Ohio.
Wood, John Morgan ('17; '22; '27) (MJNYAE), Owner, Albany Hardware Specialty Mfg. Co., 536-38 Water St.; *for mail,* Box 312, Albany, Wis.
Wood, John Thomas (J'32) (SENBF), Mech. Draftsman, Max Ams Mch. Co., Scofield Ave., Bridgeport; *for mail,* E. State St. & Rayfield, Westport, Conn.
Wood, John Wright (J'40) (KAH), Capt., Army Air Forces, 33rd Base Unit (Flight Serv.), Washington 25, D.C.
Wood, Joseph Kaye ('23; '35) (BMRSJN), V.P., Ch. Engr., Gen. Spring Corp., 500—5th Ave., New York 18, N.Y.; *for mail,* "Woodville," Westover Park, Stamford, Conn.
Wood, Joseph Leonard ('44) (RJWY), Asst. V.P., Am. Car & Fdy. Co., 30 Church St., New York 8, N.Y.
Wood, Karl Dawson ('38) (AKHBC), Prof. Aero. Engrg., Aero. & Mech. Engrg. Dept., Univ. of Colo., Boulder 7, Colo.
Wood, Kendall Musgrave (J'45) (HKO), Engr., DeWitt C. Griffin & Associates, 1101 Old Natl. Bank Bldg.; *for mail,* N 2519 Vista, Route 9, Spokane, Wash.
Wood, O. Wendell (J'44), Cons. Engr., 416 Madison Ave., Rahway, N.J.
Wood, Orla Leslie, Jr. (J'24) (SFBKJ), Engr., Turbine Engrg. Dept., Gen. Elec. Co., 1 River Rd., Schenectady 5; *for mail,* 1338 Dean St., Schenectady 8, N.Y.
Wood, Orville Tunstall ('46) (CJL), Ch. Engr., Blake Mfg. Corp.; *for mail,* 248 Chestnut St., Clinton, Mass.
Wood, R. Lee (J'43) (ONEJH), Field Engr., Hodson Corp., 5301 W. 66th St., Chicago 38; *for mail,* 7033 W. 13th St., Berwyn, Ill.
Wood, Raymond Vincent ('44) (FHOSTZ), Cons. Engr., Henry H. Fales, Cons. Engrs., 408 Grosvenor Bldg., Providence 3, R.I.
Wood, Richard Leroy (J'33) (NBZJSC), Jr. Engrg. Designer, Eastman Kodak Co., State St., Rochester; *for mail,* 142 Ridgedale Circle, Rochester 13, N.Y.
Wood, Russell Houghton (J'35) (BZJHWY), Research Engr., Norton Co., 1 New Bond St.; *for mail,* 44 Fair Haven Rd., Worcester 6, Mass.
Wood, Stanley V. ('13; '35), 334 S. Franklin St., Wilkes-Barre, Pa.
Wood, Theodore Evan (J'38) (ESMJKN), Research Engr., Gen. Mchy. Corp., 545 N. 3rd St.; *for mail,* Box 521, Hamilton, Ohio.
Wood, Thomas Johnson ('26; '35) (YCMLG), Supvr., Lamp Factory Equip., Lamp Mfg. Dept., Gen. Elec. Co., Nela Park, Cleveland; *for mail,* 16164 Glynn Rd., East Cleveland 12, Ohio.
Wood, Wallace Dean ('38; '44) (CKLZ), Dir., Stands. Lab., Taylor Instrument Co., 95 Ames St., Rochester 11, N.Y.
Wood, William Burton (J'42) (S), Asst. Engr., Land Power Engrg. Div., Steam Turbine Dept., Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.; *for mail,* 216 Peyton St., Geneva, Ill.
Wood, Winthrop Chace ('44) (ABTYWE), Application Engr., Westinghouse Elec. Corp., 1180 Raymond Blvd., Newark 2, N.J.
Woodburn, James (J'38), Md. Hall, Johns Hopkins Univ., Baltimore 18, Md.

Woodbury, Gale Franklyn (J'44) (NBDYMS), Mech. Engr., Plaskon Div., Libbey-Owens-Ford Glass Co., 2112-24 Sylvan Ave.; *for mail*, 919 Grand Ave., Toledo 6, Ohio.
Woodbury, Glen Pride (J'41) (NMBCOJ), Ch. Designer, Stapling Mchs. Co., Rockaway, N.J.
Woodbury, Roderick Vincent (J'45) (NGA), Drafting, B. F. Sturtevant Co., Div. of Westinghouse Elec. Corp., Damon St., Hyde Park; *for mail*, 5 Washington Court, Neponset 22, Mass.
Woodcock, Robert Reed (J'43) (AEF), 16 Roosevelt Rd., Weymouth, Mass.
Woodfield, Gene L. (J'38) (EKS), 509 Victoria Court, San Leandro, Calif.
Woodford, Philip Stanley (J'43) (ABE), Asst., Div. of Engrg., Brown Univ., Providence 12, R.I.
Woodger, George E. (J'39) (AEF), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford, Conn.; *for mail*, Granville, Mass.
Woodhouse, Harold William ('45) (ABSENM), Dir. of Devel., Stratos Corp., 122 W. Main St., Babylon, L.I., N.Y.; *for mail*, R.D. 2, Morrisville, Pa.
Woodhull, Albert S. (J'45) (KJAEBM), Research Engr., Heat Exchanger Div., Clifford Mfg. Co., 546 E. 1st St., South Boston; *for mail*, 152 Walnut St., Natick, Mass.
Woodman, Walter Cade ('29; '36) (SBKJHF), Stone & Webster Engrg. Corp., 49 Federal St., Boston; *for mail*, 367 School St., Watertown, Mass.
Woodnorth, Paul T. ('23; '35) (SFLCE), Cons. Power Engr., Sherwin-Williams Co., 115th St. & Cottage Grove Ave., Chicago 28; *for mail*, 7204 Bennett Ave., Chicago 49, Ill.
Woodroffe, George Henry ('16) (RBJZ), Engr., Baldwin Loco. Works, Paschall P.O., Philadelphia 42, Pa.
Woodruff, Clarence A. ('42) (CM), Deputy Regional Dir., War Prod. Bd., 119 Ann St., Hartford 4; *for mail*, S. Main St., East Hampton, Conn.
Woodruff, De Forest Douglas (J'31) (AMW), Sr. Draftsman, Glenn L. Martin Co., Baltimore 3; *for mail*, 4403 Wentworth Rd., Baltimore 7, Md.
Woodruff, Everett B. ('38) (CFZASL), Asst. Power Supvr., Wright Aero. Corp., Lockland; *for mail*, 1605 N. Dixon Circle, Cincinnati 24, Ohio.
Woodruff, Paul Alma (J'42) (ABCNJG), Design Engr., Stokermatic Div., Rheem Mfg. Co., 1415 S. State St., Salt Lake City; *for mail*, 1425 Zenith Ave., Salt Lake City 5, Utah.
Woodruff, Sturtevant ('42) (SFCM), Engr., Combustion Engrg. Co., Inc., 200 Madison Ave., New York 16, N.Y.
Woods, Baldwin Munger ('27) (BAO), Prof. Mech. Engrg., Dir. of Univ. Ext., Univ. of Calif., Berkeley 4, Calif.
Woods, George R. ('16; '25) (CEJM), Gen. Mgr., R. S. Stokvis & Sons, Inc., 17 Battery Pl., New York 4, N.Y.; *for mail*, Mendham Rd., Morristown, N.J.
Woods, Gordon Keaton (J'42) (MJBCHY), Mar. Engr., Div. Ch., Kaiser Co., Inc., Box 1632, Richmond; *for mail*, 847 Gelston Pl., El Cerrito, Calif.
Woods, Paul H. ('42) (LCO), Ch. Engr., Tide Water Associated Oil Co., Associated, Calif.
Woods, Peter Holderness (J'41) (GCMAR), Editor, Indus. Publ., Bucyrus-Erie Co., South Milwaukee, Wis.
Woods, Robert Alexander (J'38) (BMR), Engr., Design, Union Switch & Sig. Co., 1708 Braddock St., Swissvale; *for mail*, Zieger Ave., Box 430, R.D. 1, Verona, Pa.
Woods, Robert Henry (J'46), Mech. Engr., Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7, Wis.
Woods, Samuel Hamilton ('41) (EBKYH), Asst. Ch. Engr., Automotive Div., Ord. Research & Devel. Center, Aberdeen Proving Ground, Md.
Woods, William Blakeney, Jr. (J'37) (TYC), Gordon Mackay & Co., Ltd., 48 Front St.; *for mail*, 69 Wychwood Pk., Toronto, Ont., Can.
Woodson, Riley Delmer ('35; '41) (OSECH), Engr., Black & Veatch, Cons. Engrs., 4706 Broadway; *for mail*, 1800 W. 50th Terrace, Kansas City 2, Mo.
Woodward, Alan Arthur ('41) (CEFJKS), Asst. Supt., Steam Prod., Pub. Serv. Co. of Colo., 900—15th St., Denver 2; *for mail*, 2339 Bellaire St., Denver 7, Colo.
Woodward, Albert Eggleston (J'42) (BEM), Box 831, Borger, Tex.
Woodward, Arthur H. ('13), Pres., Internatl. Register Co., 2620 W. Washington Blvd., Chicago, Ill.
Woodward, Arthur Jason ('31) (RBKC), Engr., Loco. Engrg. Div., Gen. Elec. Co., 2901 E. Lake Rd.; *for mail*, 504 Kahkwa Blvd., Erie, Pa.
Woodward, E. L. ('20; '39) (REJ), West. Mech. Editor, *Railway Age*, 105 W. Adams St., Chicago 3, Ill.
Woodward, Hiram Woods ('16; '23; '35) (CONDYK), Cons. Engr., 812 Bellemore Rd., Baltimore 10, Md.
Woodward, John A. ('30), Cons. Engr., Woodward & Co., 101 E. Park Ave., Columbiana, Ohio.
Woodward, Sherman Melville ('07) (HBC), Cons. Planning Engr., Tenn. Valley Authority, 503 Union Bldg., Knoxville, 10, Tenn.
Woodward, Thomas Frederick ('45) (SCZKHF), Ch. Engr., Power Plants, Packard Motor Car Co., 1580 E. Grand Blvd., Detroit 32; *for mail*, 12710 Santa Rosa, Detroit 4, Mich.
Woody, Leonard (J'40), Engr., Draftsman, Continental Gin Co.; *for mail*, 631 S. 46th St., Birmingham, Ala.
Wool, William Lewis, Jr. (J'44) (BCM), Tech. Asst., Research, The Franklin Inst., 20th & Parkway, Philadelphia; *for mail*, 66 W. Plumstead Ave., Lansdowne, Pa.
Wooler, Ernest ('35) (EC), Ch. Engr., Bower Roller Bearing Co., 3040 Hart Ave., Detroit 14, Mich.
Woollard, Donald E. (J'40) (BKS), 6859 Olcott Ave., Chicago 31, Ill.
Woolley, Charles Henry ('43) (SKF), Tech. Supvr., Babcock & Wilcox Co., 85 Liberty St., New York, N.Y.; *for mail*, 5 Bluff St., Cranford, N.J.
Woolley, Harold Oakley ('09; '15; '35) (SFK), Ch. Engr., Steam Div., Foster Wheeler Corp., 165 Broadway, New York 6, N.Y.
Woolley, Harold Oakley, Jr. (J'41) (BZA), Devel. Engr., Instrument Div., Wallace & Tiernan Products, Inc., 1 Main St., Belleville; *for mail*, 74 Old Short Hills Rd., Millburn, N.J.
Woolley, Paul O. (J'24) (FJLWZ), Sales Engr., Am. Lurgi Corp., 120 Broadway, New York 5; *for mail*, 193-10—42nd Ave., Flushing, L.I., N.Y.
Woolley, Rew E. ('19) (CFHLSZ), V.P., Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio.
Woolrich, Willis Raymond ('19; '23; F'45) (LCOSKM), *Manager*, '38-'41, *Vice-President*, '41-'43; Dean of Engrg., Dir., Bur. of Engrg. Research, Univ. of Tex., Austin 12, Tex.
Woolsey, William S. (J'40), Aviation Cadet, Class 42-5, Aviation Cadet Det., Chanute Field, Ill.
Woolslayer, Homer Joseph ('45), Ch. Engr., Lee C. Moore & Co., Inc., 412 Philtower Bldg., Tulsa 2; *for mail*, 2698 E. 37th St., Tulsa 15, Okla.
Woolson, Clifford Griggs ('16) (CFSJYN), Mech. Engr., Mahanoy Sales & Engrg. Co. Inc., 790 Broad St., Newark 2; *for mail*, 294 Montgomery St., Bloomfield, N.J.
Woolson, Harry Thurber ('07) (BCEJN), Exec. Engr., Chrysler Corp., Oakland Ave., Detroit 31, Mich.
Worcester, Henry E. (A'03), V.P., United Fruit Co., V.P., Revere Sugar Refinery, 1 Federal St., Boston; *for mail*, 111 Church St., Winchester, Mass.
Worcester, Winthrop Sargent (J'43) (ANMWOC), Exper. Mechanic, Piper Aircraft Corp., E. Bald Eagle, Lock Haven, Pa.
Worden, Edwin S., Jr. (J'38) (CHW), 105 General Heath Ave., White Plains, N.Y.
Worden, William Henry ('45) (CDKLOY), Spec. Project Engr., Tenn. Eastman Corp., Kingsport, Tenn.
Work, R. P. (J'35) (CEH), Rate Engr., So. Counties Gas Co., 810 S. Flower St., Los Angeles 14, Calif.
Worley, Robert William ('38) (SOCFEM), Power Engr., United Engrs. & Constructors, Inc., 1401 Arch St., Philadelphia 5; *for mail*, 410 Brookside Ave., Wayne, Pa.
Worley, Will J. (J'43) (BJHYAR), Instr. in Theoretical & Applied Mechanics, Univ. of Ill., Urbana; *for mail*, R.R. 1, Sidney, Ill.
Wormser, Arthur ('40) (CGM), Ch. Offset Press Engr., Miehle Ptg. Press & Mfg. Co., 2011 W. Hastings St., Chicago 8; *for mail*, 474 Kent Rd., Riverside, Ill.
Wormser, Eric M. (J'42) (EMZC), Test Engr., Universal Camera Corp., 28 W. 23rd St.; *for mail*, 675 West End Ave., New York, N.Y.
Wormser, Hans Herman (J'44), 2nd Lt., U.S.A., 65th Army Postal Unit, A.P.O. 65, c/o Postmaster, New York, N.Y.
Worters, Charles Ralph ('44) (CDM), Staff Scheduler, Automatic Elec. Co., 1033 W. Van Buren St., Chicago 7, Ill.
Worth, Daniel Barnes ('21; '37) (EBNRCM), Asst. Ch. Engr., Cummins Eng. Co., Inc., 1138—5th St.; for mail, 214—19th St., Columbus, Ind.
Worth, Eugene Barnhardt ('40; '45) (CEFSW), Asst. Ch. Engr., Mengel Co., 1122 Dumesnil St., Louisville 1, Ky.
Worthington, Charles G. ('26; '37) (COMTA), Secy., Treas., Indus. Research Inst., Inc., 60 E. 42nd St., New York 17, N.Y.
Worthington, Emory Warren ('44; '45) (GCNO), Ch. Devel. Engr., Goss Ptg. Press Co., 1535 S. Paulina St., Chicago 8, Ill.
Worthington, John A. ('35; '35) (ERS), Mgr., Indus. Sales & Engrg., Am. Hammered Piston Ring Div., Koppers Co., Inc., Box 626, Bush & Hamburg Sts., Baltimore 3, Md.
Worthley, Leighton E., Jr. (J'42), Pvt., Air Force Enlisted Reserve; *for mail*, 1012 W. 4th St., Stillwater, Okla.
Wosak, Robert ('39) (BEMN), Devel. Engr., Elec. Boat Co., Groton; *for mail*, Tanglewyld Pl., Quaker Hill, Conn.
Wosnitzer, John (J'39), 1729—67th St., Brooklyn, N.Y.
Wottrich, Herbert (J'23) (CSGTY), Managerial Asst., Indus. Relations, Pub. Serv. Elec. & Gas Co., 80 Park Pl., Newark 1, N.J.
Woynar, Basil ('45) (BKS), Squad Leader, Design Dept., Superheater Co., 151st & Railroad Ave., East Chicago, Ind.
Wraith, William ('03) (EFJKS), Retired; Lafayette Hotel, Long Beach 2, Calif.
Wrape, William Robert, II (J'46) (OSW), Ensign, U.S.N.R., 2021 State St., Little Rock, Ark.
Wray, Edward ('44) (R), Publisher, *Railway Purchases and Stores*, 9 S. Clinton St., Chicago 6, Ill.
Wright, Allen Vinson (J'43), Sales Engr., City Coal & Coke Co., 800 Ewing Ave., Nashville 2; *for mail*, 1104 Graybar Lane, Nashville 4, Tenn.
Wright, Burnett Wilford, Jr. (J'44) (BDEMTZ), Naval Aviation Tech. Trng. Center, Ward Island, Corpus Christi, Tex.; *for mail*, West Tyne Dr., Nashville 5, Tenn.
Wright, Charles Albert (J'39) (ABCMKH), Engrg. Designer, Lockheed Aircraft Corp., Burbank; *for mail*, 11572 Riverside Dr., North Hollywood, Calif.
Wright, Charles M. (J'42), Sales Engr., Steel Cars, Bethlehem Steel Co.; *for mail*, 510 High St., Bethlehem, Pa.
Wright, Chester Stephen (J'42) (CM), Indus. Engr., Cent. Scien. Co., 1700 Irving Park Rd., Chicago; *for mail*, 2430 W. Cullom Ave., Chicago 18, Ill.
Wright, Clark McBride ('42) (SEK), Steam Turbine Specialist, Gen. Elec. Co., 840 S. Canal St., Chicago 80, Ill.
Wright, Daniel Knight ('20) (BJZ), Research Engr., Lamp Dept., Gen. Elec. Co., Nela Park, East Cleveland 12, Ohio.
Wright, Daniel Knight, Jr. (J'43) (NBJOMT), Asst. Prof. Mech. Engrg., Case Sch. of Applied Sci., 10900 Euclid Ave., Cleveland 6; *for mail*, 3618 Lindholm Rd., Shaker Heights 20, Ohio.
Wright, Donald Collier ('17; '19) (COM), Supvr. Operas., Trundle Engrg. Co., 208 S. La Salle St., Chicago 4; *for mail*, 3425 Wenonah Ave., Berwyn, Ill.
Wright, Elliott F. ('35; '41) (HSE), Asst. Ch. Engr., Reciprocating Pump Engrg. Div., Worthington Pump & Mchy. Corp., Harrison; *for mail*, Tall Oaks, Summit, N.J.
Wright, Ernest Neall ('90) (W), Retired; 619 Drexel Pl., Pasadena 2, Calif.
Wright, Harold Marshall (J'39) (EFHS), Lt. Comdr., U.S.N.R., Trng. Div., Bur. of Naval Personnel, Navy Dept., Washington, D.C.
Wright, Henry Tennant (J'42) (AJNG), Sr. Test Engr., Scintilla Magneto Div., Bendix Aviation Corp.; *for mail*, 80 E. Main St., Sidney, N.Y.
Wright, Hugh Williams (J'39) (BAEKSN), Test Sec. Head, Wheel & Brake Div., Goodyear Aircraft Corp., 1210 Massillon Rd., Akron 15; *for mail*, 847 Valdes Ave., Akron 2, Ohio.
Wright, James C. ('20; '35) (ACE), Pub. Works Inspr., U.S.N., Foot of Broadway, San Diego; *for mail*, Box K, 701 J Ave., Coronado, Calif.
Wright, Justin P. (J'44) (BCM), Supvr. Time Stands., Jones & Lamson Mch. Co., Clinton St.; *for mail*, 55 Elm St., Springfield, Vt.
Wright, L. Kay ('28) (BGM), 4161—54th St., Woodside, N.Y.
Wright, Lawrence T., Jr. (J'38), Cornell Univ., 4 W. Sibley, Ithaca, N.Y.
Wright, Leslie Austin ('39) (CMG), Gen. Secy., Engineering Institute of Canada, 2050 Mansfield St., Montreal 2, Que., Can.
Wright, Merrill C. (J'37) (CDM), 1st Lt., Army Air Forces; *for mail*, 502 Jefferson Ave., East Point, Ga.
Wright, Morris Jesse (J'45) (CJK), U.S.N.R., Box 826, Pasadena, Tex.
Wright, Orville (H'18), Pioneer Aviator, N. Broadway, Dayton, Ohio.
Wright, P. Kennard (J'40) (J), Millwright Turn Foreman, Bethlehem Steel Co., Baltimore 19; *for mail*, 3012 Dunmurry Rd., Baltimore 22, Md.
Wright, Paul ('15; '15; F'36) (CDFKOS), *Manager*, '24-'27, *Vice-President*, '27-'29; Owner, Paul Wright & Co., 244 Brown-Marx Bldg., Birmingham 3, Ala.
Wright, Paul Daniels ('17; '31; '35) (C), Pur., U.S. Rubber Co., 1230—6th Ave., New York 20; *for mail*, 214 E. 239th St., New York 66, N.Y.
Wright, Ralph Haven ('29) (AJS), Steel Mill Engr., Westinghouse Elec. Corp., East Pittsburgh; *for mail*, 164 Lloyd Ave., Pittsburgh 18, Pa.
Wright, Robert Ernest ('33) (EJKN), Asst. Planning Officer, Mchy., Naval Shipyard, Terminal Island, San Pedro; *for mail*, 540 W. Alegria Ave., Sierra Madre, Calif.
Wright, Robert Ernest (J'40) (LCKB), Project Engr., Design Dept., Monsanto Chem. Co., 1700 S. 2nd St., St. Louis 4; *for mail*, 1211 W. Big Bend Rd., Kirkwood 22, Mo.
Wright, Robert Fraser (J'44) (ACJ), 84 Rhodes Ave., Akron, Ohio.
Wright, Roger Vern (J'41) (FEHBL), Mech. Engr., Stand. Oil Co. of Ohio, Midland Bldg., Cleveland 15; *for mail*, 3415 Henderson Rd., Cleveland Heights 12, Ohio.
Wright, Roy V. ('07; F'36; 'H'43) (CR), *Manager*, '22-'25, *Vice-President*, '25-'27, *President*, '31; V.P., Secy., Simmons-Boardman Publ. Corp., 30 Church St., New York 7, N.Y.; *for mail*, 398 N. Walnut St., East Orange, N.J.
Wright, Stanley ('41) (ESK), Field Engr., Fairbanks, Morse & Co., 80 Broad St., New York 4, N.Y.; *for mail*, 450 Ridgewood Rd., Maplewood, N.J.

Wright, Stephen Conrad, Jr. ('44) (ACZMNR), Ch. Mech. Engr., Am. Time Products, Inc., 580—5th Ave., New York 19; *for mail*, 320 E. 42nd St., New York 17, N.Y.
Wright, Thomas Elmo (J'43) (SKTMLJ), Lt. (j.g.), U.S.N.R., U.S.S. *Rigel*, F.P.O., San Francisco, Calif.
Wright, William Newton (J'46) (ACH), Eclipse-Pioneer Div., Bendix Aviation Corps., Teterboro; *for mail*, Box 465, Mountain View, N.J.
Wright, William Quinby ('08) (OHSCDE), Cons. Engr., 1340 Lombard St., San Francisco 9, Calif.
Wrightman, Stanley Augustus ('43) (DHJMNS), Asst. Plant Engr., RCA Victor Div., Radio Corp of Am., 501 N. La Salle St., Indianapolis; *for mail*, 5139 University Ave., Indianapolis 1, Ind.
Wrigley, Clifford C. (J'35), 1625 Ford Court, Grosse Pointe Woods 30, Mich.
Wrock, Francis William (J'42) (CMADBH), 1687 N. Raymond, Pasadena 3, Calif.
Wuertele, Louis Meyer (J'40) (MCA), U.S. Maritime Comm., Commerce Bldg., Washington, D.C.
Wuest, Frank Wagner ('44) (CFJLS), Supt. Power & Maint., E. I. du Pont de Nemours & Co., Barksdale, Wis.
Wuest, William David (J'39) (CDJMNZ), Gen. Mgr., Wuest Bros., Inc., 936 W. Hill St.; *for mail*, 1415 Homeview Dr., Louisville 8, Ky.
Wulf, Hurst Ernest John (J'45) (CDO), Engr., Plant 3, Bohn Aluminum & Brass Corp., 1400 Lafayette Bldg., Detroit 26; *for mail*, 15000 Rosemont Blvd., Detroit 23, Mich.
Wunderlich, Milton Sidney ('20; '26) (CWLYZN), Dir. of Research, Minn. & Ont. Paper Co., 500 Baker Arcade, Minneapolis 2; *for mail*, 545 Mt. Curve Blvd., St. Paul 5, Minn.
Wunsch, Jos. W. ('17; '23; '27) (CDM), Pres., Ch. Engr., Silent Hoist Winch & Crane Co., 841—63rd St., Brooklyn, N.Y.
Wurdack, Hugo ('14), E-20 Ry. Exch. Bldg., St. Louis, Mo.
Wurster, William F. ('21; '35) (KLS), Sales Engr., Green Fuel Economizer Co., 165 Broadway, New York 6, N.Y.
Wurtz, Robert (J'42) (AEFHNS), 1st Lt., Army Air Forces, 544th Base Hdq. Sqd., A.P.O. 959, c/o Postmaster, San Francisco, Calif.; *for mail*, 114 Fairmount Ave., Hackensack, N.J.
Wyard, Jack Brainerd (J'45) (AJY), 1112 W. 62nd St., Seattle, Wash.
Wyatt, Charles Crawford (J'37) (SFK), Mgr., McManama & Co., 709 Arctic Bldg., Seattle 4; *for mail*, 16037 Maplewild, Seattle 66, Wash.
Wyatt, DeWitt H. ('24) (OMKFCB), Sr. Research Engr., Research Foundation, Ohio State Univ., Columbus; *for mail*, 123 Acton Rd., Columbus 2, Ohio.
Wyatt, Harold Wilber (J'42) (LJDBA), Instr., Graduate Student, Mass. Inst. of Tech., 33 Massachusetts Ave., Cambridge 39, Mass.
Wyatt, Ralph Moore ('44) (SKCDGO), Mgr., Westinghouse Elec. Corp., 739 Hamilton St., Allentown, Pa.
Wyburn, Wilfred ('30; '35) (OSFCLM), Engr., Fed. Paper Bd. Co., 24 River Rd., Bogota, N.J.
Wyckoff, Gerritt I. (J'34) (EBCNSF), Div. Engr., Oil Well Supply Co., 729 Kennedy Bldg., Tulsa 3; *for mail*, 1416 S. College, Tulsa 4, Okla.
Wyckoff, Norman Wilson ('34; '44) (CJMNDB), Asst. Gen. Mgr., Universal Camera Corp., 28 W. 23rd St., New York 10, N.Y.; *for mail*, 163 Forest Ave., Verona, N.J.
Wyder, Carl G. ('45) (CDM), Assoc. Editor, *Factory Management and Maintenance*, McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.; *for mail*, 159 Poplar Ave., Hackensack, N.J.
Wyer, Ramon (J'40) (CM), Treas., Diebold, Inc., Canton 2, Ohio.
Wyer, Samuel S. ('04; '11), Retired; 1325 Cambridge Blvd., Columbus 8, Ohio.
Wygovsky, Raymond N. (J'39), Asst. Foreman, Essex Mch. Co., 46 Marshall St.; *for mail*, 288 Ridgewood Ave., Newark, N.J.
Wykoff, Walter Robert (J'42) (KA), Test Engr., Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford 8, Conn.
Wyle, Frank S. (J'41) (AYMJDB), Lab. Mgr., Triplett & Barton, Inc., 826 N. Victory Blvd., Burbank; *for mail*, 1456 Crisp Canyon Rd., Sherman Oaks, Calif.
Wyler, Charles J. ('21; '31) (EHS), Asst. Ch. Engr., A. M. Lockett & Co., Ltd., 308 Whitney Bank Bldg., New Orleans, La.
Wylie, John Simpson ('34; '36) (SOECF), Engr., Fidelity & Casualty Co. of N.Y., 754 Gas & Elec. Bldg., Denver 2, Colo.
Wylie, William G., Jr. (J'41), 5747 S. Honore St., Chicago, Ill.
Wyllie, John S. (J'21) (BHJKNY), Design Engr., Titeflex, Inc., 500 Frelinghuysen Ave., Newark; *for mail*, 4 Birchwood Ave., East Orange, N.J.
Wylly, Alexander (J'41) (AH), Capt., Army Air Forces, Hdq., Air Tech. Serv. Comd., Engrg. Sch., TSAES, Area B, Wright Field, Dayton, Ohio.
Wylly, William B. (J'40), 651 Milwaukee Rd., Beloit, Wis.
Wyman, Laurence Augustus ('41) (DLMRW), Mgr., & Owner, Wyman Engrg. & Representations Internatl., 1306 Chamber of Commerce Bldg., Cincinnati 2, Ohio.
Wyman, Laurence West ('16; '30) (CHBM), V.P., Ch. Engr., Calif. Corrugated Culvert Co., 7th St., Berkeley 2; *for mail*, 40 Sharon Ave., Piedmont 11, Calif.
Wynkoop, Norman Osborn ('30) (SKEFH), V.P., McGraw-Hill Publ. Co., Inc., 330 W. 42nd St., New York 18, N.Y.
Wynn, Angus Lorenz, Jr. (J'43) (AKRH), Supercharger Engrg., Gen. Elec. Co., 920 Western Ave., West Lynn; *for mail*, 9 Kensington Pk., Lynn, Mass.

Y

Yaeger, Raymond George (J'45) (ABZN), Mech. Engr., Naval Research Lab., Washington, D.C.; *for mail*, 115-12—124th St., South Ozone Park, N.Y.
Yallalee, William Pryce (J'39) (HS), 1st Lt., Landing Vehicle Bd., Ft. Ord, Calif.
Yamauchi, Yoshikazu Wilson (J'39) (AEM), Mch. Oper., Ford Motor Co., 3000 Schaefer Rd.; *for mail*, 2954 Salina Ave., Dearborn, Mich.
Yang, C. Y. ('45), Dean of Engrg., Natl. Cent. Univ., Chungking, China; *for mail*, c/o Prof. P. B. Eaton, Lehigh Univ., Easton, Pa.
Yang, L. O. ('45), Dir., Indus. Dept., Ministry of Economic Affairs, Chungking, China; *for mail*, c/o Prof. P. B. Eaton, Lehigh Univ., Easton, Pa.
Yang, S. Y. ('45), Co-Dir., Ry. Dept., Ministry of Communications, Chungking, China; *for mail*, c/o Prof. P. B. Eaton, Lehigh Univ., Easton, Pa.
Yarber, Gordon W. (J'46), 3512 S. 137th St., Seattle 88, Wash.
Yard, Edward Madison (J'40) (EFJKZ), Staff Engr., John A. Roebling's Sons Co., 640 S. Broad St., Trenton; *for mail*, 2 Whittlesey Rd., Trenton 8, N.J.
Yarnall, D. Robert ('03; '11; F'36) (SCLZHJ), *Manager*, '17-'20, *President*, '46; Pres., Yarnall-Waring Co., 102 E. Mermaid Lane, Philadelphia 18, Pa.
Yaroshuk, Walter Roman (J'42) (MLCB), Engr., Mech. Devel. Div., Bakelite Corp., River Rd.; *for mail*, 115 W. Franklin St., Bound Brook, N.J.
Yarov, Miss Beatrice (J'46) (CDO), York Corp.; *for mail*, 542 W. Springettsbury, York, Pa.
Yarrow, Sir Harold E. (Bt. C. B. E.) ('19), Managing Dir., Messrs. Yarrow & Co., Scotstown, Glasgow, Scotland.
Yaryan, Homer L., Jr. ('38; '46) (OKSF), Product Engr., Wool Sec., Owens-Corning Fiberglas Corp., Nicholas Bldg.; *for mail*, 2109 Robinwood Ave., Toledo, Ohio.
Yates, Alan Hazeltine (J'38) (ABNC), Lt. Comdr., U.S.N., Naval Trng. Schools, Mass. Inst. of Tech., Cambridge, Mass.
Yates, Chapin W. (J'38) (BFS), Indus. Engrg. Div., Remington Arms Co., Inc., Bridgeport; *for mail*, 53 Hill St., Milford, Conn.
Yates, Jas. L. ('24; '33) (CEJ), Ch. Inspr., Worthington Pump & Mchy. Corp., Roberts Ave. & Clinton Sts., Buffalo, N.Y.
Yates, Max Edgar (J'42) (MNCFSJ), Mech. Assignment Engr., Hanford Engr. Works, E. I. du Pont de Nemours & Co., Box 100; *for mail*, 317 Armistead Ave., Richland, Wash.
Yates, Richard C. (J'33), 1837 Section Rd., Cincinnati 16, Ohio.
Yates, Robert M. (J'40), 1215—16th St., N.W., Washington, D.C.
Yates, William Floyd (J'46), Lt. Comdr., U.S.N., Ord. Inspc. Officer, Naval Shipyard, San Francisco; *for mail*, 455 N. Ridge Rd., San Francisco 24, Calif.
Yeager, Bruno John (J'40) (SFKB), Plant Engr., Cincinnati Gas & Elec. Co., Front & Rose Sts., Cincinnati; *for mail*, 583 Tusculum Ave., Cincinnati 26, Ohio.
Yeager, Thomas Homer ('43) (EFKNS), Ch. Engr., Union Diesel Eng. Co., 2200 E. 7th St., Oakland; *for mail*, Box 110, Route 1, Lafayette, Calif.
Yeager, William Francis (J'40) (DMNOAC), Designing & Servicing, R. C. Mahon Co., 231 S. La Salle St., Chicago 4; *for mail*, 1112 Asbury Ave., Evanston, Ill.
Yeager, William Theodore (J'40) (SFOMDK), Plant Engr., RCA Victor Div., Radio Corp. of Am., 501 N. La Salle St., Indianapolis 1; *for mail*, 1610 N. Auburn, Speedway City, Indianapolis, Ind.
Yeakel, A. E. ('31; '35), Bay Shore Rd., Menominee, Mich.
Yeaton, Philip Osborne ('20; '25) (CBM), Head, Dept. Indus. Engrg., 201 Engrg. Bldg., Univ. of Fla., Gainesville, Fla.
Yee, Shuck Kin (J'43) (ADBNH), Engr., Atlas-Pacific Engrg. Co., 3439 Hollis St., Oakland; *for mail*, 806 Oak St., Oakland 7, Calif.
Yelderman, George Robert (J'42) (AESRKG), Field Engr., Elec., Gen. Elec. Co., 1801 N. Lamar, Dallas 2, Tex.
Yelland, Walter Harry, Jr. (J'43), Owner, Mgr., Yelland Trailer Sales, 2920 S. Figueroa St.; *for mail*, 4100 S. Figueroa St., Los Angeles, Calif.
Yellott, John Ingle ('31; '40) (SRF), *Junior Award*, '34, *Pi Tau Sigma Medallist*, '39; Dir. of Research, Loco. Devel. Com., Rm. 1657, O'Sullivan Bldg., Baltimore 2, Md.
Yeo, Edw. J. (J'37) (BHJRAY), Ch. Draftsman, Asst. Ch. Engr., Mech. Research Dept., Am. Brake Shoe Co., Green Acre Lane, Westport; *for mail*, 58 Spring Hill Rd., Norwalk, Conn.
Yeomans, Lucien Ingraham ('43) (ONJHML), Pres., Lucien I. Yeomans, Inc., 205 W. Wacker Dr., Chicago 6, Ill.
Yerino, Lario Vincent (J'42) (ABH), Pvt., 274th Repl. Co., 14th Bn., A.P.O. 703, c/o Postmaster, San Francisco, Calif.; *for mail*, 5711 Hudson Ave., West New York, N.J.
Yerk, Henry Herman ('44) (BMR), Eastern Sales Mgr., Gen. Mchy. Corp., 230 Park Ave., New York 17; *for mail*, Hotel Ten Park Ave., New York 16, N.Y.
Yerzley, Felix L. ('31; '39) (AOCY), Product Engr., West. Elec. Co., 345 Hudson St., New York 14, N.Y.; *for mail*, 66 Harrison St., Verona, N.J.
Yessne, Harmon Stuart (J'44), 3402 Ohio St., Chicago 24, Ill.
Yetter, Gilbert L. ('46) (HJM), Mech. Engr., U.S. Bur. of Reclamation, Denver, Colo.; *for mail*, Box 102, Provo, Utah.
Yingling, Frank Beam ('17) (EARB), Pres., Midwest Loco. Works, 406—1st Natl. Bank Bldg., High St.; *for mail*, 701 Main, Hamilton, Ohio.
Yingling, John Edward (J'43) (ABE), 515 Tioga St., Johnstown, Pa.
Yoakum, Franklin Laughlin (J'43), 472 Vernon St., Oakland, Calif.
Yocom, Luke F., Jr. (J'32) (CFS), Student Engr., Consltd. Edison Co. of N.Y., Inc., 4 Irving Pl., New York 4; *for mail*, 6311 Haring St., Rego Park, N.Y.
Yocum, Wilbur Fisk (J'45) (EHSFRK), Mech. Engr., Naval Engrg. Exper. Sta., Annapolis, Md.
Yoder, Howard Duane ('20) (HLSDK), Research Engr., Penberthy Injector Co., 1242 Holden, Detroit 2; *for mail*, 26705 Middle Belt Rd., Farmington, Mich.
Yoder, Joseph Daniel ('42), Feed Water Div., Permutit Co., 330 W. 42nd St., New York; *for mail*, Chatsworth Gardens, Larchmont, N.Y.
Yopp, Paul R. ('38) (FSTW), Dist. Sales Mgr., Babcock & Wilcox Co., 1203 Candler Bldg., Atlanta 3; *for mail*, 3508 Piedmont Rd., Atlanta, Ga.
Yorgiadis, Alexander John (J'46) (ABE), Design Engr., Sonntag Scien. Corp., 1 Seneca Pl.; *for mail*, c/o Y.M.C.A., Greenwich, Conn.
York, Jesse Louis (J'44) (HDKZL), Instr. Dept. Chem. & Metal. Engrg., Univ. of Mich., 2221 E. Engrg. Bldg., Ann Arbor, Mich.
York, Max Kyle (J'40) (COMNB), Gen. Mgr., Doylestown Mch. Co., Box 100, Danboro; *for mail*, Box 25, Eagleville, Pa.
Yost, Lloyd ('26) (HBNELG), Dir. Research Lab., A. O. Smith Corp., 6310 E. Olympic Blvd., Los Angeles 22, Calif.
Yost, Richard Henry (J'46), Engr. of Mfr., West. Elec. Co., Inc., 245 N. 13th St., Lincoln; *for mail*, 2019 S. 12th St., Lincoln 2, Neb.
Yost, Samuel H. ('32; '35) (BCH), Mech. Engr., Jeffrey Mfg. Co., 1st Ave. & 4th St., Columbus; *for mail*, R.F.D. 1, Worthington, Ohio.
Youell, Leonard Lynde ('22; '25; '35) (SFCKE), Engrg. Rep., Stone & Webster Engrg. Corp., 137 Wellington St. W., Toronto 1; *for mail*, 52 Nina Ave., Toronto 10, Ont., Can.
Young, Almon Paul ('34) (BCKMNS), Assoc. Prof. Mech. Engrg., Mich. College of Min. & Tech., Houghton, Mich.
Young, Andrew J. ('28; '35; '35) (ES), Mchy. Engr., Fidelity & Casualty Co. of N.Y., 1040 Wells Bldg., Milwaukee 2; *for mail*, 4420 N. 25th, Milwaukee 9, Wis.
Young, C. Higbie ('36) (NCLM), Prof., Charge Dept. Mch. Design, Cooper Union, Cooper Sq., New York 3, N.Y.
Young, Charles Duncanson (F'42), V.P., Pa. R.R., 1752 Broad St. Sta. Bldg., Philadelphia 4, Pa.
Young, Charles Francis (A'45), Model Maker, Procter Elec. Co., 3943 Locust St., Philadelphia 4, Pa.
Young, Charles Marshall, Jr. ('16; '25) (BDLNM), Devel. Engr., Link-Belt Co., 2045 Hunting Park Ave., Philadelphia 40; *for mail*, 6946 Cresheim Rd., Philadelphia 19, Pa.
Young, Clem John (J'45) (CLY), Jr. Design Engr., Plant B, Monsanto Chem. Co., East St. Louis, Ill.; *for mail*, 8734 Partridge Ave., St. Louis 15, Mo.
Young, D. L. ('33), Bamberger & Co., Newark; *for mail*, 12 Hamilton Rd., Glen Ridge, N.J.
Young, Dana ('40) (BAH), Prof. Applied Mechanics, Univ. of Tex., Austin 12, Tex.
Young, Donald Valentine (J'39) (MDCNHJ), Cincinnati Milling Mch. Co., Marburg Ave., Cincinnati 9; *for mail*, 1423 Chase Ave., Cincinnati 23, Ohio.
Young, Edgar Wayne (J'45) (NMACBD), Tool Designer, Northrop Aircraft, Inc., Hawthorne; *for mail*, 2614 Barry Ave., Los Angeles 34, Calif.
Young, Edward W. (J'41) (JSYM), Plant Engr., Vulcan Detinning Co., Sewaren; *for mail*, 409 Madison Ave., Rahway, N.J.
Young, Einar T. ('46) (BMP), Mech. Engr. & Head, Drafting Rm. & Exper. Shop, Sun Physical Research & Devel. Lab., 410 Third St., Upland, Chester; *for mail*, 717 Stockton Circle, Ridley Park, Pa.
Young, Everett G. ('19; '24) (MR), Prof. Ry. Mech. Engrg., Univ. of Ill.; *for mail*, 304 W. California St., Urbana, Ill.
Young, Francis George (J'42) (CMJD), Naval Arch., U.S. Maritime Comm., 14th & Franklin, Oakland 12, Calif.

Young, Fred R. (J'43) (ACKMJN), Engr., Boeing Aircraft Co., Seattle; *for mail,* 13631—1st Ave., Seattle 66, Wash.
Young, Frederick Arthur (J'44) (AHNB), Test Engr., Devel. Work, Hamilton Stand. Propellers, Main St., East Hartford; *for mail,* 300 Porter St., Manchester, Conn.
Young, Frederick Lind, Jr. (J'40) (KSH), Assoc. Application Engr., York Corp.; *for mail,* 273 N. Hartley St., York, Pa.
Young, George (J'44) (CJW), Pvt., Sec. R, Bks. 205, Army Air Field, Sioux Falls, S.D.; *for mail,* 12 Laventhal Ave., Irvington, N.J.
Young, George Jayson (J'41) (ABH), 406 Nadina Ave., Millbrae, Calif.
Young, Gustave Otto (J'41) (CNMHB), Mech. Engr., Singer Mfg. Co., Elizabethport; *for mail,* 635 Park Ave., Elizabeth, N.J.
Young, H. Russell ('28; '34; '37) (BAJONZ), Supvr. Engrg., Baldwin Loco. Works, Eddystone; *for mail,* 616 Braeburn Lane, Narberth, Pa.
Young, Harvey Otis, II (J'41), Capt., Army Forward Communication Center, Army Air Base, Santa Ana, Calif.
Young, Henry Ben, Jr. (J'37) (BCM), Ch. Engr., Mission Mfg. Co., Box 4209, Houston 14, Tex.
Young, James Ellsworth (J'46) (ACL), Chugwater, Fla.
Young, James Frederick (J'43) (BCJMNO), Engr., Advance Engrg. Sec., Gen. Elec. Co., 1285 Boston Ave., Bridgeport 2, Conn.
Young, James Winfield (J'30) (RGCS), Asst. M.M., Pa. R.R., Pennsylvania Sta., Baltimore 2; *for mail,* Ruxwood, Riderwood, Md.
Young, John Graham (J'39) (MCA), Supt., Elec. Clock Dept., Barr Mfg. Corp., N. Senaca St.; *for mail,* 105 Jackson St., Weedsport, N.Y.
Young, John Hess, Jr. ('16; '21) (EF), Serv. Engr., Stand. Oil Co. of N.J.; *for mail,* 19 Lynwood Ave., Wheeling, W.Va.
Young, John Mason ('02; '10), Prof. Engrg., Univ. of Hawaii; Pres., Mgr., Pacific Engrg. Co., Ltd.; *for mail,* Box 638, Honolulu, T.H.
Young, John W. (J'41) (FKSR), c/o Jaroch, Apt. 613, 16 Fleet Walk, Ft. Green Housing, Brooklyn, N.Y.
Young, Max L. (J'40) (ESFDKM), Product Design & Process Engr., C. H. Dutton Co., 630 Gibson St.; *for mail,* 931 Wellington Ave., R.R. 2, Kalamazoo, Mich.
Young, Merrill A. ('45) (SKHFOC), Steilacoom, Wash.
Young, Paul M. ('29; '35), Mech. Engr., Anaconda Copper Mining Co., 511 Hennessy Bldg.; *for mail,* 1030 W. Porphyry St., Butte, Mont.
Young, Peter John, Jr. ('34; '42) (CHJKLR), Project Engr., Chem. Constr. Corp., (Am. Cyanamid Co.), 350—5th Ave., New York 1; *for mail,* 4410 Cayuga Ave., Riverdale, Bronx, N.Y.
Young, R. Hence ('26; '34; '35) (FSCDEK), Plant Engr., Niagara Sprayer & Chem. Co., Inc., Middleport; *for mail,* R.F.D. 3, Medina, N.Y.
Young, Rex ('42), Bryson Pipeline & Refining Co., Bryson, Tex.
Young, Robert Madison ('44) (BCDFMO), Mgr., Consltd. Citrus Products Co., Inc., 34th St. & 2nd Ave.; *for mail,* 4111 Granada St., Tampa, Fla.
Young, Robert Speed (J'39) (MCJZ), c/o L. J. Young, 1227 Forest Ave., Ann Arbor, Mich.
Young, Robert Worthen (J'41) (BAC), 133 Lincoln Ave., Portsmouth, N.H.
Young, Sam Houser, Jr. (J'42) (NHSBKM), Engr., Paul G. Bentley Co., Burt Bldg.; *for mail,* 2004 Bennett St., Dallas, Tex.
Young, Selah P. ('22; '35) (ESF), Mech. Engr., Diesel Power Design, Navy Dept., Bur. of Yards & Docks, Washington 25, D.C.; *for mail,* 828 S. Ivy St., Arlington, Va.
Young, Vincent Willard ('29) (ASKBNE), Prof. Mech. Engrg., Engrg. Sch., Okla. Agric. & Mech. College, Stillwater, Okla.
Young, Wendell Stratton (J'45) (CMKWNA), U.S.N.R., Commissary Officer, Commissary Office, Lido Beach Separation Center, Long Beach, L.I., N.Y.; *for mail,* 111 South St., Foxboro, Mass.
Young, Whitney James (J'42) (MJYACO), Sgt., 210½ S. McCann St., Kokomo, Ind.
Young, William E. (J'37) (CMNZJL), Ch. Engr., Clay-Adams Co., Inc., 44 E. 23rd St., New York 10, N.Y.
Youngclaus, William Paul, Jr. (J'35) (EAMNY), Dist. Sales Engr., Flexpansion Corp., 185 N. Wabash Ave., Chicago 11; *for mail,* 2114 Fir St., Glenview, Ill.
Youngdahl, Paul Frederick (J'42) (JD), Ensign, U.S.N.R., 142 Ivanhoe St., S.W., Washington 20, D.C.; *for mail,* 1049—9th Ave., Brockway, Pa.
Younger, John Elliott ('38) (AB), *Spirit of Saint Louis Medallist,* '41; Prof. & Head of Mech. Engrg., Univ. of Md., College Park, Md.
Youngman, Edward Florian (J'45) (AESMHJ), Sales Engr., Key Co., 27th & McCosland, East St. Louis, Ill.; *for mail,* 5431A Gertrude Ave., St. Louis 16, Mo.
Youngquist, Robertson (J'41) (ALKEHZ), Lt., U.S.N.R., Bur. of Aero., Navy Dept., Washington, D.C.; *for mail,* 197 Main St., Annapolis, Md.
Youngson, Alexander C. ('22; '30) (OSLHFD), Partner, Rice & Youngson, 140 Cedar St., New York 6, N.Y.
Younkin, Gavin S. ('46), Branch Mgr., Minneapolis-Honeywell Regulator Co., Rm. 939, 714 W. Olympic Blvd., Los Angeles 15; *for mail,* 380 Redwood Dr., Pasadena 2, Calif.
Youtie, Robert Kay (J'45) (MGBDZN), Occupational Counselor, Adj. General's Office, Co. D, 4th Bn., Separation Center, Ft. George C. Meade, Md.; *for mail,* 2 S. Hillside Ave., Ventnor City, N.J.
Yuill, Robert Russell (J'46) (ACF), Asst. Supt., Tissue Dept., E. B. Eddy Co., Hull; *for mail,* Chelsea, Que., Can.
Yulke, Samuel George (J'45) (CMO), Dir., Charge Activities, S. G. Yulke Associates, 341 Madison Ave., New York 16; *for mail,* 19 E. 98th St., New York, N.Y.
Yungbluth, John Ambrose (J'45) (NEZSBA), Designer, White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis; *for mail,* 3531A McKean St., St. Louis 18, Mo.
Yurko, Malvin Michael (J'45), Engr.-Draftsman, Socony-Vacuum Oil Co., 154 Nassau St., New York; *for mail,* 14 Eldert St., Brooklyn 7, N.Y.
Yusem, Howard Richard (J'44) (MOAC), Lt. (j.g.), U.S.N.R., PC-1184, U.S.N. Sta., Astoria, Ore.; *for mail,* 6635 McCallum St., Philadelphia 19, Pa.

Z

Zabriskie, A. Eugene (J'40) (BEM), Tool Designer, Wright Aero. Corp., Paterson; *for mail,* 63 Prospect St., Jersey City, N.J.
Zachow, Wm. C. ('37), Advis. Engr., Atlantic Refining Co., Passyunk Ave., Philadelphia, Pa.
Zack, Eugene T. (J'40) (JMR), 942 Gunderson Ave., Oak Park, Ill.
Zademach, Erich R. ('20) (BCDJMN), Engr., Partner, Metalwash Mchy. Co., 149-155 Shaw Ave., Irvington 11; *for mail,* 956 Revere Dr., Hillside 5, N.J.
Zafian, Henry John (J'42) (CAKNO), Pat. Engr., Pat. Dept., Gen. Elec. Co., Munsey Bldg., Washington 4, D.C.; *for mail,* c/o Kenyon, 62 W. 55th St., New York 19, N.Y.
Zahka, Anthony George (M'45) (KLDNZC), Job Engr., Colgate-Palmolive-Peet Corp., 1 Newark Ave.; *for mail,* 654 Bergen Ave., Jersey City, N.J.
Zahn, Otto E. E. ('42) (BGL), 218 Napoleon Blvd., South Bend, Ind.
Zalaha, John (A'43) (CRSWY), Mgr., Pacific Lumber Co., S.A., Ave. de Noviembre, 806 Pte., Durango, Dgo., Mex.
Zaleski, Walter Robert (J'44) (ABM), 515 W. Marion St., South Bend, Ind.
Zalewa, Stanley Frank, Jr. (J'41) (MCDHBN), Indus. Engr., Mch. Shop, Link-Belt Co., 300 W. Pershing Rd., Chicago; *for mail,* 1534 S. 50th Ave., Cicero 50, Ill.
Zallen, Maurice (J'41) (GKS), Draftsman, Babcock & Wilcox Co., 85 Liberty St.; *for mail,* Hotel Webster, 40 W. 45th St., New York, N.Y.
Zamboky, Andrew Nicholas (J'43) (NJBA), Research Engr., Aluminum Research Labs., Aluminum Co. of Am., Box 772, New Kensington; *for mail,* 2830 Beechwood Blvd., Pittsburgh 17, Pa.
Zamore, Wallace William (J'44) (AKMBLH), 201 Shippen St., Weehawken, N.J.
Zane, Hysler Bernard (J'41) (ACD), Designer & Draftsman, Downer, Green & Carillo, 30 Rockefeller Plaza, New York; *for mail,* 85-22—124th St., Richmond Hill, N.Y.
Zane, William (J'43) (BAN), Engr., Applied Mech. Sec., Westinghouse Elec. Corp., Lester Branch P.O., Philadelphia 13; *for mail,* 300 Pilgrim Lane, Drexel Hill, Pa.
Zapata, Juan Victor (J'44) (ACFHNS), Chief, Tech. Office, Engrg., Chilean Navy, Apostadero Naval, Talcahuano; *for mail,* Naval Arsenal, Talcahuano 3, Chile, S.A.
Zapp, Frank Clayton (J'46) (BCM), 203 Waddell Circle, Oak Ridge, Tenn.
Zareh, Edward (J'39) (FESH), Engr., Oil Tool Design, McCullough Tool Co., 5820 S. Alameda, Los Angeles 27; *for mail,* 1179 S. Lucerne Blvd., Los Angeles 6, Calif.
Zarnowski, Frank Joseph (J'43) (CMJN), Project Engr., Walter Kidde & Co., 675 Main St., Belleville; *for mail,* 165 Wheaton Pl., Rutherford, N.J.
Zattler, George W. (J'41), 2408 Patterson St., Pittsburgh 3, Pa.
Zaunmiller, Edward W. ('32; '38), Lt., U.S.N.R., 1277 Commonwealth Ave., Boston, Mass.
Zaura, Verne (J'45), Lt., U.S.A.; R.R. 1, Madison, Neb.
Zavodny, Stephen (J'37) (CKS), 2nd Lt., Ord. Dept., U.S.A.; *for mail,* 5 Cox Ave., Yonkers 5, N.Y.
Zeder, Fred Morrell ('20), Vice-Chmn. of Bd., Chrysler Corp., Box 1919, Detroit 31, Mich.
Zeffren, Henry Edward (J'37) (SZC), Instrument Engr., Chemist, Union Elec. Co. of Mo., 12th & Locust Sts., St. Louis 1; *for mail,* 5625 Enright, St. Louis 12, Mo.
Zegers, Vincent Joseph (J'45) (MEBS), Tech. Staff, Ord. Dept., U.S.A., 38 S. Dearborn St., Chicago 3; *for mail,* 2156 Windsor Ave., Chicago 25, Ill.
Zehring, Richard M. (J'40) (ABJ), 255 N. Cherry St., Germantown, Ohio.
Zeigler, George Edward, Jr. (J'41) (SCMODN), Lt. Comdr., U.S.N.R.; *for mail,* Drawer 1139, 602 Gunter St., Tallahassee, Fla.
Zeigler, Ralph W. (J'46), Fisher Governor Co.; *for mail,* 1008 S. 1st Ave., Marshalltown, Iowa.
Zeiner, Eugene Francis ('23; '28) (FJLSY), 401 N. Broad St., Philadelphia 8; *for mail,* 506 Ott Rd., Bala-Cynwyd, Pa.
Zeitler, Earl Charles (J'45) (ABC), Group Engr., Consltd. Vultee Aircraft Corp., San Diego; *for mail,* 2453 Union, San Diego 1, Calif.
Zeitlin, Alexander ('40) (COKNLB), Tech. Adviser to Pres., Rogers Diesel & Aircraft Corp., 1120 Leggett Ave., New York 59, N.Y.
Zeitlin, Eli Arnold ('39; '45) (ABNHE), Structures Engr. Charge Flight Loads & Stress Analysis Criteria, Ryan Aero. Co., San Diego; *for mail,* 2927 Preece St., San Diego 11, Calif.
Zeitner, Jesse J. (A'43) (CLM), Supvg. Engr., Ocean Accident & Guarantee Corp., Ltd., 1 Park Ave., New York 16, N.Y.
Zeliff, David Eichorn (J'35) (CEM), Engr., West. Elec. Co., 300 Central Ave., Kearny; *for mail,* 121 Brookfield Ave., Nutley 10, N.J.
Zell, Adolph William ('40) (BELC), Engr., Charge Operas., Sterling Constr. Co., 101 W. Schellenger Ave.; *for mail,* 156 W. Bennett Ave., Wildwood, N.J.
Zeller, Joseph W. ('28) (SENF), Head Mech. Engrg. Dept., Northeast. Univ., 360 Huntington Ave., Boston, Mass.
Zenaty, Bert ('28; '35) (CBFOM), Dir., Mech. Engr., Waldes Koh-I-Noor, Inc., 47-16 Austell Pl., Long Island City; *for mail,* 394 Burns St., Forest Hills, L.I., N.Y.
Zenlea, Perry (J'45), Asst. Mech. Engr., Plant Devel., Max Ams Inc., 376 Greenwich St., New York; *for mail,* 107-22—91st St., Ozone Park 17, L.I., N.Y.
Zeno, D. Raoul ('37; '44) (ABK), Engr., Newport News Shipbldg. & Dry Dock Co.; *for mail,* 43 Pear Ave., Newport News, Va.
Zepht, Ernest Eugene ('25; '35) (CJD), Maint. Supvr., E. I. du Pont de Nemours & Co., Grasselli; *for mail,* 111 Thelma Terrace, Linden, N.J.
Zeppenfeld, Carl William (J'45), U.S. Navy; 873 Devon St., Arlington, N.J.
Zeppenfeld, Sterling Dan (J'45), Ensign, U.S.N.R., U.S.S. *Massachusetts,* F.P.O., San Francisco, Calif.; *for mail,* 873 Devon St., Arlington, N.J.
Zerban, Alexander Hamilton ('27; '36) (KEBLSR), Prof. Mech. Engrg., Pa. State College; *for mail,* 536 E. Foster Ave., State College, Pa.
Zerby, Robert J. (J'43) (ABCN), Ensign, U.S.N.; 58 Shady Lane, Johnstown, Pa.
Zeskey, Charles R., Jr. (J'37) (CWMDGO), Ch. Engr., T. H. Mastin & Co., 1907 Grand Ave., Kansas City 8, Mo.
Zetterman, Harry L. ('39) (DS), Engr., 1408 Bryan Ave., Salt Lake City 5, Utah.
Zettl, Frank Wm. (J'40) (CNMDHG), Design Engr., Heil Co., 30th & W. Montana St., Milwaukee; *for mail,* 1103 S. 34th St., Milwaukee 4, Wis.
Zieber, William Edgar ('35) (BKESJA), Dir. of Research, York Corp., York, Pa.
Ziebolz, Herbert W. (J'38) (ZBHN), V.P., Charge Engrg., Askania Regulator Co., 1603 S. Michigan Ave., Chicago 16, Ill.
Ziegler, John W. ('42), 133 Wickham Rd., Garden City, N.Y.
Ziegler, Theo. F. ('39) (SFBCYO), Power Engr., E. I. du Pont de Nemours & Co., Wilmington, Del.; *for mail,* 108 Hilltop Rd., Philadelphia 18, Pa.
Zielinski, Ted (J'42) (BOC), Ensign, U.S.N.R., Naval Air Sta., San Diego; *for mail,* 279 N. Euclid Ave., Pasadena 4, Calif.
Zier, Harold George (J'39) (ABC), Capt., U.S.A., Co. D, 19th Engrs., A.P.O. 464, c/o Postmaster, New York, N.Y.
Zietlow, Fred Patrick (J'42) (KJFDCM), Mech. Engr., Allen & Kelley, 435 Archs. & Builders Bldg., Indianapolis; *for mail,* 3969 Ruckle St., Indianapolis 5, Ind.
Zieve, William A. (J'33) (TNSJ), Propr., Wil-Tex Mills, 354 Broadway, Brooklyn 11; *for mail,* 1471 E. 51st St., Brooklyn 3, N.Y.
Zigas, Arthur L. (J'44) (NOKBSY), Jr. Engr., Contractors Refrigeration Corp., 89-35 Queens Blvd., Elmhurst, N.Y.
Zikaras, Alvin John (J'46), 202 Indian Ave., Bridgeport 6, Conn.
Zilboorg, James M. ('21; '28; '35) (SEFLZC), Head of Powers Sales, Cia. Impulsora de Empresas Electricas, S.A., Apartado 8-bis, Mexico, D.F., Mex.
Zillman, Rudolf William (J'41) (J), Pvt., 1st Class, U.S.A., Hospital Lab., Medical Sec., 1962nd Serv. Comd. Unit, Ft. Ord, Calif.
Zimmer, Abraham I. (J'39) (NSKAJM), V.P., Plant Engr., Hardware Specialties Mfg. Co., Box 844, Bridgeport 1; *for mail,* 726 Merritt St., Bridgeport 6, Conn.
Zimmerli, Franz Perrine ('27) (BJ), Ch. Engr., Barnes-Gibson-Raymond, Div. of Assoc. Spring Corp., 6400 Miller Ave., Detroit 11, Mich.
Zimmerman, Albert (J'41) (BHJN), Ch. Engr., Merco Centrifugal Co., 582 Market St., San Francisco 4; *for mail,* Box 141, Tiburon, Calif.

Zimmerman, Albert (J'48) (MBHG), Mech. Engr., Charge Research Lab. Analysis, Douglas Aircraft Co., Inc., 1380 S. Sepulveda Blvd., Los Angeles; *for mail,* 1162 N. Stone St., Los Angeles 33, Calif.
Zimmerman, Arnold William (J'45) (SEHM), Engr., Power Devel., U.S. Bur. of Reclamation, Calif. Fruit Bldg.; *for mail,* 1831 Caramay Way, Sacramento 14, Calif.
Zimmerman, Carl D. ('19; '25) (EFS), Asst. Supt. of Prod., Ohio Edison Co., 47 N. Main St., Akron, Ohio.
Zimmerman, Charles ('26; '35) (JMSNLD), Supt., Raymond Lead Works, Natl. Lead Co., 900 W. 18th St., Chicago 80; *for mail,* 468 Prairie Ave., Elmhurst, Ill.
Zimmerman, Earl W. ('26) (SHYKLO), Assoc. Engr., Mech. Design Sec., U.S. Engrs. Office, Chimes Bldg., Syracuse 2; *for mail,* 704 Bellevue Ave., Syracuse 4, N.Y.
Zimmerman, Frank Edward, Jr. (J'45) (BCDLMZ), Student Engr., Trng., Ford Motor Co., 3000 Schaefer Rd., Dearborn; *for mail,* Apt. 405, 11832 Ohio, Detroit 4, Mich.
Zimmerman, Frederick Lembke (J'41) (CMSJAY), Lt. Comdr., U.S.N.R., Res. Officer-in-Charge, Naval Ord. Lab., White Oak, Md.; *for mail,* 2103 Suitland Terrace, S.E., Washington 20, D.C.
Zimmerman, Howard Thurlow ('29; '42) (CHNM), V.P., Ch. Engr., Ralph B. Carter Co., 192 Atlantic St., Hackensack; *for mail,* 14 Fairway, Maywood, N.J.
Zimmerman, John H. ('32) (BJK), Asst. Dir., Devel. Dept., Linde Air Products Co., 30 E. 42nd St., New York, N.Y.
Zimmerman, Lawrence Edward ('44) (JMR), Dist. Mgr. Charge Sales, Pettibone Mulliken Corp., 4710 W. Division St., Chicago 51, Ill.; *for mail,* Box 1559, Denver, Colo.
Zimmerman, Lloyd M. (A'45) (CMDW), Supvg. Engr., McClure, Hadden & Ortman, Inc., 111 W. Washington St., Chicago 2, Ill.
Zimmerman, Milton Joseph ('46), Asst. Ch. Engr., City Pub. Serv. Bd., Box 1771, c/o Mission Rd. Plant, San Antonio 5, Tex.
Zimmerman, Richard E. ('27; '35) (SBJWF), Designer, Consltd. Gas, Elec. Light & Power Co., Baltimore 3; *for mail,* 2710 Auchentordly Terrace, Baltimore 17, Md.
Zimmerman, William F., Jr. (J'37) (CMOL), Engr., West. Elec. Co., 23rd & Cicero; *for mail,* Univ. Hotel, 5519 Blackstone Ave., Chicago, Ill.
Zink, Adolbert Henry (J'41) (JMCZDA), Ch. Tooling Engr., Bay City Div., Dow Chem. Co., Patterson St.; *for mail,* 528 Webb Dr., Bay City, Mich.
Zink, Charles William (J'40) (CNH), Gen. Mgr., Epworth Mfg. Co., 6587 Epworth Blvd., Detroit 10, Mich.
Zink, George A. (J'29) (BMECOH), Works Mgr., Detroit Diesel Eng. Div., Gen. Motors Corp., 13400 W. Outer Dr., Detroit 23, Mich.
Zink, J. Joseph (J'45) (MCNZ), Prod. Engr., Rathbun & Bird Co., Inc., 85 Grand St.; *for mail,* Box 174, Canal St. Sta., New York 13, N.Y.
Zink, John H. ('39) (EJKSY), Pres., Heat & Power Corp., 422-28 E. Saratoga St., Baltimore 2, Md.
Zink, William R. ('39) (CMNO), Cons. Engr., Beechhurst Court, 154th St., Beechhurst, L.I., N.Y.
Zinn, Harlowe Roland ('44), Ch. Engr., Flintkote Co., East Rutherford; *for mail,* 22 Mt. Vernon Rd., Montclair, N.J.
Zion, Robert William (J'45) (ASEF), Experimental Test Engr., Experimental Test Office, Pratt & Whitney Aircraft Div., United Aircraft Corp., East Hartford, Conn.
Zircher, Francis J. (J'37) (EH), Asst. Sales & Serv. Mgr., Woodward Governor Co.; *for mail,* 1623 N. Winnebago St., Rockford, Ill.
Zirin, Maxwell J. (J'37) (NHL), Asst. Project Engr., Photo Records Div., Remington Rand, Inc., 68—34th St., Brooklyn 32; *for mail,* 809 Friel Pl., Brooklyn 18, N.Y.
Zirkel, Paul John ('45) (NMDCBA), Mech. Engr., Internatl. Harvester Co., Bueter Rd., Ft. Wayne 8, Ind.
Zisman, William Albert ('45) (BFPEH), Chief, Lub. Sec., Naval Research Lab.; *for mail,* 2027—38th St., S.E., Washington, D.C.
Zitke, Ernest John (J'45), Test Engr., Motors & Generators, Gen. Elec. Co., Broadway Plant, Ft. Wayne, Ind.; *for mail,* 65 Galen St., Brockton 11, Mass.
Zjawin, John Charles (J'40) (AERC), Sr. Test Engr., Wright Aero. Corp., Paterson; *for mail,* 2361 Boulevard, Jersey City 4, N.J.
Zlatin, Norman (J'44) (BJHNZ), Research Engr., Cincinnati Milling Mch. Co., Cincinnati 9; *for mail,* 6023 Yosemite Dr., Cincinnati 12, Ohio.
Zmola, Paul Carl (J'44) (BAEHK), Instr., School of Mech. Engrg., Purdue Univ., Lafayette, Ind.
Zober, Walter ('42) (CDFJMY), Pres., Bennel Mch. Co., Inc., 20 Grand Ave., Brooklyn 5, N.Y.
Zolkover, Melvin (J'43), 951 S. St. Andrews Pl., Los Angeles, Calif.
Zoller, Lawrence (J'41) (CKLW), Plant Mgr., Distribuidores Generales de Mexico, Calzada de Tlalpan 1633, Mexico, D.F., Mex.
Zoller, Ronald Ernest ('44) (KSFJEL), Design Engr., Charge Marine Boiler Design, Babcock & Wilcox Ltd., 34 Farringdon St., London E.C. 4; *for mail,* 42 Hillfield Park, Winchmore Hill N21, England.
Zook, Stanley Otis (J'34) (CFONB), Mech. Engr., J. E. Baker Co., 114 N. George, York, Pa.
Zorn, Joseph Francis, Jr. (J'37) (EKS), Lt., Hdq., 401st Sep. Coast Arty. Bn. (AA), Camp Haan, Calif.
Zorn, William George (J'43) (NDLOBK), Project Engr., Francis Co., 343 S. Dearborn St., Chicago 4, Ill.
Zorwitz, Jerome ('45) (NBOMLZ), Chief, Mech. Design Sec., Signal Corps Engrg. Labs., Evans Signal Lab., Belmar; *for mail,* 56 Grand Ave., Long Branch, N.J.
Zouck, George Hauer ('14; '26) (ACMNRS), Asst. to Pres., Balmar Corp., Clipper Rd., Woodberry, Baltimore 11; *for mail,* 5409 Falls Road Terrace, Baltimore 10, Md.
Zowski, Thaddeus (J'42) (NHAB), Design Engr., S. Morgan Smith Co., Hartley & Lincoln Sts.; *for mail,* 465 Pennsylvania Ave., York, Pa.
Zsuffa, Leslie F. ('32; '42) (CDMOEF), Dir. Pub. Relations, Ga. Sch. of Tech., 225 North Ave., N.W., Atlanta, Ga.
Zuberbühler, Paul ('30; '37) (BDR), No. 8 Sonnenberg Strasse, Berne, Switzerland.
Zublin, John A. ('30) (JM), Mech. Engr., Universal Engrg. Co., Box 299A, Route 1, Sable Ranch, Saugus, Calif.
Zucchero, Joseph Ernest (J'42) (JLM), 7263 Country Club Dr., Normandy 21, Mo.
Zuck, Melvin Albert (J'43) (BSZO), Jr. Partner, C. W. Zuck & Sons, Box 1183; *for mail,* 3380 W. 26th St., Erie, Pa.
Zuckerberg, Harry ('35; '45), Struc.-Weight Design Engr., Chance Vought Aircraft Div., United Aircraft Corp., Stratford, Conn.; *for mail,* 570 Ft. Washington Ave., New York 33, N.Y.
Zucrow, Maurice J. ('32) (CJL), Aerojet Engrg. Co., 285 W. Colorado St., Pasadena, Calif.
Zullo, Samuel ('40), 1 Astor Pl., Jersey City, N.J.
Zuris, Paul (J'39) (LZO), Research Engrg. Group Leader, Monsanto Chem. Co., 1700 S. 2nd St., St. Louis 4; *for mail,* 3914 Watson Rd., St. Louis 9, Mo.
Zvone, Joseph Willard ('33; '42) (DJCSM), Ch. Engr., U.S. Steel Supply Co., 1319 Wabansia St., Chicago 22, Ill.
Zwack, Raymond Theodore ('45) (HBZANE), Project Engr., Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro; *for mail,* 146 Smull Ave., West Caldwell, N.J.
Zwissler, Lewis Edward (J'44) (BONKAH), Partner & Cons. Engr., Associated Consultants, 6922 Stewart Ave., Chicago 21, Ill.
Zywiec, Albin Casimer (J'46) (BKM), U.S. Navy; 4912 Alexander Ave., East Chicago, Ind.

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Donovan, G. C.
Doyle, W. L. H.
Edwards, E. D.

Fletcher, L. J.
Griner, J. E.
Henderson, J. R.
Kent, N.
Kniese, H. G.
Krosse, G. T.
LeTourneau, R. G.
Luney, E. R.
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Mees, R. T.
Meyer, F. L.
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Porter, J. O.
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Rein, F. B.
Rosen, C. G. A.
See, G. L.
Smith, W. H.
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Stainton, R. S.
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Van Zandt, P. C.
Walch, R. H.
Wallace, R. S.
Weiner, L. P.
Wellington, Q. W.

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Nutt, R. E.

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Rouse, D. H., Jr.

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Floor, U.
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Greenman, H. M.
Hallberg, E. E.
Johnson, A. M.
Johnson, B. E.
Johnson, F. M.
Kaeding, R. R.
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Laughnan, T. G.
Mansfield, J. H.
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McCarthy, R. H.
Olsen, J. O.
Perry, R. S.
Peterson, B. A.
Riddiford, A. B., Jr.
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Tipton, H.
Zircher, F. J.

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Ham, C. W.
Henry, J. A.
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Kratz, A. P.
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Moore, H. F.
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Polson, J. A.
Ritchey, L. B.
Ryan, D. G.
Schrader, H. J.
Seely, F. B.
Seyfarth, F.
Starr, C. J.
Starr, M. O.
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Griffin, J. F.
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Jasis, P. G.
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Menson, J. L.
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Siebert, V. W.
Silverman, H. T.
Small, M. H.
Thomson, J.
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Lobley, F. A.
Loomis, A.
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Waddington, L. E.
White, W. McK., Jr.

EVANSVILLE

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Brust, G. A.
Frankland, G. E.
Garvey, R. P.
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Stone, H. L.
Von Arb, E. J.

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Frost, G. H.
Gallatin, R. E.
Goglia, G. L.
Griffin, N. E.
Hackney, E. W.
Kilcoin, F. B.
Mason, F. C.
McInerney, F. T., Jr.
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Paxson, R. D.
Planck, I. A.
Schmidt, C. W.
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Jenks, S. M.
Johnson, P. E.
Josephson, S. N.
Landen, S.
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Peet, J. L.
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Stentz, F. W.
Thiel, W. A.

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Witters, A. G.

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Bond, J. S.
Booth, P. E.
Bretzlaff, G. A.
Brill, J. B.
Bruney, R. C.
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Collins, A. R.
Crews, L. A.
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De Haven, I. C., Jr.
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Godley, E. B.
Grisbaum, L. D.
Guernsey, C. O.
Hanley, W. A.
Hannewald, B. A.
Hartley, H. D.
Heine, G. H.
Helm, P. F.
Hines, G. E.
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Holbrook, G. E.
Jacklin, H. M.
Jehle, F.
Jones, T. A.
Ker, A. W.
Kline, C. LeR.
Krannert, H. C.
Lancaster, R. G.
Langfitt, J. K.
Lembcke, R. K.
Liddle, S. W.
Little, E. W., II
Longacre, G. W.
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McMahan, R. G.
Morris, J. B.
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Nulsen, M. E.
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Olson, M. L.
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Patterson, T. H.
Pearce, E. S.
Pert, D. M.
Randall, R. D.
Reid, H.
Ridgely, H. F.
Riggs, J. D.
Roberts, J. R.
Roberts, W. S.
Roland, J.
Ross, E. R.
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Sawchyn, S.
Schanz, J. L.
Schmid, C. G.
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Schonberg, H. E.
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Simon, H. J.
Skinner, J. C.
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Smith, G. K.
Stampil, L. A.
Taylor, W. M.
Trainor, J. J.
Tramontini, V. N.
Updike, R. W.
Vance, C. B.
Van Meter, H.
Voyles, R. M.
Waddell, W. C.
Wagner, R. E.
Walker, R. S.
Weaver, F. R.
Weber, G. C.
Weiss, C. R.
Wrightman, S. A.
Yeager, W. T.
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Loman, J. K.
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Beese, C. W.
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Hockema, F. C.
Holowenko, A. R.
Jones, J. B.
Jones, Mrs. J. B.
Lindley, R. W.
McAllister, A. J.
Morse, F. B.
Mundel, M. E.
Pigage, L. C.
Rubenkoenig, H.
Solberg, H. L.
States, C. B.
Zmola, P. C.

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Ballman, H. C.

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Burns, F. D.
Davis, R. F.
Pugsley, W. H.
Sprague, P. T.

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Fay, O. K.
Firth, D.
Fitch, R. C.
Huber, G. J., Jr.
Kingsbury, R. C.
Kristl, F. R.
May, D., Jr.
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Voorhees, G. A.

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Renn, J. A.

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Grunewald, R. L.

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Fitzpatrick, J. R.
Goerky, C. M.
Killmer, H. P.
MacLean, J. A.
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Schnaible, A. P.
Smith, LaR.
Sparrow, S. W.
Spicacci, A. R.
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Zahn, O. E. E.
Zaleski, W. R.

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Binder, R. C.
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Clark, D. S.
Davis, E. J.
Fairman, S.
Geiger, J. W.
Girvin, H. F.
Greve, F. W.
Hawkins, G. A.
Ludy, L. V.
Marek, C. T.
Marsh, W. D.
Messersmith, C. W.
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Mock, L. K.

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Honza, D. W.
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Mart, L. T.
Russell, R. A.
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Helander, L.
Mack, A. J.
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Pickett, G.
Ridenour, J. O.
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Seaton, R. A.
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Henderson, C. L.
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Mills, E. B.
Morrison, L. A.
Mosbacher, B. H.
Murray, W. A.
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Stoner, J. H.
Sutton, F. M.

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BOWLING GREEN
Anderson, M. M.

COVINGTON
Cincinnati Section
Kennedy, W. C.
Smith, H. A.
Smith, H. W.

DAYTON
Cincinnati Section
Schwebel, E. C.

EARLINGTON
Louisville Section
Land, G. W.

FT. KNOX
Louisville Section
Blake, J. P.
Callahan, W. J.
Jacobson, R. F.
Meiselman, S.
Posse, E. W.
Ryan, B. E.
Salimbene, R. C.

HARRODSBURG
Louisville Section
Close, R. G., Jr.
Jones, S. B.

HENDERSON
McLain, W. R.

LEXINGTON
Louisville Section
Hawkins, R. D.
Irish, S. B.
Jett, C. C.
O'Bannon, L. S.
Walton, S. B.
Wilson, C. R.

LOUISVILLE

Louisville Section

Arnold, R. M.
Baker, C. LeR.
Benton, E. D.
Bill, C. E.
Birn, S. A.
Capstack, E. J.
Churchill, L. S.
Clower, M. G.
Credo, J.
Cummins, N. W.
Davis, J. H.
Dreyer, E. J.
Duerr, F. R.
Dunn, J. J.
Edel, W. L.
Eldridge, C. D.
Fenwick, H. H.
Garst, R. E.
Godbey, J. W.
Hall, C.
Hambleton, W. VanM.
Heuser, H. V.
Horlander, L. A., Jr.
Hunt, W. F.
Hurst, J. F.
Jackson, L. R.
Johnson, J. A.
Krause, O. C., Jr.
Lind, J. E.
Lomax, B., Jr.
Lucas, W. F.
Marshall, E. S.
Mattimore, J. D.
Metz, C. L.
Meyer, J. K.
Murphy, H. C.
Myatt, D. J.
Oreskovich, P. J.
Preising, W. J.
Roberts, C. L.
Rosenbaum, A. G.
Sack, M.
Sawyer, C. E.
Scheidt, K. H., Jr.
Shannon, F. P.
Simpson, W. M.
Skonberg, E. A.
Speed, W. S.
Tetzel, F. B.
Trosper, R. S.
Vance, L. S.
Waage, J. L.
Welsh, E. J.
Wener, N. L.
Wilkinson, F. L., Jr.
Witherspoon, D. L.
Worth, E. B.
Wuest, W. D.

LUDLOW

Cincinnati Section

Pfahler, R. D.

OWENSBORO

Patitz, G. N.
Smith, E.

PADUCAH

DeSpain, T. H.
White, W. R.

PEWEE VALLEY

Louisville Section

Cook, B. F., Jr.

WILLIAMSBURG

East Tennessee Section

Hains, C. F.

LOUISIANA

ALEXANDRIA

New Orleans Section

Goldberg, J. N.
Lemoine, S. J., Jr.

ALGIERS

New Orleans Section

Milan, D. A.

ARABI

New Orleans Section

Becnel, P. A., Jr.

BATON ROUGE

New Orleans Section

Bracken, V. P.
Broussard, J. A.
Carroll, F. T., Jr.
Chambers, J. W.
Clark, F. G.
Coco, R. G.
Cooper, C.
Crossan, T. E.
Daviet, C. E.
Decker, L. M.
Dennis, E. L.
Gurney, W. B.
Hoskins, R. I., Jr.
Hoyt, C. P.
Johnson, H.
Kerr, E. W.
Lassalle, L. J.
Le Blanc, J. A., Jr.
Lucas, J. L.
Matherne, R. A.
Matthes, G. F.
Pugsley, O. S., Jr.
Roberts, G. S.
Robertson, R. J.
Waterfall, H. W.
Whipple, W.
Whitaker, W. A.
Winslow, W. H., Jr.

BOGALUSA

New Orleans Section

Cowan, E. L.
Pierce, B. B.

CAMP CLAIBORNE

Keith, B. G.
Sheehan, E. F.

CAMP POLK

Drewry, I. O.
Phillips, A. A.

CHALMETTE

New Orleans Section

Clark, F. H.

ELIZABETH

New Orleans Section

Glasgow, C. L.

GRAMERCY

New Orleans Section

Gross, M. F.

GRETNA

New Orleans Section

Paterson, A. B., Jr.

HOUMA

New Orleans Section

Larsh, W. E.

LAFAYETTE

New Orleans Section

Henke, W.
Hughes, G. G.

LAKE CHARLES

New Orleans Section

Calongne, R. J.
Chalkley, H. G.
Fulton, G. R.
Givens, J. I.
Lamb, H. M.
Stokes, C. W.
Whyte, C. B.

LAKE PROVIDENCE

Mid-Continent Section

Hider, G. T.

MADISONVILLE

New Orleans Section

Irwin, D. B.

MARRERO

New Orleans Section

Aldinger, H. K.

MONROE

Mid-Continent Section

James, H. M.
Kroll, J.
Parsons, L. D., Jr.
Ricks, J. L.
Wilenzick, B.

MORGAN CITY

New Orleans Section

Gray, H.
Harlan, J. H.

NEW ORLEANS

New Orleans Section

Abraham, M. C.
Barbarlich, R. P.
Bell, J. L.
Black, J. E.
Breece, G. L.
Brown, H. I.
Brupbacker, B. S., Jr.
Bunker, W. B.
Cardwell, F. D.
Carito, W. A.
Chattey, J. K.
Coleman, H. F.
Colomb, C. F.
Consiglio, J. T.
Crawford, C. C.
Cucullu, L. J.
DeSimone, L. P.
Diefenthal, S. M.
Dublan, S. R.
Du Pre, F. A.
Earl, R.
Gleeson, M. J.
Goller, J. R., Jr.
Grant, A. A.
Green, C. H.
Hadden, C. F.
Hammett, G. R.
Hill, A. M.
Hill, G. W.
Hoots, P. F.
Huey, J. S.
Hultan, K. A.
Jahncke, P. F.
Johnson, W.
Johnston, W. H.
Kammer, K. P.
Kiernan, B. L., Jr.
Kramer, S. F.
Lais, I. M.
Lane, O. W.
LeBlanc, J. H., Jr.
Lee, H. C.
Lockett, R. P.
Lockett, R. P., Jr.
Luehrmann, H.
Mann, I. W., Jr.
Marion, F. I.
Mayer, J. K.
McAfee, H. C.
McLellan, E. A.
Meade, H. E.
Mendow, F. R.
Mercier, J. P.
Mitchell, R. F.
Moodie, D.
Moody, H. N.
Morehiser, J. E., Jr.
Moreland, W. B.
Muller, R. F.
Murphy, E. L.
Nairne, C. L.
Nall, R. L.
Nelson, B. S.
Nelson, L. K.
Nelson, W. S.
Norton, C. P.
Paterson, A. B.
Pender, W. R.
Perine, R. R., Jr.
Perrin, F.
Peterson, A. J.
Pottharst, J. E., Jr.
Purdum, W. C.
Reedy, F.
Reynick, H. F.
Robert, J. M.
Salathe, L. W.
Saunders, W. H., Jr.
Schmitt, G. H., Jr.
Schupp, R. W.
Seago, R. M.
Smith, S. G.
Stancliff, A. D.
Stewart, D. W.
Todd, J. M.
Voelkel, H. W.
Wadge, G. F.
Wait, W. B.
Watson, M. P.
Winship, W. E.
Wyler, C. J.

NORTH BATON ROUGE

New Orleans Section

Duff, R. B.
Grosz, M. H.
Gulick, W. C.

PLAQUEMINE

New Orleans Section

Nadler, R. A.

PORT SULPHUR

New Orleans Section

Blackstone, F. B.
Randle, W. A.
Shaw, J. E., Jr.

RUSTON

Mid-Continent Section

Bogard, B. T.

SHREVEPORT

Mid-Continent Section

Blanchard, A. G.
Breffeilh, G. A.
Carmichael, J. T.
DeLaune, H. L.
Heller, M. M.
McLean, H. D.
Stewart, M. G.

STERLINGTON

Mid-Continent Section

Boardman, C. C.
Phillips, R. L.

SULPHUR

New Orleans Section

Ricketts, R.

THIBODAUX

New Orleans Section

Toups, H. S., Jr.

MAINE

AUBURN

Bates, T. M.

AUGUSTA

Ham, M. T.
Hopkins, C. L.

BANGOR

Dixon, L. S.

BATH

Fulton, C. K.
Newell, W. S.

BELFAST

Mortimer, J. D.

BROWNVILLE

Laverty, H. H.
Stickney, W. H.

BUCKSPORT

Bearce, G. D.

CAPE COTTAGE

Tarashik, N.

CHEBEAQUE ISLAND

Collins, B. R. T.

CUMBERLAND MILLS

Gilman, R. E.
Terry, K. E.

FRYEBURG

Kroner, E. F.

KITTERY POINT

Prescott, R. W.

LEWISTON

Libbey, W. S.

LISBON FALLS

Stephenson, W. K.

MADAWASKA

Overbagh, J. S.

OLD ORCHARD BEACH

Campbell, T. J.

ORONO

Prageman, I. H.
Watson, H. D.

PORTLAND

Adams, C. E., Jr.
Anderson, R. O.
Bell, W. C.
Chute, R. E.
Crosby, R. W.
Merrill, C. J.

ROCKLAND

Thompson, G. L.

RUMFORD

Ahara, E. V.
Stearns, W. J.

SACO

Smith, H. L.

ST. ALBANS

Thorne, R. B.

SOUTH PORTLAND

Elliott, N. R.

STRONG

Starbird, C. V.

WATERVILLE

Redden, C. A.

MARYLAND

ABERDEEN

Baltimore Section

Spang, J. B.
Weiss, H. K.
Wirsing, C. E.

ABERDEEN PROVING GROUND

Baltimore Section

Apgar, J. W.
Bluestone, E.
Brandt, A. E.
Bush, R. M.
Byrne, J. J.
Cook, D. W.
Cottrell, R. B., Jr.
Debski, T. F.
Fagin, S.
Fischbach, J. W.
Gerhard, S.
Goldstein, D. L.
Goodwin, B. S.
Higgins, C. L.
Hillsley, J. J., Jr.
Josephs, L. C., Jr.
McAllister, W. J.
McNeilly, V. H.
Moore, L. S.
Nelson, L. S.
Parker, L. L.
Post, M.
Raninen, A. B.
Simon, W. R.
Steele, M. A., Jr.
Stoeckinger, R. F.
Walden, W. R.
Werner, H. B.
Woods, S. H.

ANNAPOLIS

Baltimore Section

Bacon, R. A.
Bartley, C. O.
Blose, J. F.
Bock, A. E.
Breeden, G. B.
Ferguson, C. K.
Gerstung, H. S.
Hebrank, E. F.
Hobbs, E. E.
Johnson, S. E., Jr.
Johnson, T. W.
Johnson, T. W., Jr.
Johnston, R. M.
Kiefer, P. J.
Kinney, W. F.
Lee, G. H.
Seelaus, J. J.
Senner, A. H.
Smith, H. T.
Welanetz, L. F.
West, J. P.
Yocum, W. F.

BAINBRIDGE

Baltimore Section

Heymann, C. D.
Ryan, J. M.
Smith, R. I.

BALTIMORE

Baltimore Section

(*See also Middle River*)

Abbott, H. P.
Akerman, N.
Allgaier, J. M.
Andrews, C. A.
Austin, W. S.
Baker, J. R.
Baker, S. V.
Balavitch, J. M.
Beck, W. H., Jr.
Becker, W. D.
Benjes, E. McD.
Berner, G. H.
Bessent, C. F.
Bill, R. G.
Birckhead, L.
Black, W. S.
Bohnlofink, J.
Bolgiano, G. F.
Bolin, M. E.
Bond, F. M.
Bowen, W. V.
Boynton, W. D.
Brenner, F. G.
Brilhart, G. L.
Brillhart, S. E.
Bullock, J. B.
Bunker, W. W.
Bunn, E. S.
Burggraf, J. C.
Burrill, H. G.
Carlsrud, R.
Carter, L. E.
Cassidy, H. W. D.
Chambers, E. G.
Chatard, W. M.
Chinn, G. I.
Christie, A. G.
Churchman, H. N.
Collier, W. I.
Colpitts, J. V.
Conn, T. D.
Cromwell, O. C.
Cullen, T. J.
Cutler, J. B.
Dalrymple, A. W.
Dankenbring, H. D.
Dannettel, R. C.
Davis, J. C.
Dawe, W. L., Jr.
DeCesare, R. J.
Delano, R. P., Jr.
Dennis, B. W.
Deringer, B. W., Jr.
Derrickson, G. W.
Dischinger, H. R.
Duncan, J. R.
Dupont, A. T.
Eberhart, J. M.
Eckberg, H. F.
Eggert, E. H.
Egli, H.
Einwaechter, F. H., Jr.
Engel, F. C.
Ergler, P. C.
Fambro, G. W.
Frankena, A.
Galloway, A. K.
Gardner, L. H.
Gliss, G. E.
Goller, G. N.
Gompf, A. M.
Green, R. J.
Gregory, R. S.
Hanhart, E. H.
Harding, R. A., Jr.
Harris, G. S.
Hartman, L. R.
Hassett, R. J.
Hawkins, E. C.
Hazlett, W. A.
Healy, G. F.
Herbert, L. E.
Hertenstein, E. W., Jr.
Higgins, N. B.
Hildenbrand, C. F.
Hilprecht, R. C., Jr.
Hine, R. C.
Hofstetter, E. T. O.
Hollerith, H., Jr.
Hooper, W. U.
Horlebein, E. W.
Howard, J. E.
Hughes, R. LeR., Jr.
Hyde, H. W.
Ives, A. H., Jr.
Jeffers, F. J.
Jesatko, J. M.
Keen, G. W.
Keppelman, H. S.
Knauss, G. E.
Knoedler, E. L., Jr.
Koellish, W. M.
Kohut, F. J.
Kouwenhoven, F. W.
Lane, D. F.
Lanham, C. W., Jr.
LeGates, F. E.
Leilich, F. T.
Lubbert, G. L.
Ludlum, W. J.
Macaluso, F. L.
Madey, E.
Mahaffy, H. M.
Malakoff, N. H.
Malone, J. F.
Matlat, G. W.
McDaniel, W. N.
McLane, R. M.
Merriam, C. F.
Merrick, C. M., III
Meyer, C. W.

Michetti, A. L.
Miller, E. E.
Morton, A. W.
Mousson, J. M., 2nd
Myers, J. B.
Naylor, H. A., Jr.
Nelson, N. T.
Nitka, W. A.
Nitze, C. V.
Nopper, R. E.
Olver, F. M.
Packham, E. T.
Passano, W. M.
Paugh, C. T.
Peale, W. O.
Penniman, A. L., Jr.
Perkins, W. F.
Petruccione, A. F.
Phelan, J. J., Jr.
Phillips, M. E., Jr.
Porter, J. H.
Posey, J.
Potter, J. H.
Powell, P. R.
Powell, S. T.
Proctor, W. E.
Quarles, F. W.
Quintilian, B. F.
Rabert, A. P.
Rach, J. L. W.
Robertson, S. F.
Roy, R. H.
Ruzga, J. A.
Ryan, E. J.
Schlitt, J. L.
Schluderberg, C. C.
Schmeisser, E. G.
Schneider, R. F.
Severn, H. A.
Sevier, H. A.
Sexton, S. B., III
Shea, J. R.
Sheridan, H. G.
Shoemaker, A. F.
Shure, W. H. N.
Slater, C. A., Jr.
Smith, L. G.
Smith, W. G.
Snyder, S. M., Jr.
Spamer, A. M.
Spear, A. I.
Steiner, K.
Stevens, W. R.
Stone, A. R.
Straus, W. R.
Taylor, A. W.
Taylor, R. M.
Temple, E. A.
Thomas, R. L.
Trefethen, L. S.
Vaughn, E. W.
Vodvarka, F. J.
Wager, R. C.
Wagner, D. E.
Wagner, H. C., Jr.
Wagner, R. J.
Wannall, G. L.
Warfield, H. R., Jr.
Weitzel, W. F.
Wells, H. S., Jr.
Wells, R. C.
Whitaker, C. H.
Whitman, E. B.
Whitsitt, W. B.
Willis, A. H.
Wilmer, J. W.
Wilson, E. P.
Wolfe, T. R., III
Woodburn, J.
Woodruff, De F. D.
Woodward, H. W.
Worthington, J. A.
Wright, P. K.
Yellott, J. I.
Young, J. W.
Zimmerman, R. E.
Zink, J. H.
Zouck, G. H.

BEL AIR
Baltimore Section

Addicks, L.
Josephs, J. P.

BERWYN
Washington, D. C. Section

Hartman, J. A.

BETHESDA
Washington, D. C. Section

Allen, W. G.
Cottrell, N.
Fitzsimmons, A. M. R.
Lathrop, R. P.

CAMBRIDGE

Whitham, J. M.

CAMP RITCHIE

Chaharyn, O. D.

CARDE ROCK
Washington, D. C. Section

Herrmann, E. M.

CATONSVILLE
Baltimore Section

Andrews, J. T., Jr.
Berman, R.
Elder, J. D.
Lund, C. R., Jr.
Wolfe, R. F.

CEDARHURST
Baltimore Section

Budell, W.
Libman, F. H.

CHESTERTOWN

Owens, C. T.

CHEVY CHASE
Washington, D. C. Section

Bieber, V. E.
Hawks, A. S.
Hopkins, J. D., Jr.
Wilson, L. G.

COLLEGE PARK
Washington, D. C. Section

Green, W. P.
Hayleck, C. R., Jr.
Hoshall, H. B.
Jackson, J. W.
Rivello, R. M.
Shreeve, C. A., Jr.
Younger, J. E.

CONOWINGO
Baltimore Section

Lefever, P. M.
Turner, R. E.

CUMBERLAND

Alexander, R. I.
Braccia, A. A.
Breidenstein, L. W.
Lyons, B. J.
McKaig, W. W.
Mitchell, J. G.
Pliner, N. S.
Plunkett, B.
Reddy, D.
Rowley, A. W.
Wherritt, C. R.

EDGEWOOD
Baltimore Section

Huss, H. O.
Perry, A. E., Jr.

ELKTON

King, S. L., Jr.
Thomson, F. duP.

ELLICOTT CITY
Baltimore Section

Funke, R. H., Jr.

FEDERALSBURG

Owens, H. M.

FT. GEORGE G. MEADE

Ashmead, R. R.
Eitel, W. W., Jr.
Fischer, R. J.
Kalmbach, C. F.
Youtie, R. K.

FREDERICK

James, D. T.

HAGERSTOWN

Gordon, E.
Lindner, B. A.
Triplett, J. L.
Vedder, W. O.
Watkins, C. A.

HAVRE DE GRACE
Baltimore Section

Doukas, G.
Montgomery, C. D.
Williams, D. R.

HYATTSVILLE
Washington, D. C. Section

Banks, T. K.
McFarland, J. D., Jr.

HYDES
Baltimore Section

Hendrickson, O. Q.

KENSINGTON
Washington, D. C. Section

Stresau, R. H. F., Jr.

LANHAM
Washington, D. C. Section

Carlsson, C. A. V.
Lanham, P. T.

MANCHESTER
Baltimore Section

Jones, C. W.

MIDDLE RIVER
Baltimore Section

(*See also Baltimore*)

Espy, M. P.
Gaebler, G. F.
Greenwood, O. W.
Hook, G. R.
Hopping, R. L.
Loughney, C. E., Jr.
Maynard, W. G.
Penney, R. W.
Schaidt, L., Jr.
Woersching, T. B.

PATUXENT RIVER
Baltimore Section

Brown, C. B., Jr.
Smith, F. T.
Stuber, B. V.
Switack, J. W.
Wallis, H. L.

POOLESVILLE
Washington, D. C. Section

Pole, L. A.

REISTERSTOWN
Baltimore Section

Munsell, D. D.

RELAY
Baltimore Section

Goelkel, W. L. W.
Wheeler, R. C.

RIVERDALE
Washington, D. C. Section

Krafft, J. M.
Lomonossoff, B. M.
Samburoff, S. N.
Weick, F. E.

ROCK HALL

Hatch, C. H.

ROCKVILLE
Washington, D. C. Section

Bralove, W., Jr.

SALISBURY

Bodmer, J. E.

SILVER SPRING
Washington, D. C. Section

Falk, G. E.
Hoagland, J. C.
Kessinger, J. A.

SOLOMONS

Braunschneider, F. J.

SPARROWS POINT
Baltimore Section

Anderson, R. C.
Cichelli, A. E.
Clarke, C. E.
Coffin, L. F.
Cox, F. O., Jr.
Davis, E. C.
Donovan, D. E.
Fox, W. J.
Good, S.
Hill, W. P.
Knabe, F. S.
Knust, H. R.

TAKOMA PARK
Washington, D. C. Section

Fisher, E. D.
Sturm, G. W.

TOWSON
Baltimore Section

Bartholomaei, H. A.
Beyer, J. R.
Forman, A. H.
Lyness, A. A., Jr.
Middleman, D.

WESTMINSTER
Baltimore Section

Hausman, S.

WHITE OAK
Baltimore Section

Zimmerman, F. L.

MASSACHUSETTS

ANDOVER
Boston Section

French, E. V.

ASHLAND
Boston Section

Campbell, J. R.
Saurwein, G. K.

ATHOL
Worcester Section

Butler, H. M.
Horigan, J. H.
MacKay, S.
Simpson, R. J. O.
Wise, R. T.

ATTLEBORO
Providence Section

Cole, R. W.
Cutler, A. E.
Graham, F. C.
Mossberg, F.
Olson, R. V.
Phillips, H. S.

AUBURNDALE
Boston Section

Parker, F. A.

AVON
Boston Section

Berwick, J. D.

BABSON PARK
Boston Section

Newton, R. C.

BASS RIVER
Boston Section

Davis, C. H.

BELMONT
Boston Section

Bragaw, C. G., Jr.
Flanagan, J. H.
Moultrop, I. E.
Vaughan, J. F.

BEVERLY
Boston Section

Abel, W. L.
Boothroyd, H. W.
Brand, R. S.
Cross, G. P. S.
Eastman, F. C.
Houghton, W. M.
Ireland, R. W.
Linderoth, L. S., Jr.
Severy, L. A.
Smith, J. F. D.
Talbourdet, G. J.

BOSTON
Boston Section

Alexander, W. T.
Allen, E. F.
Ames, J. B.
Andrews, B. R.
Archibald, F. L.
Atkinson, K.
Baggs, A. E., Jr.
Bailey, Miss E. H.
Bailey, R. J.
Bailey, W. S.
Baird, H. B.
Bance, E. S.
Barry, E. H.
Bayliss, W. A.
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Myhre, E. B.

EUREKA

Rebaleati, J. N.

McGILL

Mansfield, W. M.

RENO

Hutton, J. O.
Van Dyke, J. R.

NEW HAMPSHIRE

BARTLETT

Green Mountain Section

Jenkins, J. G.

CENTER SANDWICH

Green Mountain Section

Larsson, T. L. F.

CLAREMONT

Green Mountain Section

Osgood, C. F.

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Hadlock, W. K.
Waymouth, G. W.
Willoughby, V. R.

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Nye, E. P.
Seeley, L. E.
Webber, L. E.

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Moulton, R. G.

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McLaughlin, R. J.

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Tomkinson, S. E.

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Hunt, S. P.
Jerauld, C. T.
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Szepan, H. F.

NASHUA

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STRATHAM
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WILTON
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NEW JERSEY

ALDENE
Plainfield Section

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ALLENHURST
Metropolitan Section

Haaren, C. F.

AMPERE
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Rathe, A. W.

ANDOVER
Metropolitan Section

Horton, R. D.

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Chelgren, W. J.
Goodwin, L.
Gunther, C.
Kennedy, E. C.
Kerr, J. H.
Kessler, M.
Knox, S. L. G.
Lawson, W. O.
Llewellyn, W. E.
McIlhenney, W.
Pattershall, D. S.
Zeppenfeld, C. W.
Zeppenfeld, S. D.

ASBURY PARK
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Strouse, B. H.

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Glas, A.

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Buschmann, P. T.
Cross, L. F., Jr.
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Frigiola, N. F.
Gagnebin, A. P.
Gamble, W. W.
Garson, T. N.
Hammond, J. D.
Krol, P. F.
Leavitt, G. E., Jr.
Mack, J. J., Jr.
McIntyre, M.
Mitchell, H. C.
Nigh, G. W.
Searl, J.
Stanfield, R. S.
Tarzy, A. R.
Transou, A. J.

BEDMINSTER
Metropolitan Section

Beekman, H. M.

BELLEVILLE
Metropolitan Section

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Ball, W., Jr.
Bayles, C. B.
Bush, W. W.
Clark, G. C., Jr.
Corbett, C. H.
Doring, W. F., Jr.
Joeckel, S. V.
Klotz, W.
Niclaus, W. L.
Norcross, G. F.
Ogur, E.
Olschner, H. J.
Orsini, A.
Palmer, L., Jr.
Schmidlin, A. E.
Wilkens, J. A.
Woolley, H. O., Jr.
Zarnowski, F. J.

BELMAR
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Birdsall, H. C.
Frazier, Q.
Gordon, A.
Kaprelian, E. K.
Stoecker, R. J.
Zorwitz, J.

BERNARDSVILLE
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Frost, R. B.

BLOOMFIELD
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Bayles, W. H.
Buckalter, R. I.
Ciricillo, S. F.
Cygan, R.
Downey, S. F., Jr.
Drogue, J. A.
Dunnican, G. W.
Foote, W. R.
Gesell, W. H.
Haskin, L. H., Jr.
Hazard, C. S.
Herbst, R. S.
Johnson, W. E.
Karsten, W. F. R.
Knaus, W. L.
Levenhagen, F. H.
Loewen, W. L.
Male, R. LeF.
McLenegan, D. W.
Meyerhaus, R. H.
Miller, W. A.
Nascenzi, F. L.
Plass, R. H.
Price, T.
Roessler, E. W.
Rossmann, P. F.
Singer, J. H.
Sollmann, C. H.
Spofford, W. A.
Teague, H. M.
Tucker, C. L.
Unkles, E. H.
Urban, F. O.

BOGOTA
Metropolitan Section

Wyburn, W.

BORDENTOWN
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BOUND BROOK
Plainfield Section

Ahrens, C.
Andres, G. J.
Austin, R. S.
Bartlett, R. W.
Bean, C. H.
Bigelow, C. M.
Brunyate, W. R.
Carey, R. H.
Davidson, J. I.
DeVilbiss, T. A.
Greenwood, L. E.
Gunderson, G. C.
Guyette, D. F.
Hebenstreit, C. G.
Heinz, W. B.
Horne, A. W.
McMurray, J. H.
Moffatt, C. R.
Phyl, J.
Reeve, K. A.
Rohrhurst, W.
Siemon, K. O.
Thornton, H. E.
Traut, F. L.
Yaroshuk, W. R.

BRADLEY BEACH
Metropolitan Section

Irey, G. R.

BRANCHVILLE
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Overton, W. J.
Stotter, A. H.

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Hoffer, H. A.
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Karg, W. E.
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Wade, E. A.

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Anderson, E. L.
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Booton, G. H.
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Coliz, J. T.
Cottle, H. N., Jr.
Easton, E. M.
Fowler, F. H., Jr.
Genthon, F. X.
Hall, J. L.
Kalustian, E.
LeBarre, F., Jr.
Le Compte, F. M.
Peterson, W. J.
Schulte, W. C.
Sepic, J. B.
Wood, C. R.

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Blair, J. A.
Bower, H. S.
Boyd, W. W.
Bradshaw, G. V., Jr.
Brousseau, E. W.
Campbell, D. S.
Cleaver, W. H.
Felch, R. I.
Fitts, J. L.
Frankhouser, E. V.
Ginsparg, J. O.
Goble, G. H.
Hanson, M. E.
Harbeson, J. P., Jr.
Hilderbrand, E. A.
Hutchinson, D. M.
Hynes, L. P.
Jacobs, L.
Jorgensen, W.
Kearney, H. M.
Laiming, H. J.
Langworthy, W. P.
Leigh, E. D.
Lipman, R. N.
Lofft, W. A.
Longmaid, S. E.
Marshall, H. F.
Miller, J. F.
Mueller, C. A.
Orloff, S.
Pawlowski, B. J.
Rhine, C. K.
Schaaf, J. W.
Schock, H. E., Jr.
Stewart, J. G.
Teaf, J. H.
Turner, M. C.
Wachs, T.
Weissbach, E. A.
Werley, H. R.
Weschler, W. S.

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Popiel, R.

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Howard, W. M.

CLIFFSIDE PARK
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Di Lorenzo, F. J.
Parma, E. J.
Seidler, M. F.
Smyth, S.

CLIFTON
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Guenther, M. J., Jr.
Higbie, V.
Jaxon, L. H.
Kaplan, E. S.
Koenig, L., Jr.
Northup, R. W.

COYTESVILLE
Metropolitan Section

Hamilton, T. H.
Sikosek, F. J.

CRANFORD
Plainfield Section

Brescka, R. S.
Garden, J. MacK.
Gillespie, C. W.
Miller, E. H.
Spencer, F. C.
Whittaker, H.

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Harmstad, J. E.
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Stradley, G. C., Jr.
Thompson, C., Jr.

DELAIR
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Bowers, R. H.

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Crane, H. O.
De Luca, A. P.
Lukens, W. L.
Mailler, J. P.
Rutkovsky, H. D.
Thuerk, H. C.
Wallace, A. L.
Willett, F. M.

DUNELLEN
Plainfield Section

Richmond, E. L.
Schwartz, A. A.

EARLE
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Nieman, H. C.

EAST NEWARK
Metropolitan Section

Berry, G. W.
Betzler, H. W.
Jelleme, W. Q.

EAST ORANGE
Metropolitan Section

Armstrong, C. E.
Bato, A. A.
Bird, S. P.
Cook, W. D.
Crede, R. E.
Durand, N. C.
Kidd, G. F.
Law, C. J.
Libby, S. H.
Neff, J. P.
Praner, J. A.
Reilly, J. H.
Rowley, W. R.
Stewart, E. A.
Tallmadge, W.
Thomson, A. MacF.
Tillquist, D. E.
Turno, W. G.

EAST PATERSON
Metropolitan Section

Shayne, A.

EAST RUTHERFORD
Metropolitan Section

Cawley, G.
Nankivell, J. E.
Peter, B. H.
Zinn, H. R.

EATONTOWN
Metropolitan Section

Balcom, J. A.

EDGEWATER
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Letterman, F. L.

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Plainfield Section

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Bates, D. P.
Bauer, J. L.
Bonnett, L. B.
Caruthers, E., Jr.
Chaffe, W. H.
Conran, F. M.
Daubner, R. E.
Dunne, S. B.
Eberle, F. H.
Ely, A. J.
Fetske, W. A.
Foster, J. S.
Fox, F. W.
Gibson, H. D.
Gilliam, H. H.
Glimm, W. F.
Hansen, H.
Keesing, M.
Kelly, V. Q.
Kievit, W. M.
Kopf, W. F.
Leggett, J. R.
Mackay, G. W.
Matheisel, R. A., Jr.
Metzner, B. C.
Milazzo, J.
Moore, H. F.
Newcomb, F. LeR.
Nulle, J. H.
Nydegger, P. F.
Oestnaes, V. L.
Osterman, P. C.
Peets, W. J.
Pegler, J. L.
Ritter, H.
Schalscha, W. G.
Schnackel, H. C.
Steele, T. W.
Teitelbaum, A.
Thompson, W. H.
Tyroff, C. E.
Walmsley, G.
Wilson, W. G.

ELIZABETHPORT
Plainfield Section

Chason, D. H.
Dixon, W.
Melzig, A. H. J.
Nohse, W. R. E.
Young, G. O.

ENGLEWOOD
Metropolitan Section

Bates, C. J.
Black, D. R.
Cummiskey, W. M.
Johnson, N. C.
Leach, C. H.
Roberts, M. H.
Taplinger, J. R.

FAIR LAWN
Metropolitan Section

Burke, A. J.
Heinrich, C. J.
Simpson, J. H.

FANWOOD
Plainfield Section

Wilson, J. P.

FLEMINGTON
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Dunbar, N. C.

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Brown, W. A.
Erhardt, W. L.

FORDS
Plainfield Section

Fritz, C.
Kuprick, W.

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Hultberg, A. J.
Kuszczak, L.
Neilson, G. E.

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Cauble, G. B.
Sauby, W. O.

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Roome, G. R. S.

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Ruddick, R. B.

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GLEN GARDNER
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GLEN RIDGE

Boyd, W. W.
Whitesel, H. A.

GLEN ROCK
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Taylor, K. H.

GLOUCESTER CITY
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GRASSELLI
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Heger, E. F.
Zepht, E. E.

HACKENSACK

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Cohen, L.
Roth, W. E.
Wurtz, R.
Zimmerman, H. T.

HACKETTSTOWN

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Hirschberg, C. A.

HADDONFIELD

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Smith, W. E.

HADDON HEIGHTS

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Kammerhoff, H. H. M.

HARRINGTON PARK

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HARRISON

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Assaykeen, I. V.
Avigdor, R.
Bacha, C. P.
Basche, H. S.
Baser, F. R.
Bean, P. H.
Bischoff, C. A.
Borchardt, A. H.
Clement, G. P.
Collar, C., Jr.
Davis, Miss L. F.
Diserens, P.
Dorer, O. H.
Ewer, R. G.
Feld, L.
Fleming, W. M.
Garnar, L. H.
Grace, J. F.
Grobholz, R. K.
Guidon, M., III
Habach, G. F.
Hall, R. D.
Hendrie, C. F.
Hepburn, J. W.
Herman, T. A.
Hodge, G. N.
Imbornoni, A. J.
Karassik, I. J.
Lancaster, J. E.
Littlefield, W. V.
Mallon, D. J.
McBath, B. R.
McDonald, D. E.
Mockridge, C. R.
Moodie, W. C.
Nardone, P.
Naughton, F. U., Jr.
Oschwald, A., Jr.
Otterbein, M. E.
Palmer, V. M., Jr.
Peterson, A. I.
Pyne, F. S.
Raborg, J. H.
Reynell, C.
Rives, T. McD., Jr.
Schwanhausser, E. J.
Sizer, W. D.
Swanson, N. E.
Uhde, W.
Vlahakes, J. L.
Voss, W. F.
Waddell, C. L.
Watson, R. M.
Wright, E. F.

HAWTHORNE

Metropolitan Section

Pfromm, J. M.

HIGH BRIDGE

Alexander, E. E.
Hanks, G. R.
Stires, W. H.

HIGHLAND PARK

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Meseroll, V. F.

HILLSIDE

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Brengel, F. J.
Fimbel, P. N.
Goryl, W. M.
Hebler, W. O.
Strecker, C. E.
Willson, T. E.

HOBOKEN

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Arnoldi, R. A.
Ashton, R.
Bailey, R. R.
Bilyk, M. H.
Crawford, E. A.
Cuddeback, A. E.
Dalessio, J.
Danatos, S.
Davidson, K. S. M.
Davis, H. N.
Deimel, R. F.
Ehrman, B., Jr.
Erhard, W. A.
Fezandié, E. H.
Furutani, G. J.
Gunther, C. O.
Gustavsen, E.
Heitz, R. L.
Hugli, W. C., Jr.
Johns, H. B.
Jones, R. M.
Juelss, D.
Keuffel, A. W.
Keuffel, C. W.
Kimmich, H. S.
Klein, B. A.
Knopp, W. V.
Koenig, E. H.
Kotar, F. J.
Levine, A.
Lewisohn, R., Jr.
McNear, W. F.
Memory, N. H.
Moser, K. J.
Nelson, K. E.
Pineles, J.
Pohl, R. G.
Rodriguez, C. J.
Scavullo, J. J.
Smack, J. C.
Sperry, E. A., Jr.
Sutherland, W. H.
Vogel, E.
Weber, F. A.

HOHOKUS

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Miedendorp, H., Jr.

HOPATCONG

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Bronk, A. H.

IRVINGTON

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Bonanno, J. L.
Davenport, G.
Eberhardt, H. E.
Eberhardt, W. C.
Gloss, E. A.
LaTour, F. G.
Lauer, A. W., Jr.
Moesinger, F., Jr.
Pettit, J. L.
Pickett, L.
Rudiger, B. W.
Schantz, E. A.
Svenson, R. H.
Turner, L.
Wasser, R.
Zademach, E. R.

JERSEY CITY

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Angarano, J. A.
Benzien, F.
Bliss, C. De K.
Bouton, E. M.
Bryant, W. W.
Bunn, W. B., Jr.
Calman, J.
Cash, S. A.
Christy, W. G.
Cokelet, W. V. W.
Crowe, J. J.
Dalton, T. N.
Dobkin, H.
Drucker, J. H.
Eckert, H. R.
Fink, M.
Flar, H.
Foley, J. A.
Foster, R. B.
Francis, E.
Frank, M.
Gamberton, J. H.
Garrison, U. B.
Glauch, E. S.
Goerg, E. F.
Goldstein, I. R.
Hodges, J. L.
Hollowell, J. S.
Hopkins, R. K.
Huber, A. L.
Jacobs, J. A.
Jones, H. L.
Keese, D. L.
Kidd, A.
Klaus, A. J.
Konheim, H. S.
Kornylak, A. T.
Koven, G. H.
Ledgett, L. A.
Lee, L. O.
Lichtenstein, V., Jr.
Manny, E. H.
Marshall, D. McM.
Mayers, M. A.
Moro-Lin, J. J.
Mrvosh, J.
Nichols, C. R., Jr.
Nickelsen, C. J.
Nott, A. J.
Nygren, J.
Oakley, A. W.
O'Brien, J., Jr.
Pearce, L. F.
Pond, R. K.
Pozarycki, H. J.
Reddert, E. J.
Ross, A. M.
Rossheim, D. B.
Salemme, V. J.
Sartorius, W.
Scarano, R. M.
Schoerke, D. A.
Schwartz, A.
Scott, W. E.
Simpson, R. H.
Spray, E. L.
Svenson, E. B.
Tompkins, H. D.
Troger, G. F.
Ulrich, F. L.
Zahka, A. G.
Zullo, S.

KEARNY

Metropolitan Section

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Aborn, R. H.
Alden, J. L.
Ashworth, J. V.
Banton, M. W.
Bauder, B. E.
Bessemer, M. E.
Charlesworth, R. B.
Coleman, K. S.
Corbett, C. W.
Dolesh, F. J.
Fegel, A. C.
Foster, N. S.
Franz, E. E.
Gibbons, P. L.
Gioia, J. C.
Girardi, G.
Gore, R. T.
Griffin, G.
Holmberg, C. G.
Katelus, G. J.
Kutzelman, E. G.
Mayo, A. R.
McCarthy, R. H.
Menkes, S. B., III
Merwin, H. H.
Michal, F. W.
Morris, B. F.
O'Gureck, R. M.
Quackenbush, W. R.
Rahikka, R. E.
Roeder, G. P.
Roscoe, H. W.
Schenk, E. M.
Semble, S.
Shopp, C. J.
Sievers, E. J. J.
Smith, R. C.
Spencer, F. C., Jr.
Tenety, J., Jr.
Troger, H. H.
Volckhausen, W. J.
Waters, D. V.
Weaver, C. R.
Wilkinson, A. S.
Zeliff, D. E.

KENILWORTH

Plainfield Section

Barker, V. D.

KEYPORT

Metropolitan Section

Wisnik, J.

LAKEHURST

Bailey, W. H.
Hillendahl, W. H.
Snow, R. M.

LEONIA

Metropolitan Section

Ayars, W. S.
Eden, F. L.
Eger, G. W., Jr.

LINDEN

Plainfield Section

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Carpenter, M. S.
Goss, R. J.
Greenfield, B.
Grygotis, W. J.
Heffernan, W. H.
Hubeny, F. G.
Isles, F. W.
Kurzinski, E. F.
Lawrance, C. L.
Lichtenstein, I.
Pennock, D. W.
Potter, J. D.
Priff, G. N.
Solomon, J. W.
Spooner, F. M.
Ur, J. A.

LINWOOD

Philadelphia Section

Trethaway, W., Jr.

LITTLE FALLS

Metropolitan Section

Denzler, R. E.

LYNDHURST

Metropolitan Section

Boettiger, R. W.
Bravo, D. S., Jr.
Fuhro, A. A.
Hawkins, A. E.
Jack, G.
Kennedy, W. M.
Leslie, J. S.
Linden, G. R.
Meinhart, F. J.
Savage, K.
Scelba, A. J.
Soderberg, S.

MADISON

Metropolitan Section

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Olofson, E. C.

MAHWAH

Metropolitan Section

Martinson, R. A.
Thornton, A. F.
Wilson, R. L.

MANVILLE

Plainfield Section

Bradley, C. B.
Heisler, R. W.
Koch, H. G.
Sargent, R. N., Jr.
Terry, C. G., III

MAPLEWOOD

Metropolitan Section

Appe, R. C.
Carlton, J. R.
Carroll, P., Jr.
Cullen, W. J., Jr.
Darby, J.
Eberhardt, U. S.
Hesse, W. K.
Hunter, C. W., Jr.
Leonard, M. W.
Maxwell, H. S.
Rausch, V. H.
Richardson, G. P.
Rohrer, A. L.
Vassar, H. S.

MATAWAN

Metropolitan Section

Fidelius, W. R.

MAYWOOD

Metropolitan Section

Bruns, R. S., Jr.

MERCHANTVILLE

Philadelphia Section

Gagliardi, F. A.
McQuilkin, G., Jr.
Robeson, P. B.

METUCHEN

Plainfield Section

Kingman, W. W.
Miller, E. E.
Neumann, A.
Peterson, A. W.

MIDDLEBUSH

Metropolitan Section

Conover, G. W., Jr.

MIDDLESEX

Metropolitan Section

Branin, F. S.

MILLBURN

Metropolitan Section

Brice, N. E.
Fay, C. H.
Nelson, P. H.
Sampson, R. G.
Staub, M. H.

MILLTOWN

Metropolitan Section

Herbert, J. S.
Keller, L. F.
Ness, I. S.

MILLVILLE

Philadelphia Section

Krause, J. L.
Latham, G. R., III
Lewis, D. D.
Mason, J. R.
West, R. F.

MONTCLAIR

Metropolitan Section

Allen, F. B.
Braine, B. G.
Forstall, A. E.
Gilbreth, L. M.
Hull, R. D.
Iliff, W. L.
Jacobus, D. S.
Kaveny, J. G.
Kent, R. T.
Klein, R.
Lawrence, R. A.
Melcher, C. H.
Persak, G., Jr.
Prudden, P. D.
Rigdon, C.
Schlachter, C. H.
Sherman, G. L.
Turner, W. W., Jr.

MOORESTOWN

Philadelphia Section

Atkinson, J. H.
Nichols, E. B.

MORRIS PLAINS

Metropolitan Section

Gutzwiller, E. C., Jr.

MORRISTOWN

Metropolitan Section

Fitzpatrick, J. T.
Harding, A.
Kihn, W. J.
Kosley, R. W.

MOUNTAIN LAKES

Metropolitan Section

Beck, C. W.
Borcherdt, W. O., Jr.
Brock, F. C.
Thompson, O. C.
Vincent, A. S.
Wilson, J. R., Jr.

MOUNTAINSIDE

Metropolitan Section

Spainhour, C. D.

MT. HOLLY

Philadelphia Section

Denworth, H. S.

MT. TABOR

Metropolitan Section

Hickerson, F. R.

MURRAY HILL

Metropolitan Section

Baratta, H. E.
Elmer, L. A.
Shaw, P. A.
Smith, A. K.

NEWARK

Metropolitan Section

Adams, H. R.
Alden, J. D.
Aldrin, E. E.
Anderson, C. E.
Annett, E. B.
Bartsch, A. G.
Bauhan, A. E.
Beltle, W. O., Jr.
Bennett, J. W.
Berger, J. G.
Billings, J. E.
Birkland, S.
Blanchard, A. H.
Bohn, L. G.
Boniface, J. B.
Books, T. V.
Bostock, R. N.
Breckenridge, F.
Brinkman, C. F.
Brush, H. F.
Burling, H. S.
Carey, P. C.
Carr, H. R.
Carvin, F. D.
Cassedy, W. F., Jr.
Chandler, W. C.
Coates, W.
Cohan, A. M.
Conkling, W. C.
Conran, F. M., Jr.
Cullimore, A. R.
Davis, V. F.
De Bodor, A.
Diamant, S.
Dieter, W.
Dobson, J. G.
Dong, J. C.
Douwes, H. B.
Eberhardt, F. E.
Eisler, C.
Eisler, C., Jr.
Ely, R. G.
Fairchild, F. P.
Farrington, S. G.
Felber, G. S.
Fickes, A. C.
Fink, F.
Fitzsimmons, J. H.
Fleming, D. H., Jr.
Forbes, F. P.
Foster, E.
Frohboese, R. H.
Frost, V. M.
Funk, H. J.
Garey, L. L.
Gaston, W. I.
Gay, F. W.
Golden, R. F.
Grandin, F.
Granell, E. C.
Gray, H. A.
Groff, H. M.
Grosswendt, C. J.
Haag, H. C.
Hamilton, S.
Hansen, A.
Harrison, L. W.
Harrison, M. R.
Hausmann, W.
Heath, A. R., Jr.
Heller, E. W.
Helmstaedter, W. E.
Henderson, B. D.
Herron, W. L.
Hertz, D. B.
Higgins, J. T.
Hodgkinson, G. A.
Holleran, D. J.
Hope, R. DeV.
Horn, W.
Huettl, W. A.
Ingalls, D.
Jaffe, W. J.
Jansson, J. H.
Johnson, R. P.
Jordan, W. A.
Kearney, J. G.
Keener, H. J.
Keith, P.
Kennedy, P. S.
Kielczewski, E. L.
Kluesener, H. H.
Knezo, J., Jr.
Kubli, R. A.
Kuttler, J. B.
La Motte, W. R.
Lawson, J. T.
Ledden, E. B.
Lett, L. N.
Lewis, D. R.
Lipke, L. H.
Lipschultz, H. L.
Lipton, M. H.
Lister, A.
Loomis, E. G.
Lotz, L. C.
Loutrel, C. H.
Lusk, S. W.
Lynn, E. W.
Matte, H. P.
McKinty, J.
Meyer, J. A.
Meyer, P.

Moss, H. H.
Muench, E. J.
Muench, F. J., Jr.
Murphy, E. R.
Murphy, G. F.
Murphy, W. J.
Myers, C. G., Jr.
Nardone, R. M.
Oladko, Miss H.
O'Lenick, A. J.
O'Neill, J. L.
Otzmann, H., Jr.
Page, J. F.
Pausin, H. R.
Peff, I.
Phelps, F. A.
Pisarchik, H. G.
Polaner, J. L.
Pope, C. J.
Potter, P. A.
Prupis, R. I.
Rabinow, D.
Rahm, F. W.
Redmon, J. K.
Reichert, W. G.
Reilly, J. W.
Renton, V. S.
Runyon, M. E.
Russell, F. L.
Sage, D.
Salmon, P. A.
Santucci, E. D.
Schaub, H. W.
Scher, G.
Scherer, H. A.
Schneider, T. A.
Shaw, L. E.
Signorelli, A. A.
Skarbek, H. F. J.
Smith, A. T.
Smith, Miss I. M.
Snyder, E. H.
Soldan, H. M.
Somers, W. E.
Thomson, J. S.
Vasselli, A. J.
von Fabrice, R.
Walter, H. E.
Walton, E.
Wasserlauf, B. C.
Watkinson, R. M.
Watts, R. L.
Weisberg, H.
Westin, W. C.
White, J. W. H.
Wiener, M.
Williams, C. A.
Wilson, H. R.
Wood, W. C.
Woolson, C. G.
Wottrich, H.
Wygovsky, R. N.
Wyllie, J. S.
Young, D. L.

NEW BRUNSWICK
Metropolitan Section

Anthony, R. L.
Baier, E. W.
Bassett, W. G. R.
Beshers, H. M.
Blackman, V. C.
Cejka, J. B.
Creager, P. S.
Dunkle, H. H.
Ferrara, N.
Franke, K. J.
Gaum, C. G.
Goldman, J. H.
Heath, C. O., Jr.
Heck, R. C. H.
Holland, U. C.
Morgan, B. D.
Murayama, S.
Nichols, A.
Quackenbush, E. S.
Sadwith, H. M.
Taub, P. H.
Waldron, J. H.

NORTH BERGEN
Metropolitan Section

Bonini, J. D.
Casamento, J. R.
Cass, L. J.
Ives, C. B.
Orlando, P. J.

NORTH PLAINFIELD
Plainfield Section

Grosswendt, C. T.

NUTLEY
Metropolitan Section

Irion, W.
Nichols, D. K.
Oakley, H. C.
Revere, J. P.
Warwick, T. M.
Wilkenfeldt, J. W.

OAKLAND
Metropolitan Section

Hughes, R. G.

OAKLYN
Philadelphia Section

Mengers, C. F.

ORADELL
Metropolitan Section

Grube, W. McQ.

ORANGE
Metropolitan Section

Antalec, F. A.
Britten, C. R.
Dickson, C. H.
Eddy, W. T.
Fisher, W. W.
Hutchison, F. P.
Riegels, O. L.
Warthman, K. L.

PALMYRA
Philadelphia Section

Fowler, A. J.

PARAMUS
Metropolitan Section

Mackal, H. H.

PARLIN
Metropolitan Section

Braswell, C. C.
Carell, W. S.
Cleary, J. W.
Johns, W. L.
Johnson, K. H.
Mahen, K. W.

PASSAIC
Metropolitan Section

Balter, H. M.
Bradley, F. L.
Christmann, J. L.
Daskin, W.
DenBleyker, E. T.
Edwards, G. M.
Gajarsky, J. E.
Kent, F. J.
Kleinschmidt, K. T.
Levat, S. G.
Lombardi, E. F.
McIlveen, E. E.
Murray, A. W.
Nordt, P. W., Jr.
Northrup, F. B., Sr.
Perkins, W. E.
Scoville, W. E., Jr.
Sedgwick, E. H.
Starke, W. W.

PATERSON
Metropolitan Section

Beall, A. L.
Bell, K. D.
Brown, W. R.
Constantino, C. S.
Cook, J. W.
Czekalski, W. M.
Davis, B. H.
Dement, M. E.
Demougeot, M. G.
Douglas, F. S.
Dudas, J., Jr.
Edwards, A. S., Jr.
Enell, J. W.
Erdman, F. H.
Farr, E. M.
Fayerweather, F. O., Jr.
Fernald, H. B., Jr.
Ferris, J. R.
Fisher, R. H.
Frost, E.
Gagg, R. F.
Good, H. J.
Greene, R. S.
Gruenberg, O. C.
Harris, M. F.
Heath, W., Jr.
Hegge, J. I., Jr.
Hill, C. St. J.
Hugenbruch, E. R.
Jacobus, W. W.
Jaqua, G. R.
Kittredge, J. M.
Lee, H. W. F.
Light, S. J.
Lommel, R. G.
Losson, W. L.
Mathes, E.
Mendelson, I.
Miller, D. J.
Miller, J. E.
Moolhuyzen, T.
Morrison, C. W.
Moyer, C. R.
Mullis, C. M.
Naughton, J. F.
Nazzaro, R. M.
Perry, M. F.
Pfeifer, H. J.
Pilgram, W., Jr.
Prince, D.
Pryor, F. L.
Ramsey, C. H.
Ramsey, J. H.
Rasmussen, C. A.
Reimers, B. F.
Rice, C. M.
Richards, W. M. S.
Royle, V. E.
Saltzer, B. H.
Sawyer, C. L.
Scheringer, E. W.
Shelley, Van C.
Shuman, E. S.
Smith, L. C., Jr.
Snider, M.
Stark, M.
Streett, J. W.
Suppes, D. F.
Swift, J. L.
Taylor, L. C.
Uehling, F. F.
Vegosen, J.
Way, D. S.
Welsh, H. W.
Werger, J. R.
Zabriskie, A. E.
Zjawin, J. C.

PAULSBORO
Philadelphia Section

Borst, M. A.
Calkins, G. B., Jr.
Hall, A. C.
Hammell, R. H.
Landis, J. P.
Olson, C. T.
Puder, W. C.
Raymond, L.
Shenton, F. G.
Sherman, W. T.
Sibson, W. A.
Staples, C. W.
Stewart, J. P.
Thompson, N.
vander Straeten, H. J.

PENNS GROVE
Philadelphia Section

Bley, R. E.
Flynn, J. V.
Micallef, J. M.
Molinari, W. H.
Myers, A. F.
Schanes, A.

PERTH AMBOY
Metropolitan Section

Bouvier, G. A.
Brusso, A. M.
Campbell, P. F.
Fishkin, J.
Grodner, A.
Hawke, C. E.
Johnson, B. M.
Kemp, L. W.
King, J. A.
Maguire, E. L.
Palm, A. E.
Prudden, O. D.
Roscher, A. M.
Spencer, F. A.
Verner, H. J.

PHILLIPSBURG
Anthracite-Lehigh Valley

Anderson, J. H.
Attinello, J. S.
Bernhard, R. L.
Brkich, A.
Brocklebank, A. P.
Csisek, J. J.
Dawson, L. J.
Ditson, J. D.
Ford, W. B., Jr.
Hornschuch, H.
Hyde, J. P.
Johnson, R. M.
Matthews, W. E.
McBride, W. J.
McConaghy, J. W.
Ostwald, R.
Rouch, E. A.
Spilman, R. B.
Stepanoff, A. J.
Vizbara, M.

PITMAN
Philadelphia Section

Keller, E.

PLAINFIELD
Plainfield Section

Aug, W. F.
Ayer, L. S.
Barlow, DeW. D., Jr.
Bragg, S. J.
Charlesworth, R. L.
Chizmarik, J. H.
Dawley, C. A.
Donnelly, L. F.
Grubelich, M. J.
Helmer, N. A.
Hall, R. E.
Ivins, C. F., Jr.
James, C. D.
Rathgeb, A.
Skalwold, R. N.
Stampfl, E. G.
Wentworth, R. A.
Wheaton, W. E.

POINT PLEASANT

Colvin, F. H.
Holbrook, D. L.

POMPTON LAKES
Metropolitan Section

Gagne, A. F., Jr.
Ward, J. A., Jr.

PORT NEWARK
Metropolitan Section

Smith, R. W.

PRINCETON
Metropolitan Section

Bush, G. F.
Condit, K. H.
Greene, A. M., Jr.
Hay, A. D.
Maxwell, B.
Miller, A. S.
Moody, L. F.
Otterbacher, E. H.
Panagos, P.
Rahm, L. F.
Shepherd, G. W., Jr.
Sorenson, A. E.
Webster, L. B.

RAHWAY
Plainfield Section

Hirschland, F. H.
Jandrisevits, P.
Lewis, I. R.
Nugey, A. L.
Wood, O. W.

RAMSEY
Metropolitan Section

Rhodes, T. J.

RED BANK
Metropolitan Section

Castellini, D. L.
Flood, H., Jr.
Getzoff, E. M.
Maller, M. A.
McClees, A. N.
Reingold, I.

RIDGEFIELD
Metropolitan Section

Bendelius, A.
Brown, P.
Timpson, W. Q.
Weiss, A. J.

RIDGEWOOD
Metropolitan Section

Erickson, A. C.
Grant, H. C., Jr.
Hawkins, G. W.
Jackson, C. A.
Macwatty, F. L.
Marlow, A. S., Jr.
Martin, G. W.
Ridge, C. K.
Root, F. J.
Schulte, M. J. L.
Wilkins, W. B.

RIVER EDGE
Metropolitan Section

Godfrey, W.

RIVER EDGE MANOR
Metropolitan Section

Hunter, J.

RIVERSIDE
Philadelphia Section

Burhoe, L. N.

RIVERTON
Philadelphia Section

Good, P. E.
Good, W. E.
Hallinan, J. C.
Hollerith, R.
Irwin, J. W.
Somervell, P. A.

ROCKAWAY
Metropolitan Section

Cannon, J. B., Jr.
Igleheart, G. P.
Wolsdorf, H. A.
Woodbury, G. P.

ROSELLE
Plainfield Section

Davis, F. L., Jr.
De Matteo, A. J.
Reinhart, C. O.
Schafer, O. B.
Unofsky, J.

ROSELLE PARK
Metropolitan Section

Dorian, R. M.

RUTHERFORD
Metropolitan Section

Binder, A. R.
Forhecz, J. A.
Hughes, H. E.
McGibbon, D. G.
O'Connor, W. J.
Parker, A. W.
Schlick, L. F.

SAYREVILLE
Metropolitan Section

Mack, T. R.
Miller, A. A.

SCOTCH PLAINS
Metropolitan Section

Hollingsworth, S.

SEWAREN
Plainfield Section

Lawrence, W. W.
Young, E. W.

SEWELL
Philadelphia Section

Montague, J. F.

SHORT HILLS
Metropolitan Section

Peterson, R.
Renwick, E. B.

SOUTH AMBOY
Metropolitan Section

Casey, J. E.

SOUTH KEARNY
Metropolitan Section

Anzelon, G. J.
Mair, K. T.
Murphy, J. W.

SOUTH ORANGE
Metropolitan Section

Harman, G. A.
Miller, S., Jr.
Stanwick, C. A.
Town, F. E.
Wentworth, H. T.

SUMMIT
Metropolitan Section

Bell, H. S., Jr.
Boye, B. L., Jr.
Boynton, E. S.
Burtenshaw, C. D.
Hodgson, A. W.
Lembeck, O. A.
Marshall, D. E.
Molé, H. E.
Nofsinger, C. W.
Pfluger, W. R., Jr.
Stanley, C. M.
Swenson, C. A.
Tuthill, E. S.
Weber, R. E., Jr.
Wintriss, G.

TEANECK
Metropolitan Section

Grosser, W. R.
Maurer, W. R.

TENAFLY
Metropolitan Section

Wilson, J. A., Jr.

TETERBORO
Metropolitan Section

Black, J. A.
de Haven, F. I., Jr.
Goldberg, B. W.
Hollister, J. F.
Howe, J. A.
Imbeninato, P. A.
MacGrath, K.
Manning, E. V.
Masino, F. D.
Miller, A. T.
Ochtman, L., Jr.
Silverman, S.
Smith, E. S.
Steiner, W. A.
Wright, W. N.
Zwack, R. T.

TOMS RIVER

Moore, W. J.
Podell, O.

TRENTON
Philadelphia Section

Aker, D. C.
Albert, R. C.
Barr, V. L.
Beaufrere, A. H.
Benedict, W. E.
Brooks, W. S.
Brosnan, W. J.
Bryan, J. L.
Cantwell, J. W.
Carter, J. C.
Clark, J. P.
Critchlow, H. T., Jr.
De Hart, H. F.
Engvall, H.
Evans, H. S., Jr.
Farrington, S. J., Jr.
Federici, F.
Felix, S. P., Jr.
Fenton, C. M.
Fisher, D. F.
Garrabrant, F. LeR., Jr.
Gartmann, H.
Gutzwiller, J. E.
Haldeman, R. R.
Hammer, A.
Haverstick, J. S.
Healy, W. J.
Hill, W. H.
Hitch, R. A.
Hoskins, C. H.
Johnson, E. G.
Johnston, W. S.
La Porta, H. E.
LaRocque, A. E.
Lewis, D.
Litzinger, L. P., Jr.
Loidl, J. M.
Losse, P. L.
Lupke, P., Jr.
McDermott, C. C.
Mink, J. A.
Morrell, M. C.
Peterson, A.
Pfeffer, J. F., Jr.
Pinnes, R. W.
Rasmussen, H. V.
Reeve, W. A.
Schenk, J. M.
Slavenak, A. J.
Slingman, T. D., Jr.
Sonn, G. P.
Stewart, J. P.
Sutherland, D. M.
Tankle, A.
Thompson, E. R.
Titherington, W. K.
Trishman, H. A.
Waller, C. R.
Watson, H. L.
White, J. E.
Yard, E. M.

UNION
Plainfield Section

Rathman, G.

UNION CITY
Metropolitan Section

Grupe, W. F.
Seltzer, H. S.

UPPER MONTCLAIR

Metropolitan Section

Graf, W., Jr.
McMullen, H. W.
Mead, C. A.
Spurling, O. C.

VENTNOR

Philadelphia Section

Dudnick, S.

VERONA

Metropolitan Section

Blaes, V. A.
Dobbins, R. N.

VINELAND

Philadelphia Section

Harker, J. S.
Lewis, A. D.
McClure, S. S.
Van de Pol, H.
Van Gorden, LeR. C.

WASHINGTON

Anthracite-Lehigh Valley

Joyce, R.
Schlink, F. J.

WEEHAWKEN

Metropolitan Section

Bergdolt, G. J., Jr.
Blickman, B. B.
Buttron, W. C.
Cady, H. R.
Henseler, W. J.
Zamore, W. W.

WENONAH

Philadelphia Section

Weber, A. M.

WEST ENGLEWOOD

Metropolitan Section

Ketchpel, P. A.

WESTFIELD

Plainfield Section

Bryden, C. W.
Cesarz, W. J., Jr.
Crosby, G. F., Jr.
Dannehower, G. L.
Fischer, A. F., Jr.
Hahn, P. R. T.
Inglis, R. N.
Koechlein, G. J.
Mitchell, D. B.
Oppenheimer, P. H.
Quick, H. P.
Sutton, E. W.
Weaver, L. H. A.

WEST MEDFORD

Harrington, R. L.

WEST NEW YORK

Metropolitan Section

Balton, H.
Finster, G. C.
Marny, R. G.
Mueller, E. J.
Steger, G. W.
Yerino, L. V.

WEST NORWOOD

Metropolitan Section

Budell, R.

WEST ORANGE

Metropolitan Section

Berggren, K. G.
Crowley, H. L.
Dolan, R. M.
Eppler, A. L.
Kaye, J. W.
Sparkes, H. P.
Steinmetz, A. M.

WEST POINT PLEASANT

Metropolitan Section

Schwab, G.

WESTWOOD

Metropolitan Section

Mellon, G. W.

WHIPPANY

Metropolitan Section

Fisher, E. C.
Hardaway, H. Z.
Sappet, C. L.
Williams, C. H.

WILDWOOD

Philadelphia Section

Zell, A. W.

WOODBINE

Philadelphia Section

Ritchie, P.

WOODBURY

Philadelphia Section

Hemsarth, J. H., III

WOOD-RIDGE

Metropolitan Section

Barkan, H.
Clutz, C. S.
Milici, F., Jr.
Varner, S. E.

WORTENDYKE

Metropolitan Section

Lawlor, J. J.

WYCKOFF

Metropolitan Section

Ennis, W. D.

NEW MEXICO

ALAMOGORDO

Barry, R. W.

ALBUQUERQUE

Atkinson, F. W.
Biddle, W. A.
DeSelm, C. H.
Farris, M. E.
Ford, A. D., Sr.
Frank, A. C.
Grace, C. T.
James, J. K.
Kendrick, R. A.
Kendrick, Miss R. F.
Lyon, C. L., Jr.
Munn, H. F.
Prohaska, R. J.
Reed, M. A., Jr.
Schneider, R. L.

CARLSBAD

Anderson, L. D.
Nutt, J. G.

CLOVIS

Frysinger, V. G.
Kelcec, G.
Walsh, B. H.

GALLUP

Noe, R. C.

GRANTS

Gunderson, C. K.

HURLEY

Steyskal, R. F.

LOS ALAMOS

Mainhardt, R.

MESILLA

Clayshulte, N., Jr.

ROSWELL

Kidwell, G. S., Jr.
Nelson, M.
Peed, H. E.

SANTA FE

Bice, R. A.
Blechar, T. J.
Cherry, L. H.
Crumb, C. B., Jr.
Doom, L. G.
Frame, R. A.
Gilbert, N. R.
Harris, R. L.
Henderson, R. W.
Hulbert, R. E.
Riede, J. R.
Toliver, L.
White, R. H.

SANTA ROSA

Barnhart, C. E.

STATE COLLEGE

Lukens, A. M.

YESO

Smith, L. A.

NEW YORK

ADAMS

Averill, E. A.

ALBANY

Schenectady Section

Barrow, C. J.
Bond, H. A.
Day, J. A.
Fisher, J. F.
Hedberg, H. F.
Koffsky, S.
McGrew, J. A.
Pfister, C. G.
Reissig, A. R.
Reyna, L. C.
Ronkanen, V. A.
Teeling, G. A.

ALFRED

Walters, J. E.
Williams, F. S.

AMAGANSETT

Metropolitan Section

Rush, C. D.

AMITYVILLE

Metropolitan Section

Doyle, E. F.
Powell, C. H.
Wilson, W. R.

AMSTERDAM

Schenectady Section

Hogg, J. V.
Kellogg, W. D.
Maine, W. C.
Ridley, K. J.

ANGOLA

Forster, H. W.

ARDSLEY-ON-HUDSON

Metropolitan Section

Eddison, W. B.

ARVERNE

Metropolitan Section

Siegel, L.

ASTORIA

Metropolitan Section

Blaskowski, H. J.
Houska, R. W.
Kurzweil, R.
Maccaronio, V. S.
Pesqueira, J. J.
Piencinski, G.
Smith, J. J.
Suda, S.
Ullrich, W.

AUBURN

Syracuse Section

Bryson, W. D.
Foord, J. L.
Infanger, A. W.
Ireland, W. F. A.
Paul, R. C.
Smith, L. F.
Sontag, H. P.

BABYLON

Metropolitan Section

Heller, E. W.
Shames, S. J.
Stern, A. I.
Woodhouse, H. W.

BALDWINSVILLE

Syracuse Section

Forssell, A. G.
Gruber, G. J.
Jeffcock, H. W.
Richardson, H. C.

BALLSTON SPA

Schenectady Section

Knickerbacker, J.

BARKER

Buffalo Section

McAdam, L. J., II

BATAVIA

Guppenberger, F. J.
Kustas, G. J.

BAY SHORE

Metropolitan Section

Gottsch, D. C.

BAYSIDE

Metropolitan Section

De Windt, A. K.
Robba, W. H.
Vincze, A.

BEACON

Metropolitan Section

Andrus, R. E.
Berger, J. W.
Binda, P. A.
Muenger, J. R.
Prendergast, W. A.
Ritcheske, W. F., Jr.
Townsend, A. C.

BEECHHURST

Metropolitan Section

Nelson, H. D.
Nelson, W. R.
Zink, W. R.

BELLE HARBOR

Metropolitan Section

Feinstein, L.
Karp, M. E.

BELMONT

Metropolitan Section

Riggs, H. E.

BETHPAGE

Metropolitan Section

Achilich, J. H.
Baird, E. F.
Barfus, J.
Cassidy, F. R.
Coutinho, J. DeS.
Edgerly, R. M.
Evans, B. D.
Giampiccolo, J. S.
Hoffman, C.
Houghton, R. A., Jr.
Newton, R. W.
Prigozen, H. S.

BINGHAMTON

Ithaca Section

Armstrong, H. T.
Babcock, S. E.
Baird, W. G.
Burrows, J. R.
Greacen, W., 3rd
Grover, R. E.
Holford, H. E.
Kaganowich, M. L.
Kottman, J. J.
Penton, P.
Ramage, R. W.
Redpath, H. S.
Salo, E. A.
Wells, G. S.
Willcox, F. P.

BOLIVAR

Hills, R. LaR.
Phillips, J. E.

BRIGHTWATERS

Metropolitan Section

Krummel, L. C.

BRONX

(*See New York*)

BRONXVILLE

Metropolitan Section

Bailey, E. G.
Finlay, R. B.

BROOKLYN

Metropolitan Section

Adler, G.
Adler, R.
Aidlin, S. S.
Albrecht, R. J.
Aleck, B. J.
Allan, G. W.
Alpert, J. L.
Amore, J. R.
Anderson, A. W.
Anderson, D. L.
Angus, W. J., Jr.
Aronoff, L.
Asch, A. B.
Avner, S. H.
Avram, M. H.
Baecher, B. J.
Balint, A. B.
Barbieri, J. D.
Baron, J. J.
Barrett, J. W.
Baserman, K. J.
Bauman, J. A.
Bearse, R. E.
Beatty, H. R.
Beliaeff, S. B.
Benjamin, J. P., Jr.
Bennett, G. W.
Benson, P. A., Jr.
Benson, P. C.
Berner, J.
Bertagni, A. J.
Bertelsen, C.
Bialog, E. S.
Bisen, H.
Bitko, S.
Bittner, C. E.
Bleecher, R. J.
Bock, L. S.
Boenig, R. W.
Bokair, W. J.
Bolles, S. L.
Bolton, J. A.
Bower, G. C.
Brevoort, H. W.
Brickman, A. D.
Brooks, F. T.
Bruhn, N.
Brunner, M. J.
Buchsbaum, A.
Buhler, C.
Burley, J. W., Jr.
Calabretta, P. T.
Cannizzaro, S.
Capo, J. J.
Carlson, H. C. R.
Carr, A. A.
Chadwick, N. H.
Chandler, E. F.
Chevrolet, A. J.
Chodorowski, E. G.
Christ, W. O.
Christoffersen, W. L.
Christy, H. A.
Church, E. F., Jr.
Cizek, A. W., Jr.
Cohen, H.
Collins, J., Jr.
Cook, F. L.
Cook, H. E., Jr.
Cook, H. M.
Cook, R. L.
Coppersmith, F. M.
Covino, A. O.
Curreri, R. I.
Dans, B. D.
Darcy, B. E.
D'Arcy, J. M.
Dasaro, N. A.
Dashefsky, G. J.
Davison, O. S.
Dawidowicz, W. J.
Day, R. P.
Deemer, K. C.
Dellinger, L. M.
Deutschman, M. W.
Devlin, E. J.
Dickins, F. A.
Dieffenbach, E. C.
Dinsmore, J. R.
Dobson, J.
Doll, A. W.
Doppel, L.
Dougherty, W. F., Jr.
Durland, C. M.
Egilsrud, F. S.
Elkind, M. M.
Engelson, H. R.
Erb, H. E.
Estey, C. G.
Faber, P. V.
Fangemann, W. H.
Fedde, A. M.
Ferguson, W. B.
Field, C.
Fogel, J. J.
France, A. F.
Frankel, A.
Freedenfeld, M. D.
Fried, R.
Frolander, F. C.
Fromm, C. W.
Galdieri, B.
Gall, G. F., Jr.
Gardner, W. W.
Gaynor, H. F.
Gerber, S. R.
Gershberg, J.
Gibson, F. M.
Gilbert, N.
Gilman, M. A.
Gladden, C. S.
Goldsmith, C. S.
Gollin, S.
Gottilla, S. C.
Gould, J. U.
Granet, I.
Graves, C. R.
Green, F. B. K.
Green, W. L.
Greene, H. A.
Grimm, A.
Grundberg, C. A.
Gunkel, R. E.
Gwilliam, J.
Haines, E. M.
Halbfass, A. J.
Halvorsen, R. A.
Handman, S.
Hangarter, A. J.
Hartshorn, D. S.
Hatzfeld, G., Jr.
Havemeyer, H. R.
Hazen, J.
Healy, J. M.
Hedberg, J. V.
Hefner, C. F.
Heggen, O.
Helwig, A.
Hennessey, W. L.
Henry, O. H.
Herrmann, J. F.
Hiers, G. O.
Hoff, N. J.
Holgate, F. B.
Holpert, M.
Holtorp, O. H.
Howe, T. B.
Hushen, F.
Imbembo, E. A.
Jacobs, R. M.
Jacobson, C. C.
Jaklitsch, L. J.
Jenkins, S.
Joblove, L. J.
Johnson, M. M.
Johnston, W. A.
Karp, D. S.
Katz, P.
Kennedy, W. P.
Kent, R. S.
King, G. I.
King, N. M.
Kinney, J. N.
Kishibay, C. O.
Klein, A.
Knight, G. L.
Knoop, T. M.
Kosztyla, C. J.
Kraft, R. H.
Kuhn, G. W.
Kuzman, J. A.
Laboulais, J. L.
Lambertine, J. A.
Lang, H. W.
Lanzisera, J. C.
Lasciak, C.
Laskowitz, I. B.
Lawrence, A.
Lawrence, J. H.
Lawson, E. C., Jr.
Ledgerwood, B. K.
Lenfest, B. A.
Leunis, R. R.
Leverich, J. W.
Levine, H. E.
Levy, H.
Lewin, S.
Lichtman, A. M.
Lindquist, L. L.
Lord, A. E.
Lund, S. C.
Luzzatto, G. W.
Lyon, C. S.
MacDonald, E. T.
Macklin, L.
Macomber, J. K.
Mandel, L.
Manilych, S.
Martens, R. W.
Maslow, R.
Maulbetsch, J. L.
Maxwell, R. B.
Mayer, C. W.
Mazzaglia, J. D.
McArdell, W. E.
McLean, G. D.
Mesiboff, M. J.
Meyerstein, A.
Ming, F. W.
Mistron, C. A.
Mitchell, W. J.
Moen, W. B.
Morro, J. J.
Mudarri, E. G.
Munson, S.
Murray, T. E.
Nagel, T.
Napier, A. E.
Napp, A. E.

Nederman, M. R.
Neilson, E. J.
Nelson, E. J.
Nolte, F. S.
Noveck, S. J.
Olson, H. T.
Othmer, D. F.
Paine, A. P.
Palchik, E. H.
Panitz, A.
Parker, J. C., III
Parks, J. A., Jr.
Paukner, F. J.
Peck, S.
Peluso, E.
Perotto, R. V.
Peter, W. J.
Petersen, J. H.
Peterson, G. E.
Peterson, J. A.
Petzholt, E. J.
Phalen, J. J.
Platt, W. K.
Presbrey, O.
Pulito, D. M.
Purvis, E. D.
Quier, K. E.
Rasmussen, F.
Regazzi, J. J., Jr.
Reichelt, C. V.
Reissner, H. J.
Rennie, J. A.
Richardson, M. B.
Richmond, J.
Richmond, R. L.
Riconda, L. J.
Robertson, N. F.
Robertson, R. A.
Rocklein, G. W., Jr.
Roland, P. W.
Romanelli, O. C.
Rosen, S.
Rothman, N. M.
Rousku, A. W.
Rowley, R. D.
Rupp, M. E.
Ryan, W. R.
Salisbury, R. W.
Samoiloff, L. A.
Sausele, G. J. H.
Savacchio, A. N.
Scanlon, J. J.
Schapiro, P.
Schlafke, A. P., Jr.
Schneider, B. R.
Schoenfeldt, W.
Schor, R.
Schuettinger, J. G.
Schultz, R. H.
Segall, D.
Seid, R. B.
Setchell, J. E.
Sether, J. A.
Severs, B. C.
Shea, J. A.
Sherman, W. L.
Shoor, S.
Shore, J. O.
Shuckhart, J. B.
Siegel, H. J.
Signoret, A. J.
Simington, R. E.
Skaredoff, N. N.
Smith, D. A.
Solov, A.
Sperling, E.
Spoerer, E. J., Jr.
Sprague, R. H.
Staples, F. C.
Stehman, F. F.
Stehman, F. G.
Stelz, F. F.
Stengel, C. W., Jr.
Strahl, O. R.
Sullivan, S. J.
Swertloff, S.
Thomas, A. E.
Townsend, W. R.
Traver, A. E.
Troy, M.
Tucker, R. G.
Underberg, S.
Varga, G. M.
Venezia, L. A.
Vojir, J. E.
von Bremen, D. W., Jr.
Waldron, J. L.
Wardle, T. F.
Wehrmann, W.
Weiser, S.
Weiss, L. T.
Welinder, R. A.
Wells, W. D.
Wells, W. F.
Welsh, W. H.
Wetter, P. T.
Whaley, F. H.
Whitley, F. N.
Wilson, L. O.
Wilson, W. S.
Wippermann, E. H., Jr.
Wockenfuss, W.
Wolf, A. M.
Wood, H. O., III
Wosnitzer, J.
Wunsch, J. W.
Young, J. W.
Zieve, W. A.
Zirin, M. J.
Zober, W.

BUFFALO

Buffalo Section

Albrecht, E. G.
Angstadt, J. W.
Armstrong, W. M.
Babington, N. G.
Bassett, G. B.
Batson, J. E. D.
Baxter, A. H.
Baxter, E. D.
Beane, J. A.
Beck, G. D.
Beir, R. M.
Best, R. D.
Bierbaum, C. H.
Bloss, W. S.
Booth, C. A.
Bos, P. H.
Bridgman, R. R.
Brown, M. N.
Brownell, C. E.
Bryant, R. E.
Buerk, B. C.
Burgess, D.
Burnham, L. F.
Cannon, J. P.
Carlzen, C. F.
Carr, H. R.
Chandler, R.
Charlamb, A.
Cole, W. N.
Crewson, G. G.
Dean, R. W.
Deverall, C. N.
DiAddario, A. N.
Dibble, H. L.
Dunnington, F. A.
Duppstadt, J. R.
Dyer, W. E. S.
Eckstrom, A. W.
Elliot, W. R.
Evarts, H. M.
Finnegan, T. J.
Fishkin, C. I.
Furnas, C. C.
Gambert, G.
Gayman, W. M.
Georger, R. S.
Gibson, N. R.
Godfrey, W. G.
Grace, W. A.
Gradisar, A. A.
Haglund, G. O.
Hall, H.
Hamilton, M. B.
Harding, L. A.
Harper, K. W.
Helfter, F. S.
Heuser, F.
Hughes, B. S.
Hughes, J. S.
Johnson, F. R.
Johnson, S. O.
Kenefick, J. C.
Kerker, H. F.
Kermer, M. J.
Koontz, G. M.
Laroche, J. J.
Lehn, H. C.
Lew, J.
Lilienthal, F. C.
Lind, J. A.
Linnenbruegge, H. H. G.
Lloyd, R. S.
Love, R. O.
Lyon, L. E., Jr.
Madison, R. D.
Manger, P. A.
Mensonides, S.
Mitchell, W. H.
Mittman, C. F.
Mohn, P. E.
Moore, R. P.
Munschauer, F. E., Jr.
Ohart, T. C.
Page, S. C.
Parker, K.
Pawelczyk, J. A.
Phelps, A. S.
Price, G. L.
Pritchard, W. B.
Rapp, P. C.
Raymond, A. A.
Redington, R. W.
Rees, T. W.
Reimann, H. C.
Reiss, A. E.
Ross, C. A.
Saharoff, A. V.
Schmid, B. J.
Schwartz, A. A.
Schwarz, E. A.
Sexton, G. S.
Simonson, J. W.
Simpson, J. W.
Smith, H. G.
Smith, H. L.
Snyder, N. S.
Sowers, D. W.
Sprague, W. P.
Strauss, L.
Strowger, E. B.
Taylor, G. R.
Teree, B. R.
Tessmer, R. G., Sr.
Tinker, T.
Traudt, W. F.
Trebes, H. H.
Veck, M. F.
Voisinet, W. E.
Wade, E. A.
Wagner, S. S.
Walter, M. A.
Wendel, D. P.
Wendt, E. F.
Wentz, H. H.
Whelchel, C. C.
Whiting, H. W.
Winter, O. W.
Yates, J. L.

CANAJOHARIE

Martin, W. T.
Robinson, P. C.

CANASTOTA

Syracuse Section

Dew, D. H.

CATSKILL

Kelley, F. W., Jr.

CEDARHURST

Metropolitan Section

Carr, L.

CENTRAL VALLEY

Metropolitan Section

Bullard, J. E.

CHAPPAQUA

Metropolitan Section

DeBlois, L. A.

CHERRY VALLEY

Cox, A. B.

CLARENCE

Buffalo Section

Pelletier, E. J.

CLARENCE CENTER

Buffalo Section

Fogelsonger, R. B.

COHOES

Schenectady Section

Allen, D. P.
Wondisford, F. E.

COLLEGE POINT

Metropolitan Section

Friedman, M. H.
Gordon, S.
Kussman, W.
Laun, F. C.
Sharko, S.

CORNING

Ithaca Section

Aptulsky, W.
Blizard, J. R.
Crawford, C. A.
Dahlman, F. A.
Fairchild, W. L.
Fairman, S. W.
Gray, W. T.
Hinkley, R. A.
Kriger, R. S.
Littleton, N. G.
Oakley, W. W.
Owen, J. M.
Pond, L. N.
Ruck, T. F.
Schweighofer, H. M.
Schweizer, P. E.
Stroud, W. E.
Stutske, W. A.
Sutherland, E. K.
Ticknor, W. A.
Vaksdal, A.
Wellech, E. H.
Wickersham, N. R.

CORTLAND

Syracuse Section

Illmer, L.

CRESTWOOD

Metropolitan Section

Lindenmeyer, R. E.
Lyon, J. A.

DANSVILLE

Daniels, R. L.
Novak, H. M.

DELMAR

Schnectady Section

Japp, A. L.

DEPEW

Buffalo Section

Rowland, R. H.

DOUGLASTON

Metropolitan Section

Stork, A. H.

EAST AURORA

Buffalo Section

Lowell, C. McK.

EAST FARMINGDALE

Metropolitan Section

Penny, W. F.

EAST HAMPTON

Metropolitan Section

Dickerman, W. C., Jr.

EAST ROCHESTER

Rochester Section

McChesney, I. G.

EAST WILLISTON

Metropolitan Section

Bushfield, F. T.

ELLENVILLE

Weinstein, I.

ELMHURST

Metropolitan Section

Hori, T.
Jones, W. A.
Michel, J. N.
Mittelberger, F.
Russ, D. C., Jr.
Stern, A.
Van Overveen, J. P.
Weber, P. F.
Zigas, A. L.

ELMIRA

Ithaca Section

Evans, L. R.
Jones, L. W.
Kennedy, J. C.
Kennedy, M. E.
Kinsman, R. E.
Lindsay, C. H.
Ryan, J. W.
Sokol, S. T.
Whitney, M. P.

ELM PARK

Metropolitan Section

Brown, F.

ELMSFORD

Metropolitan Section

Densen, D. A.

ENDICOTT

Ithaca Section

Anderson, C. O.
Barber, E. A., Jr.
Corbett, L. B.
Daly, G. F.
Hendrich, H. A.
Mazzola, J. A.
Weidenhammer, J. A.

FALCONER

Turner, R. H.

FARMINGDALE

Metropolitan Section

Andreini, J. I.
Barth, R. C.
Carbone, J. E.
Carleton, W. P.
Clark, A. J., Jr.
Davidoff, M.
Felberg, L.
Hauser, G. H.
Hube, A. B.
Inglee, C. F.
Jackson, R. C.
Mann, F. L., Jr.
Martin, M.
Otto, H. C. L.
Petrovsky, A.
Planiol, A. P. E.
Scheinman, B.
Stern, J. H.
Tarantino, A. F.
Theologitis, J. S.
Van Valkenburgh, H.
Wellenkamp, P.
Wittman, L.

FAR ROCKAWAY

Metropolitan Section

Duffy, E. C.
Howland, L. A.

FLORAL PARK

Metropolitan Section

Benes, R. J., Jr.
Gardner, J. A.

FLUSHING

Metropolitan Section

Aldrich, H. M.
Atkins, D. F.
Baumeister, P. A.
Becker, J.
Burdick, T. A.
Coleman, F. A.
Curlee, C. J.
Gallow, J. J.
Harris, H. I.
Hoch, A. W.
Houglund, N. O.
Kending, E. K.
Pansky, E. J.
Takesuye, T. T.
Thomson, S. G.
Wallach, B. H.

FOREST HILLS

Metropolitan Section

Baer, S.
Diringer, O.
Lione, L. V.
Lucy, S. G.
Marmorek, E. F.
Nazzaro, A. L.
Olvany, W. J., Jr.
Roller, C. A.
Schneider, F.
Stengren, J. S.
Williamson, J. L.

FT. EDWARD

Williamson, R. C.

FT. NIAGARA

Buffalo Section

Heston, R. B.

FREEPORT

Metropolitan Section

Abbott, D. T.
Holbreich, M.
Thompson, W. C.

FULTON

Syracuse Section

Couture, J. W.
Ennis, R. L.
Haskell, J. D.

GARDEN CITY

Metropolitan Section

Antony, C., Jr.
Cook, W. P., III
Cope, A. P.
Di Prima, A.
Knoll, H.
LaGambina, J. C.
Magee, G. H.
Petroman, O. M.
Sammis, E. A.
Schmidtchen, R. P.
Urquhart, N.
Walker, R. K.
Ziegler, J. W.

GARDENVILLE

Buffalo Section

Nelson, G. W.

GLEN COVE

Metropolitan Section

Bernstein, R. H.
Clegg, D.
Gould, G. D.
Nilsen, L.
Olson, F. S.
Reyling, T. R.

GLENDALE

Metropolitan Section

Steffan, C. H.

GLENHAM

Metropolitan Section

Furman, G. R.

GLENS FALLS

Schenectady Section

Buckley, R. W.
Hoopes, M.
Jamison, G. S.
Starbuck, R. A.

GLENWOOD LANDING

Metropolitan Section

Exley, L. M.

GREAT BEND

Oliver, H. G., Jr.

GREAT NECK

Metropolitan Section

Ashkinazy, S. B.
Buccola, C. H.
Dimm, W.
Eder, J. P.
Hale, W.
Howard, J. F.
Hutt, A. R.
Lea, R. B.
Mengel, W. A.
Morris, W. C.
Pick, W. J.
Skene, E. M.
Speh, H. A.
Thurn, J. A.
Tietjen, W. D.
Van Auken, H. C.
Vander Veer, J. H., Jr.
Wagner, H. C.
Wheeler, J. W.
Wiley, J. N.

HAMMONDSPORT

Ithaca Section

Meade, J. F.

HARRISON

Metropolitan Section

Shneider, H. M.

HARTSDALE

Metropolitan Section

Roth, H.
Struckmann, H., II

HASTINGS-ON-HUDSON

Metropolitan Section

Reynolds, T. W.
Roberts, F. K.
Tomey, J. G.
Twaddell, R. W.
Wadleigh, G. R.

HEMPSTEAD

Metropolitan Section

Erlick, P. S.
Hoffman, C.
Maier, H. J.
Seifried, P. E.
Williams, C. E.

HERKIMER

Lundstrom, C. B.

HICKSVILLE

Metropolitan Section

Hendrick, W. M.
Luer, G.

HOLLIS

Metropolitan Section

Hanauer, E. A.
Krieg, J. L.

Schuricht, O. O., Jr.
Unger, F. C.

HUDSON

Andres, C. S.
Speich, C. J.

HUDSON FALLS

Avery, T. M.
Crocker, J. W.
Marshall, A. R.

HUNTINGTON

Metropolitan Section

Bent, W. E.
Carlsson, E. R., Jr.
Taggart, J. M.

ILION

Colliander, C. T.
Haldeman, W. A.
Hayes, C.
Luster, D. R.
Quist, S. H.
Stitzer, D. J.

ITHACA

Ithaca Section

Adams, A. S.
Albert, C. D.
Barnard, W. N.
Black, P. H.
Carpenter, G. D.
Cheney, L. T.
Conta, L. D.
Cornell, W. R.
Crone, J. M.
Ellenwood, F. O.
Erdman, F. S.
Fried, J. A.
Gage, V. R.
Garrett, S. S.
Geer, R. L.
Goodier, J. N.
Hinkle, R. T.
Hollister, S. C.
Johnson, W. A.
Katz, I.
Kimball, D. S.
Lafferty, H. C.
Lang, H. A.
Loercher, H. W.
Mabie, H. H.
Mansky, C. B.
Morse, R. V.
Otto, C. R.
Perkins, H. C.
Perry, D. B.
Quinn, B. E.
Randolph, F. H.
Rogers, F. S.
Sawdon, W. M.
Schickel, N. H.
Vail, C. W.
Wright, L. T., Jr.

JACKSON HEIGHTS

Metropolitan Section

Comstock, C. W.
Daniele, E.
Frohlin, C. R.
Kaufman, D.
Maxfield, E. D.
Nelson, W. A.
Seiden, N. J.
Sherman, W. P.
Uhlhorn, W. F.
Vogt, R. S.
Walter, S. T.

JAMAICA

Metropolitan Section

Berger, P.
Braun, R.
Fraser, T. H.
Gall, A. H.
George, H. F.
Goetz, F. L., Jr.
Haynes, W. E.
Heil, A. L.
Hiersch, F. A.
Jerger, J. J.
Jezierski, W. F.
Moglia, G. A.
Nickerson, A. T.
Rodman, W. L.
Scheller, O. A.
Seitz, R. J.
Spector, B.

JAMESTOWN

Cummings, L. A.
Harrod, C. F.
Saff, L. C.

JOHNSTOWN

Schenectady Section

McAndrew, R. G., Jr.

KATONAH

Metropolitan Section

Mitcham, E. H.

KENMORE

Buffalo Section

Bergsland, C. H.
Bogden, E. P.
Maunz, H. W.
Mindnich, C. E.
Sniader, D. H.

KEW GARDENS

Metropolitan Section

Green, C. E.
Meier, C. K.
Nurnberger, G. R.
Tolman, C. P.

KINDERHOOK

Van Alstyne, J. J.

KINGS PARK

Metropolitan Section

Street, C. F.

KINGSTON

Burger, G. E.
Robinson, W.

LACKAWANNA

Buffalo Section

Eckhardt, F. S.

LAKE PLACID

Torrance, K. R.

LAKE SUCCESS

Metropolitan Section

Schubert, F. J.

LAKEWOOD

Walters, J. C.

LARCHMONT

Metropolitan Section

Coykendall, W. E., Jr.
Hardy, W. A.
Mullikin, H. F.
von Phul, W.

LAURELTON

Metropolitan Section

Terry, E. B., Jr.

LAWRENCE

Metropolitan Section

Lish, K. C.
Steinberg, H. G.

LEONARDSVILLE

Metropolitan Section

Degen, D. J.

LEXINGTON

Ruder, W. S.

LIBERTY

Metropolitan Section

Aldrich, H. E.

LITTLE FALLS

Snyder, H. W.

LITTLE NECK

Metropolitan Section

Barta, G. L., Jr.
Chiger, A.

LOCKPORT

Buffalo Section

Douglass, R. B.
Goodwin, C. L.
Kelso, J. M.
Teal, F. M.
Upson, C. A.
White, J. A.
Whitmore, W. W., III

LONG BEACH

Metropolitan Section

Pearman, E.
Young, W. S.

LONG ISLAND CITY

Metropolitan Section

Adams, W. E.
Bannerman, C. R., Jr.
Bartle, G. R.
Benda, H. R.
Berntsen, C. B.
Blair, M.
Booth, R. L.
Brady, J. V.
Burns, A.
Burns, A. E.
Campbell, L. G.
Choyce, N. B., Sr.
Churan, J. F.
Clancy, J. R.
Clark, P. J.
Clarke, W. E.
Crawford, K. Z.
Ellis, S.
Euvrard, LeR. E.
Evans, C. S.
Fiala, F. W.
Garrett, E. E., Jr.
Germer, F. W.
Gilman, F. W.
Gleeson, J. M.
Goldman, J.
Groschoff, E. H.
Haggerty, R. T.
Hamilton, G. S.
Hamilton, W. I.
Havens, L. A.
Hewitt, F. McC.
Hundley, F. G.
Igoe, B. J.
Ihasz, J. M.
Jacobson, F. F.
John, A., Jr.
Keller, F. J.
Klein, B. D.
Kocher, E. H.
Kohn, D. F.
Kulik, N.
La Fetra, C. W.
Lane, P. E.
Lawrence, J. VanH.
Leasenfeld, C. J.
Liptay, J. M.
Long, H.
MacDonald, M. J.
MacGowan, J. F.
Marshall, R. W.
Mayhew, B. A.
McDowell, R. W.
McKittrick, W. K.
Melichar, J. T.
Milford, A. M.
Mills, H. H.
Momot, W. E.
Monaco, A. P.
Murphy, R. E.
Muska, V. F.
Nordstrom, R. F.
Olsson, T. K.
Peterson, O. F.
Pinney, E. F.
Piuck, D.
Reid, W. C.
Rosenzweig, S.
Sackson, M.
Sheldon, S. S.
Smith, T. J.
Somers, J. C.
Stad, A. N.
Strader, R. H.
Thomas, T. R.
Thomson, J. B.
Van Derveer, W. W.
Wadiaeff, M.
Welling, L. H.
Werner, R. W.
Wines, H. T.
Zenaty, B.

LYNBROOK

Metropolitan Section

Guden, J. C.
Sossner, T. T.
White, H. J.

MALVERNE

Metropolitan Section

Haas, E. H.

MAMARONECK

Metropolitan Section

Loeffler, A. G.
Meyer, E. C.
Warren, A. K.
Williams, G. S.

MANHASSET

Metropolitan Section

Barbour, R.
Parthesius, H. J.
Sanderson, R. R.

MARYKNOLL

Metropolitan Section

Beckman, J. G.

MASPETH

Metropolitan Section

Ballenger, J. M.
Hopkins, A. G.
Merrill, G. H.
Miller, D.
Montagno, A. M.
Morris, D. L.
Pusnikas, K. A.
Sitarski, A. W.
Visnick, A.

MASSENA

Becker, D. W.
Brooks, M. E.
Schnacke, R. N.
Stickley, P. E.

MENANDS

Schenectady Section

Switzer, W. P.

MIDDLEPORT

Buffalo Section

Chater, J. A.
Young, R. H.

MIDDLETOWN

Metropolitan Section

Bradner, A. F.
Fellows, O. B.
Pease, I. R.

MIDDLE VILLAGE

Metropolitan Section

Janousek, J.

MILLBROOK

Metropolitan Section

Miller, R.

MINEOLA

Metropolitan Section

Baumann, G. W.
Frauenthal, H. L.
Livingston, R. T.
Parlette, H. L., Jr.
Post, A. E.
Reyling, G.
Welch, W., Jr.

MITCHEL FIELD

Metropolitan Section

Clain, C. E.

MONGAUP VALLEY

del Fungo-Giera, P. L. E.

MONROE

Metropolitan Section

Calvet, A. M.

MONTOUR FALLS

Ithaca Section

Price, L.

MORTON

Rochester Section

Skutt, H. A.

MT. VERNON

Metropolitan Section

Buckley, G. M.
Chan, W. K.
Haslach, J. P.
Hollander, E.
Peyser, L. F.
Schmelz, W. F.
Sellman, N. T.
Steck, R. C.
Wigle, R. A.

NEWARK

Rochester Section

Collart, K. S.

NEWBURGH

Metropolitan Section

Keto, J. R.
Lang, F. A.
Merritt, G. I.

Schilling, B. J.
Trabold, H. S., II

NEW HARTFORD

Orton, A. W.

NEW ROCHELLE

Metropolitan Section

Bourne, T. G.
Cohen, E. F.
Connon, J. A.
Copley, A. G.
Hunter, J. F.
Johnson, F. P.
Jones, F. S.
Neill, W. A.
Potts, M. W.
Schwartz, H. L.
Shank, M. E.
Steinmetz, H. G., Jr.
Stevens, G. E.
Sutherland, K. W.

NEW YORK

Metropolitan Section

Aarflot, M. G.
Aaron, C.
Abbey, H. G.
Abbott, J., Jr.
Abrams, A.
Ackert, G. F.
Adams, E. E.
Adams, O. L., Sr.
Adamson, A. V.
Addams, H.
Addington, H. B.
Alcock, G. W.
Aldrich, H. E.
Alexion, J.
Alico, J.
Alker, H. R.
Allardice, T. B.
Allen, A. R.
Allen, B. J.
Allen, E.
Allen, O. F.
Allen, R. K.
Allen, R. R.
Allen, S. G.
Allison, C. O.
Allman, W. N.
Allwork, R. R.
Almirall, J. A.
Ambrose, E. R.
Amrein, J.
Anastasi, N. J.
Anderson, G. A.
Anderson, G. E.
Anderson, H. M.
Anderson, J. A.
Anderson, J. N.
Anderson, N., Jr.
Anderson, P. W.
Andrew, J. D., Jr.
Andrew, W. G., Jr.
Angus, W. J.
Applebaum, S. B.
Apt, S. R.
Argersinger, J. I.
Armacost, W. H.
Arms, J. H. R.
Armstrong, E. H.
Armstrong, G. S.
Arnold, C. B.
Arnold, DeW. R.
Arnold, J. S.
Arnstein, L. A.
Aronofsky, J. S.
Arthur, P. D.
Ashley, E. E.
Askew, M. A.
Atkins, H. B.
Atkinson, H. S.
Atlas, R. A.
Auer, G.
Auld, R. A.
Aust, L. G. R.
Austin, H. R.
Autenrieth, G. C.
Avnsoe, T.
Babcock, C. L.
Babin, A.
Bachman, W. C.
Backus, R. A.
Bacon, G. W.
Bailey, E. G.
Bailey, R. O.
Baker, A. L.
Baker, D. B.
Baker, E. G.
Baker, H. A.
Baker, H. D.
Baker, R. M.
Bakhmeteff, B. A.
Ball, E. T.
Ball, H. F.
Ballantine, J. H.
Ballard, L.
Baller, P. W. G.
Bancroft, E. K.
Bangser, W.
Banner, W.
Barbieri, C.
Barker, C. H., Jr.
Barker, H.
Barker, J. W.
Barlow, E. H.
Barnes, H. H., Jr.
Barnett, S. A.
Barrett, W. F.
Barrie, J. G.
Barron, C. M.
Basci, J.
Bastian, E. L. H.
Basuino, F. M., Jr.
Bateman, G. F.
Bates, G. H.
Bates, J. C., Jr.
Bateson, H. W., Jr.
Bauer, H. J.
Baum, R. T.
Baumeister, T.
Bavetta, V. E.
Baxley, C. H.
Bean, G. M.
Beanfield, B. F.
Beauchemin, A. O.
Bechert, F. J.
Beck, L. J.
Becker, R. F.
Beckwith, C. F.
Begley, R. W.
Behringer, C. D.
Beier, P. H.
Beline, W. E.
Belinkin, J. A.
Bell, P. W.
Bell, W. R.
Bellegia, F.
Bellinger, L. D.
Beltran, E. V.
Bender, R. J.
Bendixen, A. H.
Benedetti, G. G.
Benedictus, A. R.
Benneche, C. P.
Bennett, D. P.
Bennett, H. G.
Bennett, H. M.
Bennett, L. W.
Benoit, L. W.
Benson, G. A.
Benson, J. G.
Benson, R. E.
Bentley, D. M.
Beran, C. F.
Berger, A.
Berger, G. G.
Berkley, H. W.
Bermingham, J. R.
Berolzheimer, H.
Berrian, K. J.
Berry, B. C.
Berry, F. R.
Berryman, H. O.
Bertram, E. A.
Bettman, R.
Betts, W. L.
Bice, G. W.
Bickford, H. L., Jr.
Biddle, R. S.
Bidstrup, H. A.
Biedermann, R. G.
Bilhuber, P. H.
Billhardt, F. A.
Billings, E. J.
Bischof, G. J.
Bises, G. R.
Bishop, F. G.
Black, A. R.
Black, J. T.
Blackburn, C. H.
Blackburn, J. E., Jr.
Blackmon, H. N.
Blair, S. M.
Blake, A. D.
Blake, J. H., Jr.
Blake, W. G.
Blanchard, R. K.
Bliss, C. P.
Blitz, E.
Blizard, J.
Blossom, F.
Blum, J. K.
Bochenek, A. F.
Boehm, W. H.
Boesch, J. C., Jr.
Boggs, R. W.
Bogot, A.
Bokum, W. H.
Boldi, H. A.
Bond, R. E.
Bonnett, L. B.
Booij, M. L.
Boswell, W. L.
Bourne, W. H.
Bovin, H.
Bower, J. G.
Boyd, J. T.
Boyer, E. S.
Bragg, J. H.
Brand, F. P.
Brand, H. H.
Brandes, C. H.
Braun, J. L.
Braverman, J. H.
Breckenridge, C. E.
Breen, P. J.
Breitman, M.

Brendlin, H. J.
Brenneke, H. J.
Breunich, P. E.
Brewer, A. F.
Bright, E. W.
Bright, J. R.
Brinton, W. C.
Bristol, R. W.
Broas, R. F.
Brockel, W. E.
Broderick, R. E.
Broeze, J. J.
Brohl, H. T.
Bromberger, D.
Brooks, J. G.
Brooks, L. E.
Brooks, T. C.
Brotzman, I. C.
Brown, C. B.
Brown, H. H.
Brown, H. W.
Brown, J. H.
Brown, R. S.
Browne, B.
Bruce, A. W.
Bruckner, R. E.
Bruehl, L.
Brugler, M. W.
Brummerstedt, E. F.
Brune, C. E.
Bruning, J. M.
Brunn, F. M.
Bruns, H. R.
Bryans, W. R.
Bubar, H. H.
Bubar, W. C.
Buchhagen, W. H.
Buckalter, L. A.
Buckholtz, I. E.
Buckowski, H. J.
Budden, E. LeR.
Buensod, A. C.
Buffinton, A. L.
Bullock, H. L.
Bunge, R. W.
Bunker, W. L.
Bunnell, S. H.
Burack, W. D.
Burchfield, W. F.
Burgess, C. G.
Burgess, W. E.
Burke, John J.
Burke, Joseph J.
Burnett, D. J.
Burnett, W. E.
Burnette, A. R.
Burns, T.
Burroughs, E. E.
Busch, F.
Butler, H. W.
Butler, W. E.
Butt, H.
Byrne, W. H.
Cady, C. I.
Cadzow, M.
Cahill, J. E.
Caldwell, W. E.
Callahan, J. G.
Callahan, V. T.
Callaway, C. R.
Cammer, M.
Campbell, D.
Campbell, E. C.
Campbell, E. D.
Campbell, J., Jr.
Cangialosi, J. C.
Carlson, A. R.
Carlson, H.
Carman, C. H., Jr.
Carney, J. F.
Carney, W. H.
Carpenter, H. B.
Carreau, G.
Carroll, E. J.
Carroll, J. D.
Carse, D. R.
Carswell, J. S.
Carter, C. W.
Carter, D. S.
Carter, E. B.
Carter, T. L., Jr.
Cartinhour, J. W.
Carver, F. S.
Case, W. L.
Casey, J. S.
Cassotti, M.
Cave, J. R.
Ceely, F. J.
Cella, J. T.
Chamberlain, L. H.
Chamberlin, W. T.
Chambers, N. C.
Champion, E. L.
Chanoux, T.
Chao, P. H.
Chapin, W. W.
Charlesworth, R. E.
Chasteen, J. S.
Chen, C.-T.
Cherdantzeff, P.
Chesler, I.
Chisholm, C. R.
Chisholm, J.
Chow, M. C.
Christie, W. D.
Church, A. H.
Church, B. A.
Churchill, A. W.
Churgin, L.
Clark, E. E.
Clark, J. M.
Clark, L. E.
Clark, W.
Clarke, C. M.
Clarke, W. H.
Clarke, W. J.
Clavin, C. G.
Clement, E. J.
Clement, R. W.
Clifford, J. J.
Clinedinst, W. W.
Coates, H. T.
Coates, R. E.
Cobb, W. H.
Cobleigh, H. R.
Codding, E. H.
Coes, H. V.
Cohn, I. A.
Colby, E. M.
Coldwell, E. S.
Cole, E. S.
Coles, V. L.
Colvin, C. H.
Combes, C. L.
Conhagen, A.
Connor, N. J.
Conover, F. H.
Constance, J. D.
Contant, P. M.
Conway, J. J.
Cook, G. C.
Cook, T. R.
Cook, W. P., Jr.
Cooledge, M. M.
Coonley, H.
Coonradt, A. C.
Cooper, F. S., Jr.
Cooper, H.
Cooper, L. J.
Copp, E. M.
Corbin, E. M.
Cornell, R. L., Jr.
Cornell, W. B.
Cornwell, H. V.
Corrigan, B.
Corrough, H. M.
Corwin, B.
Cory, D. C.
Cosentini, W. R.
Costigan, J. T.
Cotton, E. R.
Cotton, H. W.
Couch, D. H.
Cowgill, W. W.
Cox, J. W.
Craig, R.
Crane, H. M.
Crapon, H. D., Jr.
Crawford, C. A.
Crawford, C. H.
Creutz, C. E.
Crimp, G. B.
Criswell, W. W., Jr.
Croley, J. G.
Crosby, E. S.
Cross, B. J.
Crotty, J. J.
Crovatto, P. R.
Crowell, H. W.
Cudebec, A. B.
Cuff, H. B.
Cummings, J. D.
Cummings, O. P.
Cummings, R. F.
Cumner, M. S.
Curley, M. H.
Curren, R. L.
Curry, M.
Cushing, H. J.
Czajkowski, E. C.
Dahl, N. F.
Dalby, V. L.
Dale, R. B.
Dalton, H. H.
Dalton, T. E.
Dam, C. K.
Dame, F. E.
Damon, R. S.
Danforth, J. P.
Daniels, F. A.
Dankenbring, S. F.
Danker, F. R.
Danneman, F. C.
Danziger, M. J.
D'Arcy, A. J.
Dashiell, W. W.
Daudet, A. C.
D'Auria, A. R.
Davantzis, J. C.
Davenport, J. E.
Davey, P.
David, A. T.
David, E. V.
Davidson, J. L.
Davidson, S.
Davidson, W. F.
Davies, C.
Davies, C. E.
Davis, A. S., Jr.
Davis, C. C., Jr.
Davis, G. H.
Davis, H. R.
Davis, L. E.
Davis, R. W., Jr.
Day, R. R.
Dean, D. K.
Dean, E. N.
Dean, H. C.
Dearborn, C. B., Jr.
Decker, C. A.
de Crecy, J.
Deeds, E. A.
de Florez, L.
Degen, J. W.
de Jonge, A. E. R.
De Lear, N. L.
Dellis, P. L.
Delmonte, J.
de Lorenzi, O.
De Lorenzo, A. P.
De Marco, R. P.
Demarest, L. McG.
Denise, J. V.
Denison, G.
Deppeler, J. H.
Dettloff, A. M.
Deutsch, R. A.
Deutsch, Z. G.
Deutschman, A. D.
Develin, W. J.
DeWitt, C. O.
deZafra, C.
Dezeeuw, W. J.
Dicke, A. A.
Dickinson, G. S.
Dickinson, J. D.
Dickinson, W. N.
Dickson, R. P.
Didriksen, H.
Diehl, H., Jr.
Diepenbrock, J. B.
Dierckx, J.
Dilg, W. C.
Di Monico, G. S.
Dinger, H. C.
Dinges, R. M.
Ditmars, W. E.
Dixon, J. E.
Dmitrieff, B. A.
Dodge, C. C.
Dohn, C. W.
Dohrmann, H. C.
Doig, G. D.
Dolengo-Kozerovsky, W. P.
Dolle, A. R. C.
Dominguez, A. R.
Donald, W. J.
Donley, E. H.
Donley, R. E.
Donnelly, F. J.
Donnelly, G. E.
Donovan, E. L.
Donovan, W. J.
Doremus, G. A.
Doughty, C. J.
Doughty, W. F.
Dowding, L. F.
Dowling, D. L.
Downey, W. W.
Downs, C. R.
Doyle, J. P.
Drachman, M. J.
Drew, T. B.
Drew, W. B. F.
Driscoll, J. M.
Drutzu, S. T.
Drypolcher, W.
Du, D.
Dudko, N.
Duer, R. K.
Dugan, P. D.
Duncan, C. A.
Duncan, J. C.
Dunham, L., Jr.
Dunlap, R. H.
Dunn, G.
Dunning, H.
Dusenbury, W. H.
Dussel, A. N.
Dutcher, F. H.
Dutton, F. O.
Dwyer, J. J.
Dwyer, P. F.
Dyer, S., Jr.
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Eaton, C. L.
Ebdon, H. G.
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Morin, L. H.
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Morrissey, J. P.
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Morrow, L. C.
Morrow, R. L.
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Morton, B. B.
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Moss, D. F.
Motheral, H. H.
Mould, A. E.
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Mudge, S. W.
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Nicolai, A. L.
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Hussey, W. E.
Pavlik, W. J.

NORTH TONAWANDA

Buffalo Section

Bartram, P. R.
Bowen, P. P.
Kindl, F. H., Sr.
Mueller, H. S.
Williams, A. J., Jr.

NYACK

Metropolitan Section

Cross, B. J., Jr.
Masek, C. A. M.

OGDENSBURG

Stafford, C. E.

OLEAN

Buckingham, W. H.
Chaffee, R. A.
Fritton, W. J., II
Kirkpatrick, R. L.
MacKendrick, J. N.
Trumpler, W. E.

ONEIDA

Syracuse Section

Burton, W. McK.
Keller, M. W.
Keller, R.
Noyes, R. W.

ORCHARD PARK

Buffalo Section

Abbott, W. D.

OSSINING

Metropolitan Section

Hopf, H. A.
Packard, H. N.

OSWEGO

Syracuse Section

Demarest, R. T.
Green, B. H.
Hallock, H. F.
Lyons, H. R.
Mohn, H. LeR.
Shetland, D. V.

OZONE PARK

Metropolitan Section

Mongello, T.

PAINTED POST

Ithaca Section

Alford, W. A.
Cammen, M. M.
Carpenter, A. O.
Newcomb, W. K.

PALMYRA

Rochester Section

Hubbard, C. R.
McCray, C. R.
Thorn, F. C.

PEARL RIVER

Metropolitan Section

Blixt, G. F.
Bogdanoff, J. L.
Brosene, W. G., Jr.

PEEKSKILL

Metropolitan Section

Bunzel, E.
Das, P.
Holley, J. J.
Morgan, G. E.
Nyffeler, O. W.

PELHAM MANOR

Metropolitan Section

Wilcox, J. S.

PINE CAMP

Mather, D. W.

PLATTSBURG

Henshaw, C. N.
Sussdorff, E. L.

PLEASANTVILLE

Metropolitan Section

Carrick, G. S.
Dibble, F. B.
Hayek, A. F.
Karelitz, M. B.
Little, H.
McLaughlin, E. F.

PORT CHESTER

Metropolitan Section

Havey, E. J., Jr.
Patchen, M. S.

PORT EWEN

Bourke, F. E.
Lefren, E. K.
Schlatter, H.

PORT JERVIS

Metropolitan Section

Weill, M. K.

PORT RICHMOND

Metropolitan Section

Crapo, P. W.
Lockwood, G. B.
Seybolt, R. S.

PORT WASHINGTON

Metropolitan Section

Groth, H. F. A.
Hazard, G. C.
Kirkup, J. P.
Kirkup, R. P.
Puller, G.

POTSDAM

McHugh, E.
Ross, J. A., Jr.
Weiss, H. A.

POUGHKEEPSIE

Metropolitan Section

Andres, S. J.
Beaman, D. W., Jr.
Brill, G. M.
Carlson, H. N.
Collins, I. W., Jr.
Dexter, H. E.
Durbeck, A. C.
Evans, F. H.
Hargrave, R. W.
Horn, R. J.
Kosecoff, I. W.
Miller, T. H.
Richards, K.
Schlobach, G. F.
Slote, I.
Weddle, A. H., Jr.
Weiss, P. A. H.
Winchester, H. F.

PRINCE BAY

Metropolitan Section

Heim, W.
Talbot, J. M.
Uhler, W. P.

QUEENS VILLAGE

Metropolitan Section

Chandler, H. McC.
Muller, W. G.
Pinnolis, S.
Sawyer, G. N.

REGO PARK

Metropolitan Section

Schluderberg, D. C.

RICHMOND HILL

Metropolitan Section

Baxa, J. F.
Eiskamp, E.
Krauth, J. A., Jr.
Lucas, J. A.
Pflug, H. R.
Schnitzer, A. J.
Smith, D. I.
Sutherland, H. J.
Thomas, A. E.

ROCHESTER

Rochester Section

Ahrendsen, L. K.
Albert, D. J.
Aldridge, C. F.
Alexander, C. A.
Alman, L. C.
Ancona, J. F.
Arnold, W.
Barger, L. W.
Barrows, D. S.
Bausch, C. L.
Bausch, W. G.
Baxter, M. L.
Baybutt, J. W.
Beecher, C. Y., Jr.
Birkicht, E. R.
Bliss, D. S.
Boutros, R. D.
Boyd, K. R.
Brenner, K. W.
Brook, V.
Brown, C. H.
Brown, W. J.
Cala, C. F.
Camp, L. F., Jr.
Candee, A. H.
Carr, H. H.
Castle, K. B.
Cather, J. H.
Clark, H. K.
Cowell, W. T.
Crocker, A. S.
Darling, L. B.
Davidson, J. R.
Day, C. A.
Decker, H. A.
DeWolf, D. W.
Dungan, E. R.
Eaton, L. S.
Edwards, R. W.
Ekaten, C. E., Jr.
Englehardt, H. M.
Everett, H. J.
Falls, E. K.
Fellers, C. I.
Ferrari, F. A.
Field, R. A.
Fitzgerald, T. W., Jr.
Flint, C. K.
Foard, C. W.
Freeman, H. S.
Gleason, J. E.
Goeltz, P. H.
Goldey, J. S.
Gormel, E. M.
Greenawalt, R. F.
Greenfield, H. D.
Greiner, J. E.
Hamilton, A. S., Jr.
Hands, R. C.
Hanken, G. H.
Harding, H.
Hartman, E. C.
Hartsig, A. L., Jr.
Hill, J. L., Jr.
Hineline, B. C.
Hooker, T. F.
Howe, W. K.
Hoyle, R. J., Jr.
Hubbard, K. H.
Jamron, R.
Jenkins, F. G.
Jenkins, J. L.
Jobes, H. W.
Jones, A. I.
Kimmell, P. M.
Kittrell, J. B.
Koerner, A. E.
Kraus, C. E.
Kreuter, V. C.
Kurtz, H. F.
Lander, R. A., Jr.
Langdon, H. H.
Lawrence, G. E.
Lindsey, J. T.
Lusink, C. I.
Matthews, N. H., Jr.
McBean, D. M.
McCathron, C. B.
McGuire, E. J.
McLean, W. G.
Meisenzahl, T. W.
Miley, H. W.
Miller, F. D.
Minshall, J. R.
Mosher, H. A.
Moxon, A. W.
Neunes, R. W.
Odell, I. N.
Odenbach, R. C.
Palme, R. B.
Parkin, R. E.
Parlon, W. L.
Peragallo, J.
Phelps, S. M.
Philipp, G.
Pope, C. L.
Punnett, F. D.
Query, L. H.
Rengert, J. S.
Ritter, C. A.
Rogers, A. B.
Rorick, J. A.
Ross, C. C.
Ross, R. C.
Russell, S.
Ryan, C. M.
Schell, A. E.
Schell, W. A.
Scherer, F. R.
Schmitt, W. C.
Schuster, A. W.
Scott, H. W.
Setterholm, V. M.
Seymour, E. D.
Smith, L. H. V.
Snyder, E. V.
Snyder, J. H.
Sprague, O. V.
Stacy, S. C.
Steiner, O.
Storrier, J. M.
Sullivan, H. H.
Summerhays, L. J.
Summerhays, R. L.
Swift, L. B.
Trueheart, H. P., Jr.
Utz, J. R.
Watkins, R. A.
Welch, L. B.
Wesson, P. B.
White, J. E.
Wier, W. P., Jr.
Wilcox, P. S.
Wilde, F. G.
Wilder, C. L.
Wildhaber, E.
Willard, W. H.
Willsea, J.
Wolfe, B. J.
Wood, R. L.
Wood, W. D.

ROCKAWAY BEACH

Metropolitan Section

Cahn, R. D.
Evans, M. H.
Finkel, J. J.

ROCKVILLE CENTRE

Metropolitan Section

Fleischer, I.
Ketler, C. P.

ROME

Ballway, R. C.
Dalzell, R. C.
Schaffner, J. W.
Stadler, N. M.
Steele, M. G.

ROMULUS

Ithaca Section

Ruckman, J. H.

ROOSEVELT

Metropolitan Section

Schenk, F. E.

ROSLYN HEIGHTS

Metropolitan Section

Terrell, W. A.

RYE

Metropolitan Section

Billipp, E. H.
Duff, J. A.
MacNamara, M. J.

ST. ALBANS

Metropolitan Section

Eberhard, W. O.
Fairchild, C. O.
Gibbons, J. W.

SAMPSON

Syracuse Section

Keefer, R. A.

SCARSDALE

Metropolitan Section

Aldworth, E. H.
Butze, R. J.
Dexter, G. M.
Frankenhoff, C. A.
Hogin, P. E.
Horn, N. E.
Nichols, W. W.
Pape, P. F.
Slauson, H. W.
Starkweather, J. O.
Wilson, C. E.

SCHENECTADY

Schenectady Section

Adamson, A. P.
Albert, P. W.
Alger, P. L.
Alstadt, L. R.
Anderson, D. F.
Andrews, J. H.
Apperson, J. S.
Arney, T. R.
Atwood, H. M.
Azua, R. V., Jr.
Baker, F.
Barton, R. B.
Beard, R. D.
Bennett, A. L.
Bennett, F. S.
Bennett, K. S.
Bensen, I. B.
Berg-Johnsen, J., Jr.
Berkey, D. C.
Bigger, T. W.
Blatz, W. L.
Bliss, R.
Blowney, W. E.
Boals, R. L.
Bobbe, R. A.
Bookmyer, R. F.
Boring, M. McK
Bowen, R. H.
Boynton, E. R.
Bramhall, G. H.
Brecht, D. C.
Bredlau, A. E., Jr.
Brodsky, S. M.
Brunelle, H. E., Jr.
Buckland, B. O.
Bukaty, R. M.
Bunke, E. W. D.
Bushman, A. K.
Chandler, O. W.
Clark, A. G.
Cochran, D.
Coggeshall, C. S.
Collins, M. J.
Concordia, C.
Corbin, S. W.
Cornell, W. G.
Cowles, W. E.
Crary, S. B.
Dalton, W.
De Ferranti, M. A.
Delahay, D. S.
Diakoff, A. J.
Dichtel, R. N.
Eberle, J. W.
Elsworth, R. M.
Ernest, E. W.
Fisher, H. W.
Fisher, R. M., Jr.
Fleischmann, W. L.
Foster, R. G.
Fowler, F. R.
Fowler, J. E.
Foy, T. D.
Fusner, G. R.
Gamarekian, S. E.
Gardiner, C. M.
Gerding, J. E.
Gibbons, J. B.
Gibney, J. J., Jr.
Gillum, R. G.
Goryainoff, A. N.
Graham, J. J.
Gross, J. W., Jr.
Grumblatt, V. J.
Hackett, H. N.
Hallock, R. W.
Hardy, A. L.
Hatch, B. D.
Haughton, F. A.
Hendrickson, R. L.
Hidley, R. W.
Hilke, J. L.
Hobart, H. M.
Howard, A.
Howard, T. W.
Howell, H. M.
Howell, J. M.
Huber, H. J.
Hull, E. H.
Hull, T. N., Jr.
Hunt, H. E.
Hurley, R. B.
Hushen, T. M., Jr.
Ipsen, P. G.
Jackson, J. A.
Jackson, L. B.
Jackson, R. L.
Jameson, S. L.
Jenseth, H. C.
Johannes, E. G.
Johnson, R. H.
Johnson, T. L.
Jones, H.
Kandiko, J. C., Jr.
Keller, G. M.
Kellogg, A. P.
Kimball, E. E.
Kirschke, N. W.
Klint, R. V.
Knabe, C. F.
Knapp, E. C.
Knowlton, P. H., Jr.
Kohn, J. A.
Langdon, W. R.
Langmuir, I.
Lawrence, J. A., Jr.
Lee, E. S.
Levy, B. W.
Lewis, E. E.
Lindberg, Miss A. K.
Linder, C. H.
Linn, F. C.
Linville, T. M.
Lipetz, A. I.
Lovercheck, C. L.
Luce, G. G.
Lufkin, C. R.
MacGowan, G. F.
Marquis, D. H.
Marshall, D. Q.
Marshall, J. M.
Marston, R. G.
Martinez, H. G.
May, J. P.
McAndrew, R. G.
McClure, J. B.
McCoy, R. K.
McInerney, F. C.
McLane, W. J.
Miller, E. H.
Miller, S. N.
Mitchell, H. M.
Mohler, L. J.
Mosbacher, K. J., Jr.
Muir, R. C.
Murphy, T. H.
Nace, J. F.
Neal, S.
Neblett, R. S.
Nelson, D. B.
Nerad, A. J.
Nichols, W. M.
Nolan, J. B.
Noon, A. W.
Norris, R. H.
Nowacki, L. M.
Olson, W. H. M.
Otto, H. M.
Parent, D. F.
Parker, E. E.
Patterson, M. M.
Paul, S. B.
Perugi, A. H.
Petersen, M. E.
Poritsky, H.
Pragst, E.
Preston, F. W.
Prince, D. C.
Quentin, C. R.
Rankin, A. W.
Rede, G. R.
Reed, K. F.
Rhyne, C., Jr.
Riford, C. P.
Ringwalt, V. G., Jr.
Robb, H. W.
Roberts, H. E.
Roberts, J. L.
Robinson, E. L.
Rose, L. M.
Rowell, K. B.
Ruiz, A. L.
Ryan, J. E.
Salisbury, H. G.
Salisbury, J. K.
Sayre, M. F.
Schabtach, C.
Scale, W. J.
Sennstrom, H. R.
Sheldon, L. A.
Sheppard, R.
Shirrell, C. P.
Slonneger, J. L.
Smith, A. N.
Smith, A. R.
Smith, B. M.
Snell, J. K.
Spitler, T. M.
Stanton, C. H.
Stevenson, A. R., Jr.
Strang, H. E.
Suppe, C. A.
Swanson, M. C.
Syrovy, G. H.
Szmachlo, W.
Taylor, H. D.
Thearle, E. L.
Tucker, J. B.
Turner, A. D.
Vernon, R. S.
Walker, C. J.
Wallace, F. X., Jr.
Ware, D. H.
Warren, G. B.
Weber, A., Jr.
Weinberg, H. L.
White, A. O.
White, R. H.
Wieber, G. A.
Wild, A. F.
Winne, H. A.
Wood, O. L., Jr.

SENECA FALLS

Syracuse Section

Garnsey, H., Jr.
Gould, N. J.
Mann, J.
Smith, E. R.

SHORTSVILLE

Rochester Section

Preston, C. H.

SIDNEY

Faatz, E. D.
Rice, M. H.
Wright, H. T.

SOLVAY

Syrause Section

Craig, H. B.
Larsen, A. M.

SOUTHAMPTON

Metropolitan Section

Silver, M.

SOUTH FALLSBURG

Levine, H. P.

SOUTHOLD

Metropolitan Section

Cox, H. N., Jr.

SPRINGBROOK

Buffalo Section

Dollar, W. M.

SPRINGVILLE

Buffalo Section

Harrington, C. E.

STATEN ISLAND

Metropolitan Section

Fendel, F. A.
Fletcher, E. H.
George, A.
Hannan, R. Q.
Hurley, G. H.
Morse, E. P., Jr.
Puishes, A.
Rise, K.
Robinson, H. H., Jr.
Vanden Heuvel, G. R.

SUFFERN

Metropolitan Section

Wilson, G. P.

SYRACUSE

Syracuse Section

Adams, J. F., Jr.
Allen, R .F.
Aronson, C. A.
Avery, H. T.
Backity, S. M.
Barnard, N. C.
Bennum, G. C.
Briggs, J. O.
Bryans, D. R.
Bump, B. N.
Burns, R. C.
Carrier, W. H.
Chadwick, J. S.
Christensen, S. H.
Clune, J. P.
Costello, R. J.
Diefendorf, D. W.
Dietz, C. F.
Dillaway, R. B.
Failmezger, E.
Feldman, M. M.
Fletcher, J. L.
Frayer, L. W.
Freberg, C. R.
Glassey, P. P.
Gold, D. H.
Graham, Miss L.
Greenwood, H.
Hart, S. T.
Hensel, F. G.
Henwood, G. L.
Hildreth, W. O.
Hopton, W. E.
Howe, H. L.
Irons, D. E.
Johnson, E. S.
Keller, H. C.
Ketchum, S.
King, J. A.
Lang, E. H.
La Vaute, L. A.
Lazan, B. J.
Lester, E. J.
Logue, C. H.
Lyons, F. T.
Meek, G. W.
Mitchel, A. H.
Moen, L. W.
Moyer, M. B.
Murphy, E. T.
Murphy, H. W.
Murray, W. H. G.
Palmatier, E. P.
Petura, R. C.
Potter, L. E.
Pulley, F. L., Jr.
Renner, W. E.
Rhea, H. H.
Rhodes, E. K.
Robinson, A. L.
Scheiner, C. J.
Schmidt, J. F.
Schug, K. W.
Smith, E. L.
Stewart, R. W.
Stratton, J. C.
Theroux, Q. O.
Vincent, G. I.
Voos, F. W., Jr.
Walsh, G. W., Jr.
Werner, G. H.
Wilkins, V. B.
Wood, C. O.
Zimmerman, E. W.

TOMPKINSVILLE

Metropolitan Section

Junas, L. J.
Otto, H. R., Jr.

TONAWANDA

Buffalo Section

Britt, W. H.
Clemow, G. A.
Czapek, E. L.
Dickerson, K. J.
Enger, R. C.
Goga, G. F.
Kratzer, J. C.
Maienknecht, G. E.
Manney, C. J.
Parker, H. F.
Payne, W. M.
Potts, L. D.
Riede, P. M.
Steinmeyer, L. A.

TROY

Schenectady Section

Amstuz, J. O.
Bailey, N. P.
Bischoff, R.
Bordt, F. J., Jr.
Campbell, J. S., Jr.
Cluett, A. E.
Cluett, S. L.
Cook, M. A.
Crockett, C. H.
Day, C. I.
Fairfield, J. G.
Fessenden, E. A.
Franzen, C. J.
Hallum, T. E.
Harthorn, P. D.
Hill, F. C.
Houston, L. W.
Kidder, W. E.
Maloney, M. J.
Menz, C. N.
Moreland, W. J.
Newkirk, B. L.
Niemeier, B. A.
Osborne, W. C.
Palsgrove, G. K.
Parkhurst, E. R.
Riordan, H. E.
Rollins, J. P.
Rutledge, E. A.
Schubert, A. G.
Shirley, J. G.
Spence, R. S.
Stevens, H. E.
Van Dervort, A. O.
Van Wie, J. A., Jr.
Weske, J. R.
White, K. H.
Wilson, H. A.

TUCKAHOE

Metropolitan Section

Golom, J. P.

UTICA

Catlin, W. G.
Clement, W. J.
Eyres, S. A.
Fisher, B. J., Jr.
Hirsch, S. R.
Oakey, W. E., Jr.
Osuch, E. B.
Schiller, F. M.
Sherry, L. B., Jr.

VALLEY STREAM

Metropolitan Section

Kahrs, H. G.
Rossi, B. E.
San Fanandre, A. J.
Schnabel, J. W.

WALDEN

Metropolitan Section

Peck, C. V.

WANTAGH

Metropolitan Section

Dotter, R. A.

WATERTOWN

Boyer, E. D.
Chamberlain, G. L.
Dobbs, J. F.
Elsworth, J. Van V.
Gulick, H. S.
Halladay, H. F.
Hamilton, T. P.
Hess, R. G.
Kanik, R. M.
Kinne, C. E.
Laird, A. W.
Lenno, E. J.
Maxson, J. E.
Needham, P. E.
Sillcox, L. K.
Sudduth, H. N.
Tyler, W. D.
Vroman, E. C.
Whittingham, D. J.

WATERVLIET

Schenectady Section

Blake, F. E.
Burgess, W. M.
Janson, N. F.
Shoemaker, F. R.
Smith, F. B.
Tardi, J. T.

WEBSTER

Rochester Section

Foster, J. W.

WEEDSPORT

Syracuse Section

Young, J. G.

WELLSVILLE

Church, M. D.
Curry, E. C.
Field, K. A.
Gellert, T. S.
Hauselt, J. D.
Karlsson, H.
King, M. A.
MacDonald, K.
McBride, E. J.
McKee, W. R.
McPheters, L. S.
McVicker, T. E.
Mochel, M. G.
Parker, L. M.
Schaller, A.
Stewart, J. A.
Waitkus, J.

WESTBURY

Metropolitan Section

Wittig, F. E.

WESTFIELD

Buffalo Section

Vaksdal, S. R.

WEST HEMPSTEAD

Metropolitan Section

Lechthaler, C. K.
Molter, F. H.

WEST NYACK

Metropolitan Section

Fredericks, H. S.
Schimpf, A. J. D.

WEST POINT

Metropolitan Section

Fornes, G. G.
Taul, H. W.

WHITE PLAINS

Metropolitan Section

Haight, D. LeR.
Smith, E. M.
Vehslage, H. E.
Worden, E. S., Jr.

WHITESTONE

Metropolitan Section

Knowles, G. W.

WILLIAMSVILLE

Buffalo Section

Sawyer, J. G.

WILLISTON PARK

Metropolitan Section

Furedy, A. S.

WOODBURY

Metropolitan Section

Miller, E. L.

WOODSIDE

Metropolitan Section

Gussack, S. I.
Speciall, J. V.
Wright, L. K.

YAPHANK

Metropolitan Section

Avey, H. T.

YONKERS

Metropolitan Section

Arnold, P. J.
Ashcroft, A. G.
Beckwith, O. P.
Blair, J. G.
Bozenhard, C. C.
Brush, C. B.
Deutchman, J. M
Doke, G. E.
Free, A. V.
Hausel, W. M.
Henn, R. M.
Hodge, C. A.
Jenkins, P.
Keleher, J. E.
Leach, J. L.
Linscott, L. N.
Lofstedt, C. J.
Longbucco, J. R.
Midgley, F. W.
Moody, C. F.
Parsons, G. K.
Prudden, N. P.
Riggs, K.
Rogers, G. A.
Skinner, H. N.
Smolderen, F. V.
Zavodny, S.

NORTH CAROLINA

ASHEVILLE

Greenville Section

Dodge, W. W., Jr.
Osborne, B. D.
Vanderhoof, A. H.

BADIN

Piedmont-North Carolina Section

Douthit, J. H.

BILTMORE

Greenville Section

Fuller, R. B.

BREVARD

Greenville Section

Brombacher, M. H. C.
Dworetzky, L. H.

BROADWAY

Raleigh Section

Avent, J. S.

CAMP DAVIS

Houck, F. W.
Miller, R. S.

CAMP LE JEUNE

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CANTON

Greenville Section

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McBerty, D. R.

CHAPEL HILL

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West, H. I.

CHARLOTTE

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Brandt, E. H., Jr.
Brown, N. H.
Burkholder, C. I.
Crockford, R. H.
Dewey, C. A., Jr.
Evans, W. R., Jr.
Henninger, F. A.
Heyward, T. C.
Heyward, T. C., Jr.
Hosmer, A.
Jackson, F. R.
Kincaid, L. M.
LeClerc, A. B.
Leggett, I. W.
Meadors, J. W., Jr.
Nabow, D.
Oden, C. G.
Olive, R. W.
Orr, C. H.
Potter, J. T.
Steele, C. H.
Stolp, W. J., Jr.
Terrell, E. A.
Williams, E. E.
Wilson, T. W.

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Schuler, J. E.

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CONCORD

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Hardy, W. M.
Hinton, W. A.
Kenyon, V. L., Jr.
Korstian, R. J.
Lewis, R. E.
May, J. W.
Mills, F.
Reed, F. J.
Strickland, W. B.
Tew, G. W.
Theiss, E. S.
Tyren, T. T.
Wilbur, R. S.

ELIZABETH CITY

Kramer, F. K., Jr.

ENKA

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Kriek, P. P.
Moritz, A. J. L.

FT. BRAGG

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Boss, M. O.
Hauffe, F. H.
Schooley, H. E.
Semchuk, P.
Sopenoff, L. P.
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GASTONIA

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Stine, C. E.
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GOLDSBORO

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GREENSBORO

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Foust, J. D., Jr.
Gibbs, F. O.
Jones, A. D.
Kerchner, C. E.
Makasiar, V. V.
Powell, R. V.
Schaffert, G. A.
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Weatherly, R. M.

HENDERSONVILLE

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HIWASSEE DAM

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KANNAPOLIS

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Haswell, H. L.
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McCrary, O. F., Jr.
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Turner, M. E.

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ELLENDALE

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FARGO

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Dolve, R. M.

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Bezbatchenko, J. W.
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Burkley, J. K.
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Carlsten, S. V.
Carnegie, A.
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Deist, H. H.
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Frye, J. H.
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Klinedinst, L. M.
Lewis, R. B.
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Paca, J. P.
Phipps, A. J.
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Wyer, R.

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Lapp, R. J.

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Botts, E. A.
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Drucker, N.
Du Brul, E. F.
Dwight, H. S.
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Einstein, S.
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Ellis, G. P.
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Eubank, C. J., Jr.
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Faig, J. T.
Field, M.
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Fogarty, W. B.
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Frank, C. F. W.
Franken, T. L.
Freeman, B. W.
Freiberg, J. L.
Freiberg, J. M.
Frey, G. J.
Geers, J. F.
Geier, F. V.
Graney, R. W.
Grimmer, E. A.
Groene, W. LeB.
Groom, J. H.
Gudmens, H. W.
Hamill, S. M., Jr.
Hardy, C. F.
Hartmann, C.
Hassman, F. A.
Haworth, H. L.
Hecker, R. F.
Heekin, D. M.
Hehemann, F. H.
Heilig, W. E.
Heimbrock, J. H.
Hess, S. E.
Hickson, W. McL.
Hilmer, O. E.
Hoblitzell, W. H., Jr.
Howorth, R. W.
Huddle, H. E.
Jergens, A. N.
Joerger, C. A.
Johnston, P. K.
Jordre, W. S.
Judd, C. M.
Junker, W. H.
Kaiser, E. A.
Kelly, T. C.
Ketcham, C. B.
Kiefer, C. J.
Kiewit, A. L.
Kinney, A. M.
Kinney, W. S.
Koehler, C. L.
Kramer, V. J.
Kraus, G.
Kraus, W. R.
Laubach, H. E.
LeBlond, R. E.
LeBlond, R. K.
Lennart, R. P.
Lockeman, G. F.
Lontz, D. M.
Macan, W. A., III
Macneale, N.
Maier, H. J.
Malany, H. S.
Manley, S. M.
March, J. P.
March, P. G., III
Martellotti, M. E.
Martin, E. J.
Martin, J. G.
Marx, H.
McCool, H. S.
Merchant, M. E.
Metzger, H. A.
Meyers, F. H.
Mitchell, C. A.
Mitsch, E. H.
Mittendorf, W.
Morris, W. S.
Muller, E. A.
Murphy, G. W.
Muse, H. T.
Nenninger, L. F.
Nicholls, M. B.
Northrup, F. B., Jr.
Noyes, M. S.
Oster, E. A.
Paque, E. J.
Parker, A. R.
Parker, R. S.
Parsons, H. K.
Pickel, W. F.
Piecuch, A. J.
Pope, H. L.
Porter, H. T.
Pugh, A. H.
Rand, I. J.
Ransohoff, N.
Reese, E. W., Jr.
Reiser, F., Jr.
Rhame, F. P.
Rindsberg, H. D.
Ritter, H. P.
Rivoira, E. J.
Robinson, L. G.
Romaine, M.
Russell, T. F.
Sauerbrunn, E. S.
Sayre, R. L.
Schaefer, C. T.
Schaefer, F. LeR.
Schaeuble, C. H.
Schaller, P. H.
Schumann, A. P.
Schuster, E. C.
Severding, H. W.
Smith, R. L.
Somogyi, C. E.
Spiehler, C. H.
Strauchen, D. M.
Tangeman, W. W.
Taylor, J. W.
Taylor, R. M.
Tuechter, A. H.
Uihlein, H. C.
Vancil, E. D.
Verkamp, W. F.
Vinnedge, E. W.
Vockell, W. H.
Walter, J. M.
Wanamaker, G. K., Jr.
Wenner, J. J.
White, M. G.
Wilcox, E. M.
Willey, F. W.
Wong, Y. W.
Wood, C. E.
Wyman, L. A.
Yates, R. C.
Yeager, B. J.
Young, D. V.
Zlatin, N.

CIRCLEVILLE
Columbus Section

McCoard, A. P.

CLEVELAND
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Adelson, J. S.
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Alpert, S.
Amadon, F. W.
Ancell, R. M.
Apperson, J. S., III
Armstrong, G.
Arnold, G.
Austin, S. F.
Avery, J. W.
Baker, R. E.
Baker, W. C.
Balthasar, F. L.
Bareis, F.
Barnes, G. E.
Baron, B.
Bates, A. H.
Beedle, A. K.
Benes, G. P.
Benzon, G. H., Jr.
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Blackmore, W.
Blundell, E. E.
Bobrowsky, A. R.
Bodensieck, E. J.
Bohanon, H. R.
Bolz, R. E.
Bolz, R. W.
Bostic, J. A.
Brooks, C. P.
Brooks, F. W.
Brown, J. R.
Browne, K. A.
Burger, W. J.
Cagin, H.
Campbell, A. L.
Carman, E. S.
Carman, T. S.
Carmichael, C.
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Case, G. S.
Casey, A. A.
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Churchill, W. R., Jr.
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Cieslik, W. J.
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Cross, S. S.
Cummings, E. H.
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Dahl, B. F.
Danielsen, A. L.
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Dauber, C. A.
Davies, R. H.
Davis, A. F.
Davis, D. L.
Day, W. H.
Delamater, R. E.
de Lapotterie, H.
Desmon, L. G.
Dickey, P. S.
Diggs, D. R.
Doan, T. H.
Doubrava, E. N.
Dowden, E. V. D.
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Drumpelmann, C. T.
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Dueringer, W. E.
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Eisele, P. T.
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Engelman, W. H.
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Fykse, L. D.
Gabrielson, G.
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Gates, R. L.
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Githens, T. F.
Glazer, M. R.
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Gorrie, H. H.
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Green, T. A.
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Hadlow, H. R.
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Hall, J. H.
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Hardgrove, G. A., Jr.
Harper, D. R.
Hartmann, M. J.
Hauer, E. R.
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Hays, L. C.
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Heintz, R. M.
Henderson, H. H.
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Herron, J. H.
Hervey, E.
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Howe, J. L.
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Illes, S. R.
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Isidin, B. J.
Jacklitch, J. J., Jr.
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Johnson, W. A.
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Kaddeland, C. R.
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Kreiner, A. J.
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La France, R.
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Lane, R. S.
Lange, M. E.
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Lansing, J. H.
Larner, H. R.
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Lawrence, A. D.
Leisenheimer, R. H.
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Marshall, H. B.
Martindale, R. H.
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Mather, T. H.
Matlin, C.
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Matthes, M. H.
Matthews, B. H.
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Moore, W. R.
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Oberst, H. E.
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Resek, M.
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Sessions, R. C.
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Sheflin, B. W.
Shulman, M.
Sich, E. J.
Siefert, C. G.
Sipko, M.
Slaymaker, R. R.
Slobey, R. J.
Smith, R. C.
Smith, R. H.
Smith, V. J.
Sokol, T. J.
Spatz, D.
Spence, H. de L.
Sprenkle, R. E.
Stahl, J. F.
Stanbrook, R. C.
Stanitz, J. D.
Stephan, W. G.
Stephenson, A. M.
Sterbentz, W. H.
Stern, A. P.
Stewart, W. C.
Stock, A. J.
Stoeckel, A. L.
Stroud, E. G.
Stubau, C. J.
Sutherland, O. R.
Sutton, R. I.
Taylor, J. I.
Thalman, A. J.
Thomas, R. G.
Thompson, W. L.
Thurston, S.
Ticke, W.
Tobias, N.
Toma, J.
Torrence, G. P.
Townsend, C. D.
Troller, T. H.
Troth, H. C.
True, L. M.
Trumbull, A. G.
Trundle, G. T., Jr.
Tuve, G. L.
Vaisman, W.
Van Hamersveld, J. J. N.
Varady, J. C.
Vose, F. H.
Voss, W. J.
Waite, W. H.
Wallace, E. E.
Wallace, G. MacK.
Wallace, L. W.
Wallene, F. O.
Walser, R. D.
Webb, E. C.
Weeks, J. H.
Wellman, S. K.
West, H. F.
Wetherill, R., Jr.
White, H. P.
Whitehouse, I.
Whitlock, E. H.
Whitmyre, G. R.
Wickenden, W. E.
Wiitanen, W. W.
Wilcox, W. W.
Wilhelm, J. E.
Wilke, C. W.
Willard, L.
Williams, D. W.
Williams, R. L.
Wilson, H. M.
Wilson, W. M.
Wood, T. J.
Woolley, R. E.
Wright, D. K., Jr.
Wright, R. V.

CLEVELAND HEIGHTS

Cleveland Section

Cosper, W. R., Jr.
Fabel, D. C.
Graham, G. R.
Leeds, M. A.
Mercer, A. L., Jr.
Schnitman, A. R.
Straus, F. N.
Walker, H. T., Jr.
Ward, W. E.
Winship, J. T.

COLUMBIANA

Youngstown Section

Bleshelt, O.
Woodward, J. A.

COLUMBUS

Columbus Section

Allan, W. E.
Allen, C. M.
Annis, R. K.
Armstrong, E. T.
Arrigo, V. P.
Bagby, F. L., Jr.
Barre, W. E., Jr.
Beatty, R. J.
Beitler, S. R.
Bell, W. D.
Bettison, A. M.
Birk, P. M.
Blank, H. M.
Boer, E. J.
Boyd, J. E.
Brown, A. I.
Browne, W. H.
Bucher, P.
Buckley, F. D.
Carlson, H. M.
Cooney, R. L.
Cooper, C. D.
Davis, W. L.
Dayton, R. W.
Dickerson, H. S.
Engdahl, R. B.
Erickson, E. A.
Faust, H. M.
Finneran, J. E., Jr.
Frank, G. M.
Gillivan, C. Z.
Glazer, E.
Haney, G. E.
Hazard, H. R.
Herndon, L. K.
Hinckle, D. A.
Hirsch, G.
Hitchcock, E. A.
Holding, J. B.
Holt, A. D.
Jeffrey, R. H.
Johnson, Albert Pemberton
Johnson, Allen Parke, Jr.
Johnson, G. T.
Keller, E. G.
Kendrick, J. F.
Kerr, T. H.
King, W. J.
Kousnetzoff, V. P.
Kuska, M.
Kuska, N.
Landry, B. A.
Lehoczky, P. N.
Lindahl, E. J.
Marco, S. M.
Marquis, F. W.
May, J. E.
McCoy, W. W.
McCracken, W. C.
Medley, H. C.
Mercier, S. M.
Miller, R. P.
Moeller, H. A.
Moffat, G. N.
Myers, C. O.
Neil, E. B.
Norman, C. A.
Okey, P.
Paffenbarger, R. S.
Purdy, J. L.
Rader, D. A.
Reul, R. P.
Roberts, C. P.
Rumfelt, H. F. C.
Sampson, E. M.
Schumacher, E. B.
Sherman, R. A.
Simons, E. M.
Smith, C. D.
Smith, P. E.
Stang, J. H.
Stertzbach, H. W.
Stinson, K. W.
Swetting, J. R.
Tansley, L. R.
Thomas, A. H.
Thomas, T. N.
Thompson, M. B.
Tucker, R. N.
Vierck, C. J.
Walendy, M. F.
Welcker, W. A., Jr.
Widerman, M. A.
Williams, C. E.
Wyatt, DeW. H.
Wyer, S. S.
Yost, S. H.

CUYAHOGA FALLS

Akron-Canton Section

Hamlen, E. K.
Lemley, B. W.
Walker, A. G.

DAYTON

Dayton Section

Aller, W. F.
Anderson, A. S.
Anderson, W. B.
Baker, R. L.
Bardwell, A. G., Jr.
Barrows, W. I.
Baugh, E. L.
Benscoter, D. B.
Berry, E. A.
Blank, W. E.
Bridge, L. R.
Broderick, E. L.
Buvinger, G. A.
Canby, H. B.
Castellano, C.
Chasman, B.
Chryst, W. A.
Conrad, E. K.
Coors, R. M.
Curtis, A. J.
Darling, W. M., Jr.
Depue, C. A., III
Downs, W. D.
Dworzan, J. S.
Ernst, W.
Eschborn, R. J.
Feicht, E. R., Jr.
Foster, T. G.
Gano, H. McL.
Gayer, J. D.
Gebhart, H.
Gerritzen, H. K.
Gibbons, M. J.
Gismond, H. E.
Goerlitz, J. J.
Goldberg, M. M.
Gregory, W. K.
Hamilton, E. B.
Hartnell, G. F.
Hege, J. C.
Heiney, L. E.
Hunter, J. S.
Ilg, H. L.
Isely, F. D.
Johnson, E. G.
Jones, M. M.
Kemmer, P. H.
Kennard, D. C., Jr.
Keusch, R. B.
Kimmel, A. W.
Koenig, W. C.
Krase, J. M.
Krueger, J. W.
Landis, F. F.
Lau, F. L.-S.
Levine, B.
Linsker, E.
Loeffler, W. B.
Magrath, H. A.
Maiersperger, W. P.
McCormick, F. H.
McKeand, M. L.
Medici, J. J.
Mele, T. W.
Miller, W. B.
Moon, L. C.
Moore, R. C.
Newton, A. B.
Olt, R. G.
Paul, E. E.
Polk, A. F.
Polk, L. F.
Post, N.
Rathburn, J. A.
Ringo, W. R.
Robinson, J. P.
Romanowich, R. R.
Ross, W. C.
Ryan, W. W.
Schlub, C. F.
Schmudde, A. A.
Schulz, F. J.
Sealey, P. T.
Shaw, H. W.
Shimer, J. M., Jr.
Sloma, L. V.
Smith, E. D.
Smith, W. P.
Spiller, W. R.
Staley, D. C.
Stein, F. W.
Steurer, G. E.
Sussman, W.
Tate, B. E., Jr.
Terpenny, G. E.
Thieme, W. I.
Toulmin, H. A., Jr.
Townsend, J. A.
Treadwell, B.
Voak, S. P.
Von Pein, E. J.
Weber, A. R.
Wenig, H. G.
Wiggenhorn, R. H.
Wight, C.
Wilde, R. S. M., Jr.
Woll, I. E.
Wood, J. B.
Wright, O.
Wylly, A.

DEFIANCE

Toledo Section

Champlin, D. W.
Hummer, R. G.
Kinker, C. C.
Saari, T. A.

DELPHOS

Saunders, O. B.

DORSET

Freeman, A. W.

EAST CLEVELAND

Cleveland Section

Quigley, E. P.
Siegel, G. J.
Stoll, C. H.
Wright, D. K.

EAST LIVERPOOL

Youngstown Section

Wilhelm, D. M.

ELYRIA

Cleveland Section

Boeckling, G. A.
Comber, W. R.
Hannon, W. W.
Kimball, D. S., Jr.
Shipman, W. H.

ERIESIDE

Cleveland Section

Ober, G. C.

EUCLID

Cleveland Section

Chesters, F. C.
Hoffman, H. T.
Isabella, B. J.
Stuecheli, R. B.

FAIRFIELD

Dayton Section

Shirley, H. J.

FAIRPORT

Cleveland Section

Schwendler, R. H.

FAIRVIEW VILLAGE

Spencer, F. W.

FINDLAY

Flowers, H. F.

FOSTORIA

Toledo Section

Hodge, B. T.
Koester, C. J.

FRANKLIN

Dayton Section

Martindale, R. M.

FREMONT

Toledo Section

Brooks, H. W.

GERMANTOWN

Dayton Section

Zehring, R. M.

GLENDALE

Cincinnati Section

Davidson, G. E.
Miller, W. D.

GRANVILLE

Columbus Section

Powers, M. M.

HAMILTON

Cincinnati Section

Boyer, F. G. L.
Buechler, R. M.
Fahrlander, H. W., Sr.
Glaser, C. E.
Kahn, B. B.
Kutter, H. L.
Kutter, R. L.
Morain, W. A.
Randall, G. B.
Reimer, C. C.
Schlichter, F. W.
Staege, S. A.
Thomas, L. G. L.
Trowbridge, F. C.
Wood, A.
Wood, T. E.
Yingling, F. B.

HOWARD

Akron-Canton Section

Salls, D. M.

HUDSON

Cleveland Section

Wanski, M. H.

IRONTON

O'Leary, J. J.

IVORYDALE

Cincinnati Section

Hyatt, R. S.
Miller, D. E.
Montillon, G. D.
Short, R. J.
Spalding, F. W.
Welge, H. B.

KENT

Akron-Canton Section

Gebhardt, C. W.
Propper, C. H.
Saginor, S. V.
Soper, C. C.

KENTON

Columbus Section

Breidenbach, P. H.

LACARNE

Toledo Section

Bell, D. L.
Keyserling, B. H.

LAKEWOOD

Cleveland Section

Annin, J. B.
Burdg, G. W.
Hitz, R. E.
LeMay, J. E.

LANCASTER

Columbus Section

Richards, T. E., Jr.

LIMA

Abderhalden, R. B.
Deane, C. A.
Donovan, M. J.
Ellis, D. S.
Else, H. D.
Galvin, J. E.
Jones, C. E.
Mitchell, W. H.
Mulheim, J. E.
Peterson, I. L.
Schultz, O. B.
Shook, W. M.
Smith, G. W.
Snyder, H. W.
Townsend, A. J.
Weierich, A. C.
Wheeler, B.

LISBON

Youngstown Section

Huston, E. H.

LOCKLAND

Cincinnati Section

Spohr, J. C., Jr.
Woodruff, E. B.

LORAIN

Cleveland Section

Bowden, O. L.
Brown, C. F.
Evans, J. N.
Kerr, D. C.
Llewellyn, H. Z.
Wales, C. C.

LUCKEY

Toledo Section

Jenks, F.

MACEDONIA

Akron-Canton Section

Peck, E. C.

MADISON

Cleveland Section

Lovett, L. E.

MANSFIELD

Akron-Canton Section

Boyd, L. R.
Edwards, H. E.
Grant, W. W.
Wolf, C. F.

MAPLE HEIGHTS

Cleveland Section

Newton, D. L.

MARIETTA

Akron-Canton Section

Park, F. C.

MARION

Barnhart, H. J.
McNeil, M. C.

MARTINS FERRY

Pittsburgh Section

Roberts, C. H.

MASON

Dayton Section

Boynton, W. S.

MASSILLON

Akron-Canton Section

Buchanan, D. D.
McMullen, G. C.
Nelson, E. H.
Wilson, R. C.

MENTOR

Cleveland Section

Moulton, L. J.

MIDDLETOWN
Dayton Section

gronin, T.
Brown, R. W.
Foley, M. J.
Fullam, H. O.
Healy, J. J.
Kenyon, R. L.
Ladd, G. H.
Mahan, F. T., Jr.
Martin, R. W.
Sooy, W. E.
Stitt, A. B.
Suhs, G. H.

MONTPELIER
Schwartz, H. A.

MORAINE CITY
Dayton Section

Whistler, C. O., Jr.

MORROW
Dayton Section

Wetz, L. R.

MT. GILEAD
Anderson, G. P.
Semple, G. P.

MT. VERNON
Albright, A. A.
Gehres, H. A.
Lai, W.
Spencer, E. R.

NEWARK
Columbus Section

Kleist, D.
Reed, T. W.
Skubik, E. B.

NEW CARLISLE
Dayton Section

McNeill, F. R.

NILES
Youngstown Section

Schaefer, F. R.

NORTH CANTON
Akron-Canton Section

Acheson, L. K.
Beckwith, C. R.
Boger, C. E.
Daiger, G. P.
Kaufman, G. E.
Kirsch, C. W.
Mummery, C. R.
Smellie, D. G.

NORWOOD
Cincinnati Section

Langhorst, R. T.
Rubendunst, R. F.
Saveland, W. T., Jr.
Schubert, E. H.

OAKLEY
Cincinnati Section

Rollman, M. E.

ORRVILLE
Akron-Canton Section

Shriner, C. R.
Williams, R. S.

OSBORN
Dayton Section

Burns, L. G.

OTTAWA
Eckert, J. S.

PAINESVILLE
Cleveland Section

Braski, N. J.
Brumbaugh, C. O.
Butler, C. A., Jr.
Callouette, J. E.
Campbell, C. L.
Fleming, R. M.
Hobbs, J. C.
Hudson, E. L.
Moore, J. A.
Shie, C. H.

PATTERSON FIELD
Dayton Section

Sietsma, S. J.

PERRYSBURG
Toledo Section

Bell, F. S.

PHILO
Barr, H. V.
Wolpert, P. J.

PIQUA
Dayton Section

Grimes, R. V.
Ketcham, H. H.
Stacy, T. F.

POLAND
Youngstown Section

Ovesen, H.

PORTSMOUTH
Carson, G. B.

READING
Cincinnati Section

Curtis, E. H.

ROCKY RIVER
Cleveland Section

Crane, E. C.

ROSSFORD
Toledo Section

Rogers, W. D.

ST. BERNARD
Cincinnati Section

Peterson, R. C.

SALEM
Youngstown Section

Emeny, F. J.
Hunt, N. C.
Mawhinney, M. H.
Potter, R.

SANDUSKY
McWane, G. R.
Millspaugh, W. H.
Wied, J. P.

SEBRING
Youngstown Section

Chicoine, H. D.

SHADYSIDE
Pittsburgh Section

Hadden, G. L.

SHAKER HEIGHTS
Cleveland Section

Graham, J. J.
Kain, E. M.
Porter, W. E.
Roberts, A.

SHARONVILLE
Cincinnati Section

Patterson, H. R.
Pfeffer, G. E.

SHELBY
Akron-Canton Section

Dewey, R. E.

SIDNEY
Akron-Canton Section

Bickel, C. A.
Brandenburg, S. A.
Weinbrecht, J. F.
Whipp, W. E.

SOUTH EUCLID
Cleveland Section

Kline, P. A.

SPRINGFIELD
Dayton Section

Anderson, C. M.
Ball, C. W.
Bauer, C. L.
Biddle, R. W.
Biggs, G. A.
Clingerman, R. L.
Collier, R. H.
Downey, B. F.
Kraus, C. E.
Layton, J. W.
McAdams, J. E.
Ostborg, J.
Schmid, A. W.
Schmidt, R. B.
Shouvlin, P. J.
Wheeler, J. E.
Wise, H. A., Jr.

STEUBENVILLE
Pittsburgh Section

Di Cesare, F. P.
Lyngstad, A. E.

SWANTON
Toledo Section

Baker, L. R.
Langenderfer, R. C.

TIFFIN
Friedman, J. H.
Hall, R. W.

TOLEDO
Toledo Section

Alexander, C. A.
Bame, M.
Bennett, H. A.
Bennett, H. D.
Bevin, S.
Borden, J. H.
Boyd, B. E.
Carter, H. W.
Clark, W. L., Jr.
Dean, J. W.
de Coriolis, E. G.
Dicks, N. A.
Dobrow, H. P.
Dorman, N. W.
Frank, D. S.
Gillett, J.
Gravelle, P.
Graves, F. E.
Greiner, J. C.
Hallenbeck, G. E.
Hallenbeck, T. L.
Hankins, C.
Hem, H. O.
Hill, R. F.
Jobst, C.
Junker, A. J.
Kerr, H. H.
Knapp, P. R.
Kranich, H. O.
Linker, J. J.
Lufkin, G.
Marker, R. H.
McBee, E. L.
McCandless, W.
Meyer, F. J.
Middleton, L. H.
Mooney, J. D.
Moran, W. R.
Mugfor, R. J.
Mullen, J. J.
Nelden, W. A.
Oberst, D. A.
Pafenbach, C. H.
Palmer, D. M.
Peters, C. J.
Platou, L. S.
Pomeroy, C. R.
Robbins, I. P.
Rud, G. F.
Schroeder, W. C.
Schutz, H. R.
Spaulding, D. C., Jr.
Spence, C. T.
Tawresey, J. S.
Vogel, H. H.
Vogtsberger, A.
Weaver, E. W.
Williams, R. M.
Wislicenus, G. F.
Woodbury, G. F.
Yaryan, H. L., Jr.

TROY
Dayton Section

Paschall, A. L.

UNIVERSITY HEIGHTS
Cleveland Section

Shaw, A. D.

VAN WERT
Ft. Wayne Section

Tinker, O. A.

WADSWORTH
Akron-Canton Section

Doster, H. G.
Golder, J. D.
Lindsay, D.

WARREN
Youngstown Section

Brown, A.
Headley, L. M.
Krebs, E. P.
Massare, Q. P.
Melin, H. E.
Miller, P. W.
Parker, W. E., Jr.
Pealer, R. B.
Sudranski, L. L.
Wean, R. J.

WEST LAFAYETTE
Akron-Canton Section

Florck, R. S.

WILLOUGHBY
Cleveland Section

Critzer, R. D.
Fisch, R. V.
Trofimov, L. A.

WILMINGTON
Dayton Section

Murphy, J. V.

WOOSTER
Akron-Canton Section

Fabens, A. L., Jr.
Morris, T. B.

WORTHINGTON
Columbus Section

Fitz, E. M.

YELLOW SPRINGS
Dayton Section

Lawson, S. C. D.

YOUNGSTOWN
Youngstown Section

Brown, G. B.
Charignon, M. J.
Duffey, P. R.
Jantsch, E.
Jones, L.
Kerr, W. E.
Klein, A. S.
Kline, L. A.
Koran, S. E.
Lindemuth, F. L.
McCormick, C. E.
Miller, C. B.
Muller, H. E.
Pugh, G. A.
Rider, H. N.
Thurman, A. L.
Wadlock, J. P.
White, T. L.
Wick, J. L., Jr.
Wills, C. A.

ZANESVILLE
Greer, E. S.
McCoy, W. I.
Peterson, V. R.
Raymond, T. E.
Rodenbaugh, D. I.

OKLAHOMA

BARTLESVILLE
Mid-Continent Section

Blades, R. T.
Cooper, D. J.
Creel, W. H.
Dolezal, E.
Elder, H. J.
Fairley, G. G.
Fanshier, C.
Hefty, P. M.
Kleinmann, E. E.
Meade, L. P.
Ogden, P.
Schoenthaler, R.
Sears, H. T.
Sibbitt, W. L.
Simonson, H. E.
Tarr, L. L.
Thomas, R. W.
Thomas, W. M.
Walker, V. J.
Weber, L. J.

CHICKASHA
Mid-Continent Section

Thomas, J. H.

CUSHING
Mid-Continent Section

Donahue, E. B.

DUNCAN
Mid-Continent Section

Herndon, L. J., Jr.
Morrisett, O. L.
Oliver, O. N.
Owsley, W. D.
Tomlinson, J. R.

FT. SILL
Mid-Continent Section

Aufrichtig, J. I.
Gero, G.
Johnson, W. F.
Vecchiotti, F.

HOBART
Mid-Continent Section

Spotts, W. M.

KILDARE
Mid-Continent Section

Gosting, R. E.

LAWTON
Mid-Continent Section

Mallon, F. J.

McALESTER
Mid-Continent Section

Chapin, G. W.
Hall, L. W.

MUSKOGEE
Mid-Continent Section

Smith, P. M.

NORMAN
Mid-Continent Section

Carson, W. H.
Creech, M. D.
Dawson, E. F.
Felgar, J. H.
McDonald, A. A.
Myers, C. L., Jr.
Sims, E. M.

OKLAHOMA CITY
Mid-Continent Section

Blake, J. W.
Blanchard, E. E.
Bullen, C. K.
Christensen, B. A.
Cook, K. F.
Feagles, R. L.
Howard, O. N.
Huckle, M. S.
McFarland, R. E.
Rankin, R. L.
Raymond, G.
Sherman, W. S.
Sherman, W. S., Jr.
Siess, L. E.
Smith, H. E., Jr.
Stueve, W. H.
Will, D. S.
Willis, C. C.
Wood, H. V.

OKMULGEE
Mid-Continent Section

Roach, J. W.

PERRY
Mid-Continent Section

Malzahn, G. E.

PONCA CITY
Mid-Continent Section

Miller, W.
Schwenk, H. O.

SEMINOLE
Mid-Continent Section

Hurt, R. M.
McDonald, J. F.

SHAWNEE
Mid-Continent Section

Critz, R. G.

STILLWATER
Mid-Continent Section

Aldrich, B. M.
Baker, E. C.
Bandelier, G. E.
Leonard, C. M.
Thuesen, H. G.
Worthley, L. E., Jr.
Young, V. W.

TULSA
Mid-Continent Section

Aude, T. R.
Ayers, R. G.
Ballin, A. E.
Barrett, D. O.
Bauman, W. E.
Bernard, H. B.
Boyd, J. J.
Brindel, H. F.
Caldwell, J. I. MacL.
Cant, D. A.
Coleman, G. A.
Colgin, R. A.
Cook, F. V.
Daasch, F. J.
Ducker, W. L.
Fagg, L. W.
Farnham, D. W.
Fields, D. E.
Foster, D. E.
Gale, P. C.
Gibbs, L. T.
Glasgow, C. O.
Hanssen, A. J.
Heltzel, W. G.
Hoevel, K. O.
Holway, W. R.
Horne, A. N.
Hutchcraft, D. B.
Hutchcraft, D. K.
Janco, N.
Johnson, D. O.
Jones, J. D.
Keplinger, C. H.
Kershner, S. G.
Keyes, J. H.
Lane, R. K.
Lewis, O. L.
McConnell, G.
McIlhiney, J. G.
Miller, S. M.
Parker, E. H.
Pierce, H. R.
Polston, J. R.
Pope, S. H.
Pyeatt, F. E., Jr.
Ratcliffe, F. D.
Reagan, L. S.
Reed, R. Dell.
Shields, W. H.
Smith, E. E.
Spears, E. C.
Stevens, C. A.
Stewart, W. F.
Tuttle, R. B.
Wells, C. G.
Wienecke, H. A.
Williams, G. F.
Wilson, J. A.
Woolslayer, H. J.
Wyckoff, G. I.

OREGON

ASTORIA
Oregon Section

Gamon, T. H.
Yusem, H. R.

CORVALLIS
Oregon Section

Arents, C. A.
Engesser, W. F.
Graf, S. H.
Hutchinson, D.
Klein, L. M.
Martin, W. H.
Mater, M. H.
Richardson, R. LeV.
Skinner, W. J.
Siegel, L.

EUGENE
Oregon Section

Boals, R. B.
Jack, O. R.
Von Volgtlander, O.

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Oregon Section

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LA GRANDE
Oregon Section

Lottes, R. F.

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Judson, C. J.
Pearl, W. A.

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Aston, G. F.
Bates, E. N.
Caldwell, E.
Calkin, E. D.
Chaput, A. J.
Christerson, P. D.
Clayton, M. M.
Collins, G. A.
Daley, J. R.
Davis, F. L.
DeVine, D. F.
Dick, B. G.
Dickson, J.
Dunn, R. G.
Falkovich, O. C.
Foley, E. M.
Ford, A. D., Jr.
Hall, E. L.
Hays, L. T.
Hoyt, R. D.
Johannsen, J. F.
Joost, G. E.
Jordan, W. McC.
Kerzel, A.
Kierulff, J. L.
Kramer, B. L.
Matter, G. O.
McCanna, F. J.
McDougall, G. F.
McGonigle, C.
Meany, J. M.
Miller, E. B.
Mills, V. B.
Myers, M. E.
Nelson, O.
Ober, T. M.
Ofner, F. R.
Osipovich, A. A.
Othus, J. C.
Perry, T.
Purdy, F. A.
Rawson, R. H.
Robins, A.
Rowan, E. D.
Rowan, R. B.
Rutherglen, J. A.
Schaefer, D. D.
Shepherd, B. P., Jr.
Siecke, K. H.
Supove, L.
Tupling, C. G.
Ullman, J. R.
Weiser, E. P.
Whitcomb, C. F., Jr.
Whittlesey, D. W.
Whittlesey, J. W.

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Durham, J. E., Jr.
Engle, M. D.
Frick, C. H.
Fuller, F. M.
Groff, J. C.
Hamilton, J. S., Jr.
Hatfield, H. F.
Hensinger, O. E.
Jones, M. D.
Matthews, B. S.
McNeely, G. G.
Morton, W. B.
Moyer, R. E., Jr.
Reinicker, N. G.
Schmoyer, R. L.
Shelly, L. W.
Williams, D. G.
Wyatt, R. M.

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Koch, G. B.
McDonnell, W. P.
Rhoads, G. E.
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Battle, J. R.
Turner, W. E.

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Maclay, W. R.

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Livingstone, E. A.
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Wheat, O. G.

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Central Pennsylvania Section

Skawden, O. J.

BELLE VERNON
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Dietrichson, W. F.
Folmsbee, C. H.
Martis, J. M.
Udstad, S. F.

BETHLEHEM
Anthracite-Lehigh Valley Section

Andrews, A. H.
Barney, J.
Bates, A. C.
Bates, A. E.
Bates, R. E.
Bliss, J. W.
Burns, W. A.
Butterfield, T. E.
Clemens, A. W.
Coffey, E. J., Jr.
Cox, J. S.
Dudley, J. H.
Erdoss, B. K.
Fine, L.
Fleischer, T.
Gould, L. J.
Hartman, J. B.
Hill, H. O.
Hilpert, M. G.
Holme, T. T.
Jackson, T. E.
Klein, A. W.
Larkin, F. V.
Lehr, C. E.
Mengel, J. F.
Morris, M. K.
Phillips, J. L., Jr.
Randich, E. A.
Redfield, S. B.
Richardson, E. A.
Robinson, C. H.
Stampfle, R. B.
Struble, G. W.
Stuart, M. C.
Webster, J. F.
West, J. T., Jr.
Whitehead, R. C., Jr.
Willis, R. L.
Wright, C. M.

BIRDSBORO
Anthracite-Lehigh Valley Section

Japikse, B.
Laussucq, H. P. L.
Peterson, E. T.

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Ewart, H. E.

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Mullhaupt, A., Jr.
Rose, J. H.

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Oberholtzer, R. E.

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Masters, W. C.
Williams, M. F.

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Hetherington, I. J.
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Schneider, G. W.
Sinton, J. J.
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Stamm, J. D.

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Rickley, S. S.

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Kruchen, W. S.
Myers, W. K.
Richmond, J. H. M.
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Wirtsen, E.
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Lewis, H. F.
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Mitchell, J. E., Jr.
Oldham, P. T.
Snyder, G. L.

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Sargent, R. B.

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Rainesalo, C. I.
Whittlesey, F. E.

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Cox, S. F.
Downes, D. T.
Lytle, W. O.
Orr, L.
Robson, C. J.
von Kaske, K. H.

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York, M. K.

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Kerr, E.
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Smiley, C. B.

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Giacchino, G. E.
Hara, E. E.
Hill, C. H.
Johansen, H.
Kuen, W. E.
McCarty, R. A.
Midtlying, C. R.
Shull, D. E., Jr.

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Pittsburgh Section

Irey, L. D.
Llewellyn, J. E.
Mayer, W. J.

DUSHORE
Anthracite-Lehigh Valley Section

Roushey, R. E.

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Anthracite-Lehigh Valley Section

Anderson, W. E.
Casey, G. R.
Dowson, H. R.
Eaton, P. B.
Fernald, E. M.
Freytag, J. R.
Gish, J. A., Jr.
Graf, J. R.
Hammerstone, J. E.
Hulbert, W. G.
McKelvy, F. G.
McMackin, G. E.
Neff, J. W.
Raymond, W.
Reaser, W. E.
Snovel, E. R., Jr.
Snyder, W. E.
Sweeney, R. J.
Tydeman, W. A.

EAST PITTSBURGH
Pittsburgh Section

Baudry, R. A.
Brecht, W. A.
Bruzelius, E. M.
Burke, E. B.
Candee, A. H.
Carl, W. C.
Chaffey, E. K., Jr.
Criner, H. E.
Dague, A. D.
DeZubay, E. A.
Dickens, C. G.
Eagles, J. R.
Erhard, R., Jr.
Florschutz, F. E.
Fort, T.
Graham, P. S.
Griffiths, E.
Heller, P. R.
Hershey, A. E.
Herwald, S. W.
Kazik, J., Jr.
Kocsis, J.
Laffoon, C. M.
Langer, B. F.
Lessmann, G. P.
Linsenmeyer, J. Z.
Longabaugh, G. P.
Macha, E. A.
Mallick, R. W.
Manjoine, M. J.
Manuele, J., Jr.
McVetty, P. G.
Metz, G. W.
Mikina, S. J.
Monteith, A. C.
Montgomery, O. D.
Morse, W. H.
Mullan, E.
Nadai, A. L.
Peck, C. E.
Peterson, R. E.
Phillips, T. I.
Powel, C. A.
Rashevsky, M.
Robertson, B. P.
Rushing, F. C.
Schneider, W.
Thornton, F., Jr.
Tingquist, S. C.
Toolin, P. R.
Truxal, O. S.
Vollmer, P. L.
Wahl, A. M.
Warneke, M. J.
Way, S.
Welch, W. P.
Wilson, E. L.
Wommack, K. L.
Wright, R. H.

EAST STROUDSBURG

Anthracite-Lehigh Valley Section

oushell, O. C.
rake, C. E.
onaco, J. C.
aylor, E. H.

EDDYSTONE

Philadelphia Section

she, W. O.
euter, A. J.
urdick, W. E.
ooke J. W.
ievers, G. E.
riemeyer, G. F.
awain, T. H.
iordano, J.
eck, J. A.
oward, K. S.
acobson, C. A.
uchler, T. C.
un, I. B.
owy, R.
pila, F. A.
chimpf, F. J., Jr.
heehan, W. M.
teffens, J.
aylor, B. W.
ravilla, J. C., Jr.
Whitehead, C. P.
Young, H. R.

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Erie Section

Nye, R. G.

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Dunn, J. J.
Moore, F. E.
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Polick, J. W.

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Kalmbach, F.

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Freed, D. W.
McLean, M. L.

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Bach, G. W.
Baldwin, H. P.
Binder, A. F.
Blunt, R. R.
Borgel, B. F.
Bowlus, B. H.
Bradt, M.
Brinig, F. G.
Bunting, F. W.
Cain, B. S.
Campbell, B. W.
Crompton, E. E.
Curtiss, H. C.
Emmet, H. L.
Goetz, H. E.
Guy, J. M.
Hobson, R. R.
Horstkotte, E. H.
Hunter, W. L.
Ims, E. C.
Joyce, H. B.
Kaemmerling, G. H.
Kimber, H. A.
Kirkpatrick, F. M.
Kreitler, F. C., Jr.
Lindsay, G. L.
Lower, N. M.
MacKenzie, J. B.
McIntyre, R.
Metcalf, G. R., Jr.
Metzner, M. W.
Morey, A. H.
Mosher, F. D.
Mueller, H. G.
Nick, E. W.
Obermanns, H. E.
Pasick, J. M.
Perkinson, T. F.
Petersen, C. A.
Reed, M. S.
Riddle, G. W.
Roach, J. H.
Roberts, E. H.
Root, F. V.
Rouy, A. L. M. A.
Rumsey, C. S.
St. Lawrence, J.
Scheidemantle, H. S.
Scheidt, H. J.
Schlieder, H. A.
Schmock, C. J.
Schneider, F. B.
Schrader, T. O., Jr.
Selden, D.
Shenk, R. H.
Showler, A. J.
Smalenberger, E. A., Jr.
Smith, M. E.
Sprau, B. W.
Strickler, H. K.
Surdy, C. J.
Upson, R. S.
Veenschoten, V. V.
Wadsworth, J. F.
Woodward, A. J.
Zuck, M. A.

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Hutchinson, D. W.
Rodgers, O. E.

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FINLEYVILLE

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Henry, H. J.
Jackson, A. A.
Nash, R. L.

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Corl, J. G.

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GLENSIDE

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Goentner, W. B.

GREEN RIDGE

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Rodgers, E. G.

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Curran, H. M.
Dauphinais, G. A.
McManus, J. DeL.
Wong, G. D.

GREENVILLE

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Webster, H. D.

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Burkey, I. R.
Putt, J. W.
Schumo, E. M.

HANOVER

Susquehanna Section

Abendschein, E. J.

HARRISBURG

Susquehanna Section

Caskey, K. H.
Clausen, J.
Eck, A. E.
Harrison, J. L.
Kohl, E. C.
Pearson, R.
Whitaker, U. A.

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Brewer, N.
Burke, J. B.
Diament, W. A.
Fischer, K.
Hangey, T. G.
Harris, H. C.
Head, V.

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Cahill, E. H.
Hetzel, T. B.
Holmes, C. W.
Mills, F. C., Jr.

HAZELTON

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Bell, C. W., Sr.
Haentjens, O.
Schillinger, C.

HERSHEY

Susquehanna Section

Snavely, A. B.

HOLLIDAYSBURG

Central Pennsylvania Section

Kiesel, W. F., Jr.

HOMESTEAD

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HONESDALE

Anthracite-Lehigh Valley Section

Demer, L. J.

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Stokes, J. S., Jr.

INDIANTOWN GAP

Susquehanna Section

Cullen, W. J.
Johnson, D. S.

ITHAN

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Atterbury, G. R.

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Campana, J. A.
Carter, W. W.
Davis, H.
Dennison, E. S.
Dunmire, C. W.
Edris, C. J.
Evans, C. T., Jr.
Evans, P., Jr.
Flinn, W. S.
Graff, T. W.
Harms, C. F.
Hennig, F. O.
Hohenstein, F. C.
Honc, H. J.
Jameson, A. S.
Kamps, H. E.
Land, M. L.
Livingston, B. E.
Mattern, J. F.
McClure, A. W.
Moody, A. M. G.
Multhaup, R. H.
Peterson, H. S.
Piasecki, R. F.
Raye, A. H.
Riester, R. A.
Schwarz, H.
Shipley, G. B.
Smith, R. B.
Steen-Johnsen, H.
Tobin, R. K.
Vawter, W. D.
Vazsonyi, A.
Wilson, W. A.

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Statler, C. W.
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Yingling, J. E.
Zerby, R. J.

KIMBERTON

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KINGSTON

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Rogers, B. F.
Tallgren, W.

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Frank, W. E.

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DeSipin, T. J.

KITTANNING

Pittsburgh Section

Shoffner, J. M.

LANCASTER

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Brown, F. A. J.
Deegan, J. W.
De Foreest, E. T.
Dryer, E. L.
Frost, D. C.
Griffith, S. R.
Gundrum, J. H.
Handel, R. R.
Homsher, R. L.
Jackson, H. W.
Jones, A.
Kelley, J. E.
Knapp, W.
Latimer, R. A.
Lazarus, R. A.
Leng, R. B.
Long, D. R.
Louden, J. K.
McDivitt, E. T.
Mentzer, R. B.
Moen, R. H.
Noyes, W.
Patterson, A. R.
Reese, E. O., III
Robinson, H. A.
Roth, H.
Rush, J. E.
Strayer, R. K.
Von Till, L. A.
Warfel, J. R.
Weingarten, M. R.
Wickersham, J. H.

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King, A. T.
Moll, H. D.

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Harris, LeR. S.
Klintman, P. C.
Koerwer, R. J.
Koethe, W. E.
Palladino, N. J.
Powers, W. J.
Ryan, R. E.
Sabol, A.

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Anthracite-Lehigh Valley Section

Lovell, G. H., Jr.

LATROBE

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Anderson, A. A.
Gill, J. P.
McKenna, P. M.
Smith, G. P.
Weber, T. R.

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Susquehanna Section

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Tapparo, J. A.
Wilson, W. F.

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Flagle, C. D.
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Gove, N. D.
Grassi, R. N.
Hague, F. T.
Hamm, J. R.
Hart, D. A.
Harvey, C. R.
Harvey, M. E.
Hildestad, H. L.
Hoffman, D.
Knapp, C. A.
Koetting, J. L.
Markell, J., Jr.
Morgan, H. T.
Olson, P. G., Jr.
Pauly, B. H.
Ponomareff, A. I.
Redding, A. H.
Reynolds, R. L.
Sinton, W.
Spahr, J. C.
Stahl, W. F.
Stine, S. S.
Thurlo, J. A.
Van Valkenburg, J. F.
Vogel, L. B.
Walsh, E. P.
Weiler, R. E.
Wells, R. L.

LEWISBURG

Central Pennsylvania Section

Burpee, F. E.
Donehower, R. W.
Fryling, G. R.
Garman, W. DeW.
Kelly, S. J.
Kunkel, G. M.
Reed, J. C.

LEWISTOWN

Central Pennsylvania Section

Cushman, J. A., Jr.
Gooden, M. P.

LINCOLN PARK

Anthracite-Lehigh Valley Section

Wheelock, B. R., Jr.

LOCK HAVEN

Philadelphia Section

Hulsizer, R. L.
Worcester, W. S.

MANHEIM

Susquehanna Section

Horvath, G. E.

MANOA

Philadelphia Section

Pearson, W. H.

MARCUS HOOK

Philadelphia Section

Griscom, E. W.
Knoll, H. J.
MacHenry, R.

McKEESPORT

Pittsburgh Section

Cornell, H. L.
Judkins, M. F.
Weaver, E. L., Jr.

MEADVILLE

Erie Section

Bauer, E. K.
Freund, H. E.
Gilmore, C. G.
Kreamer, W. H.
Maabs, C. E.
Robinson, W. M.
Warren, E. M.

MECHANICSBURG

Susquehanna Section

Lowther, J. G.

MEDIA

Philadelphia Section

Evans, D. F.
McMahon, J. P., Jr.
Trumpler, W. E., Jr.

MELROSE PARK

Philadelphia Section

Braun, R. L.

MERION

Philadelphia Section

Frick, S. W.
Webre, A. L.
Wiggins, W. D., Jr.

MIDDLETOWN

Susquehanna Section

Herr, J. G.
Keep, H.
Locke, R. A.

MIDLAND

Pittsburgh Section

Sar Vant, W. N.

MILFORD

Anthracite-Lehigh Valley Section

Buchan, L. P.

MILLBOURNE

Philadelphia Section

Barish, N. N.

MILL HALL

Central Pennsylvania Section

Wanner, R.

MILMONT PARK

Philadelphia Section

Koenig, C. F., III

MILTON

Central Pennsylvania Section

Fisher, S. S.
Glasgow, W. H.
Hindman, W. P.
Nonemaker, F., Jr.

MINERSVILLE

Anthracite-Lehigh Valley Section

Williams, L. K.

MIQUON

Philadelphia Section

Grauel, J. L.

MONACA

Pittsburgh Section

Sellers, A. A.

MONESSEN

Pittsburgh Section

Stockus, R. L.

MORTON

Philadelphia Section

Ackerman, R. A.

MOSCOW

Anthracite-Lehigh Valley Section

Brooks, C. W.

MOYLAN
Philadelphia Section

Riddle, K. W.

MUNCY
Central Pennsylvania Section

Messer, E. W.

MUNHALL
Pittsburgh Section

Bennett, W. C. M.
McKean, R. K.
Savko, J.

NARBERTH
Philadelphia Section

Blum, W. W.
Kuylenstjerna, A. L.
LaFore, J. A.
Lambert, F. M.

NAZARETH
Anthracite-Lehigh Valley Section

Miller, J. A.
Norvig, J.
Ursprung, H.

NEW CASTLE
Pittsburgh Section

Allen, J. C.
Love, R. LeR.
Mikels, J. W.
Rowland, R. W.

NEW CUMBERLAND
Susquehanna Section

Miller, A. J., II

NEW KENSINGTON
Pittsburgh Section

Alexander, J. B.
Feil, G. W., Jr.
Howarth, E. S.
Kort, E. G.
Lathrop, E. S.
Marshall, P. W.
Meyer, L. W.
Piper, F. F., Jr.
Rearick, W. S.
Rieke, V. W.
Templin, R. L.
Wareham, J. K.
Zamboky, A. N.

NEWTOWN

Birmann, R.

NICETOWN
Philadelphia Section

Fine, B. M.

NORRISTOWN
Philadelphia Section

Adams, R. P.
Brownback, H. L.
Hannold, J. R.
Muller, G. J.
Simons, R. M.

NORTH HILLS
Philadelphia Section

Burns, A. E.

OAKMONT
Pittsburgh Section

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Dunham, B. W.
Johnson, C. E.
Manifold, G. O.
Scaife, J. V., Jr.
Weinthal, S. E.

OIL CITY
Erie Section

Daugherty, S. B.
Dreher, R. H.
Fortmann, E. H.
Gnade, E. R.
Keim, C. J.
Knapp, S. A., Jr.
Maier, A. R.
Quayle, A.

OXFORD
Philadelphia Section

Dickey, T. A.
Ware, J. H., III

PALMERTON
Anthracite-Lehigh Valley Section

Enterline, S. M.
Gotherman, C. W.
Grimes, C. E.
Hammond, S. I.
Martin, R. A.
McMackin, C. A.
Olson, F. A.
Peters, F. C.
Trewin, C. S.

PASCHALL
Philadelphia Section

Whisler, F. D.

PENARGYL
Philadelphia Section

McKenzie, D. H.

PHILADELPHIA
Philadelphia Section

(See also Lester and South Philadelphia)

Aber, J. S.
Adams, C. A.
Adler, A. C.
Agner, O. B.
Allen, H. K.
Allen, L. H., Jr.
Allen, P.
Allen, S. L.
Allwein, A. F.
Alpern, M.
Alteneder, T. G.
Althouse, E. G.
Andersen, A. T.
Andersen, E. A.
Andersen, J. R.
Anderson, R. T.
Andriola, A. D.
Andromidas, T. T.
Anthony, J. T.
Armstrong, E. R.
Arnow, S. M.
Ashby, W. B.
Astley, W. C.
Avery, J. R.
Baas, P. B. R., Jr.
Bachman, J. L.
Bacon, H. E.
Badenhausen, J. P.
Bailey, W. J.
Baillie, R. R.
Bainbridge, H.
Bainbridge, T. W.
Baird, C. A.
Baker, H. R., Jr.
Baker, J. B.
Ball, F. S.
Baltzly, C. C.
Bancroft, W.
Bangs, J. R., Jr.
Barba, C. E.
Barber, K. B.
Bark, M. E.
Barker, R. H.
Barnard, J. A.
Barnes, G. M.
Barnes, H. B.
Barnes, J. M.
Baron, M. E.
Bassett, R. M.
Batt, W. L.
Battey, W. A.
Bayles, A. L.
Beane, J. R. L., Jr.
Belcher, W. E., Jr.
Bender, E. W., Jr.
Bennett, J. S., 3rd
Bennon, M.
Benson, C. N.
Benton, L. A.
Bentson, H. J.
Berdan, F., Jr.
Bergner, F. A., 3rd
Berman, B. P.
Bernstein, H. J.
Berry, E. H., Jr.
Betz, L. D.
Billings, J. H.
Bird, L. G.
Black, E. N., 3rd
Blakeman, S. P.
Blatchley, C. G.
Blum, S.
Bodenschatz, A.
Bohjelian, A. G.
Bolton, B. F.
Bonine, C. E.
Bonner, H. H.
Bookout, E. J.
Borden, M. M.
Borton, G. W.
Bosler, L. C.
Botta, A. F.
Bourgarde, L. F.
Bowes, E. H.
Bowes, R. J.
Bowes, T. D., Jr.
Bowman, H. T.
Bowman, R. A.
Boyajian, R. D.
Boyer, E. G., Jr.
Boyer, V. S.
Boyle, J. E.
Bracegirdle, J., Jr.
Brackett, N.
Brackin, R. F.
Bradbury, D.
Bradley, J. J.
Brendlinger, W. B.
Brick, G. S.
Brill, H. A.
Brinley, C. E.
Bristol, E. S.
Brodin, E. C.
Broudo, S.
Browe, E. L.
Brown, A. K.
Brown, R. P.
Browne, A. T.
Browne, E. V.
Brunkhardt, F. W.
Brusca, L. J.
Bryans, H. B.
Bryans, H. T.
Budd, E. G.
Buerer, W. W.
Buntin, R. W.
Burckhalter, F. O.
Burmistroff, J. G.
Burton, R. C.
Bush, H. W.
Bush, R. R.
Busler, K. M.
Bye, N. C.
Cadwallader, H., Jr.
Cafiero, D. F.
Cain, J. N.
Campbell, C. B.
Campbell, D. M.
Campbell, R. J.
Carliss, O. S.
Carlson, J. R.
Casey, R. J.
Cassel, H. H.
Cassola, C. A.
Catlin, H. M.
Cavanaugh, J. P.
Cernea, R. E.
Chaplin, F. S.
Chase, P. H.
Chiavetta, V. V.
Childs, J. N.
Cirrito, A. J.
Clark, E. O.
Clark, T. F.
Clarke, C. W. E.
Clyman, H. J.
Coe, F. C.
Cole, C. S.
Coles, W. F., Jr.
Conner, J. L.
Conner, K. B.
Connors, R. T.
Cook, G., Jr.
Cooke, M. L.
Corl, H. E.
Cowell, A. T.
Cox, J. L.
Crewdson, H.
Criswell, J. C., Jr.
Crofoot, G. E.
Cross, M. A.
Crowell, Miss L.
Cummings, W. J. K.
Cushing, T. E.
Dadley, J. W.
Daley, J. A., Jr.
Daniels, E. R.
Dannemann, H. F., Jr.
Darmody, W. J.
Daugherty, E. S.
Daugherty, F.
Davidson, W. H.
Davis, F. R.
Davis, S. H.
Dawson, L. N.
Decker, H. L.
DeHuff, H.
Dell, William H.
Dellplain, M.
DeLong, A. F.
de Mauriac, W. J.
Derby, R. E.
Deutsch, J. T.
Diamond, A. R.
Digan, T. J., Jr.
Dimmick, H. S.
Disston, W. D.
Divan, L. S.
Dixon, C. F.
Dockstader, E. K.
Dodge, K.
Dodge, R. M.
Doering, J.
Dowell, D.
Doyle, E. D.
Doyle, W. J.
Dykes, J. R.
Dzurka, J. J.
Ebenbach, R.
Eckman, D. P.
Edge, M. P.
Edge, W. C.
Edwards, J. A., Jr.
Ehlers, H. E.
Eigenbrot, J. L.
Eklund, T. I.
Eksergian, R.
Elliott, O. M.
Elsey, W. R.
Eni, L. J.
Estrada, H.
Etchen, H. G.
Exler, D. C.
Fairbanks, C. M.
Falvey, J. A.
Farley, J. J.
Fassbender, W. J.
Feldman, N. R.
Fernstrom, F. S.
Filippone, F. S.
Finkel, J. B.
Fisch, A., Jr.
Fischer, F. K.
Fischer, U. W.
Fisher, G. K.
Fitzgerald, J. M.
Flickinger, J. H.
Fogg, W. R.
Foley, W. S.
Forrest, H. D.
Fraher, B. J.
Frame, J. S., Jr.
Franck, C. C.
Franks, E. H.
Fransema, J. A.
Frey, C. A.
Frohmuth, R. L.
Fry, A. G.
Fry, A. H.
Fry, H. P.
Fuchs, E. A.
Fuller, W. D.
Fulton, G.
Fulweiler, W. H.
Funk, N. E.
Gallagher, E. I.
Galloway, C. D.
Gammell, J.
Garity, W. E.
Gavit, W. P.
Geertz, A. O.
Gerdes, H. A.
Gerstmyer, R. G.
Gess, L.
Gethen, G. S.
Gill, C. A.
Gladeck, F. C.
Glasby, J. B.
Glenn, E. R.
Glinske, E. J., Jr.
Glueck, F. J.
Godfrey, R.
Goedecke, M.
Goehring, W. W.
Goff, J. A.
Goldsmith, L. M.
Goodale, F.
Goodwin, R. M.
Gordon, V. J., Jr.
Gotwals, C. S.
Graf, J. C.
Grandinetti, J. R.
Gratch, S.
Gravitz, E.
Green, L. S., Jr.
Groothuis, H.
Gross, S.
Guenzel, R. A.
Gulick, L. N.
Gysling, M. H.
Hall, A. E.
Hall, P. P.-G.
Haller, K. R.
Hamill, J. R.
Hammershaimb, G. T.
Hammond, N. B.
Hanger, S. R.
Hansen, V.
Hare, W. E.
Harlow, J. H.
Harman, W. H., Sr.
Harris, H. S.
Harris, J. E., Jr.
Harris, R. J.
Hartman, W. W., Jr.
Harwick, H. K.
Haug, J. R.
Haug, J. S.
Havens, K. B.
Hehemann, R. F.
Heilig, C. E., Jr.
Helvig, W. J.
Hemenway, S. H.
Hendrix, H. W.
Henwood, J. B.
Hepke, W. C.
Herb, J. H.
Herr, W. A.
Hess, F. O.
Hess, J. R.
Hewitt, A. R.
Hickman, C. D.
Higgins, E. M.
Hill, C. S., Jr.
Hires, J. E.
Hobbs, W. S.
Hoell, G. S.
Hoffman, W. M.
Hogg, J. W.
Holden, F. R.
Holmes, R. B.
Holton, P. H.
Hooker, H. W.
Hoopes, P. R.
Hopkins, J. R.
Hopper, T. W.
Hopping, E. LeR.
Hornberger, F. C.
Housley, T. P.
Howarth, H. A. S.
Huber, G. L.
Hunt, C. T.
Hunt, J. E.
Hunter, A. T.
Hunting, R. W.
Huttinger, W. R.
Hyde, E. M.
Irmer, C. B.
Irwin, K. McH.
Irwin, P. L.
Irwin, R. D.
Jackson, A. C.
Jacoby, N. P.
Jamison, C. C., III
Jansson, O. E.
Jensen, J. A.
Johler, R. A.
Johnson, D. C.
Johnson, R. P.
Jones, C. C.
Jones, E. L.
Joos, C. E.
Jud, J. J.
Juram, A. E.
Katz, N. J.
Keating, D. J., Jr.
Keech, G. B.
Keene, B. F.
Kellem, J. C.
Keller, H. G.
Keller, H. H.
Keller, W. McK.
Kenney, L. H.
Kent, L. R.
Kerr, S. L.
Kerrick, J. H.
Kessler, H. H.
Kinderman, W. J.
Kiproff, P.
Kisner, A. G.
Kite, H. J.
Kitt, H. G.
Kittredge, A. E.
Klumpp, J. B.
Knight, S. H.
Koch, C. H., Jr.
Kocyan, G. H., Jr.
Kop, P. E.
Koplin, R. D.
Kothny, G. L.
Kramer, K. S.
Krauss, H. E., Jr.
Kroon, R. P.
Kulick, S.
Kuljian, H. A.
Kuntny, G. W.
Kyle, J. J.
Lakey, A. B.
Lamb, D. B.
Landis, C. W.
Lang, H. J.
Lang, W. A.
Latshaw, E.
Lawrence, H. F.
Lee, R. J.
Lemmon, J. R.
Lepley, R. D.
Levin, B. S.
Levinson, H. J.
Levitt, S.
Levy, L. F.
Light, J. A.
Lilly, C. A., Jr.
Linch, E. P.
Linton, W. R.
Lippenholz, J. S.
Liversidge, H. P.
Locke, D. H.
Longenecker, C.
Lorenz, G. J.
Lorenz, J. G., Jr.
Lovekin, R. E.
Lowenstein, S.
Lucke, F. C.
Luehrs, D. M.
Lyon, P. S.
Machold, C. E.
Mackenzie, S. T.
MacNair, D. J.
MacNamee, W. R.
Major, W. S.
Mallon, C. J.
Malone, J. L.
Mangler, W. E.
Manning, T. J., Jr.
Manning, W. T.
Manz, L. C.
Markey, H. I.
Marthens, R. S.
Mattingly, R. D.
Mattis, R. J.
Maximon, L. B.
Maxwell, G. L.
McBride, T. C.
McCants, R. P.
McCarthy, J. J.
McCormick, H. B., Jr.
McDowell, C. H.
McIlvaine, J. H.
McIntyre, D. F.
McKaig, A. W.
McKendrick, L.
McMahon, T. E.
McMeekin, B. M.
McNeal, D. R.
McNickle, R. C.
Mears, E. W.
Medlar, L. A.
Meenen, R. J. A.
Melas, W.
Messaros, F. C.
Midgette, E. L.
Miller, E. F.
Miller, F. W.
Miller, K. O.
Miller, R. I.
Moffett, J. T.
Moore, C. B.
Moore, H. T.
Morgan, D. W. R.
Morrison, T. J.
Moyer, S.
Mudd, J. P.
Munro, R. W.
Murphy, W. B.
Murr, C. H.
Muschamp, G. M.
Mylting, L. E.
Napier, E. T.
Needs, S. J.
Neiva, R. V.
Nestler, D. E.
Nevells, I. L.
New, W. R.
Newton, G. H.
Newton, J. S.
Nibecker, K.
Nicholson, E. K.
Nielsen, S. G.
Noor, R. A.
Nusbaum, L.
Oberhuber, W. F.
Obrant, L. J.
O'Brien, F. L., Jr.
Odenweller, H. F.
Ogden, N.
Ogle, E. Des F.
Orr, J. L.
Packer, J. B., Jr.
Parker, J.
Parker, J. C.
Parker, R. E. B.
Parr, G. C.
Paul, R. F.
Peirce, W. H.
Peller, L.
Penrose, C.
Perry, T. D.
Peters, J. C.
Peterson, E. C.
Peterson, J. D.
Pettinos, G. F.
Pettit, A. R.
Pew, J. N., Jr.
Pfeiffer, C. G.
Pfeiffer, F. F.
Phillips, C. B.
Phillips, G. W. MacP.
Pitman, R. W.
Podolsky, J. P.
Polak, I. P.
Porro, R. M.
Powell, C. E.
Powell, W., Jr.
Pressey, R. W.
Preston, H. E.
Price, M. M.
Prior, J. A.
Prohammer, F. G.
Prostrednik, E. J.
Ptacek, F.
Pursel, H. R.
Putz, T. J.
Quaid, J. A.
Quast, W. F.
Quick, G. C., Jr.
Quick, R. S.
Quinn, A. M.
Quinn, J. J., Jr.
Rabe, J. S.
Ragsdale, E. J. W.
Ralston, A. H.
Ramsden, J. T.
Ranck, C. E.
Randall, M. C.
Rawson, A. J.
Raynes, S. H., Jr.
Read, M. H.
Reed, D. R.
Reed, H. W.
Rehfuss, W. C.

Reidl, A. L.
Reilly, C. T.
Reinhardt, A. E.
Reiniger, E. M.
Repscha, A. H.
Reynolds, H. F.
Reynolds, S. D.
Richardson, W. H.
Riehl, H. B.
Riesenberg, F. W.
Rincliffe, R. G.
Rizzo, J. F.
Robbins, L.
Robinson, H. I.
Robinson, P. A.
Robinson, W. V.
Rodenbaugh, H. N.
Rogers, F. H.
Rohlin, V. A.
Rohrer, J. H.
Rollo, W. S.
Roome, B. R., Jr.
Roseman, T.
Rosen, A.
Rosenfeld, M. S.
Rossetto, L.
Roth, P. V.
Rothenberg, G. S.
Ruck, G.
Rudbarg, F.
Rude, R. L.
Ruff, H.
Ruley, B. T.
Russ, D. G.
Ryan, F. J., Jr.
Saldin, H. B.
Samans, W.
Sampter, H. C.
Sanderson, V. L.
Sauerwald, J.
Sauter, W. V.
Savarese, J. J.
Savaro, V. G.
Schade, A., III
Schaefer, J. W.
Schaum, O. W.
Schear, A.
Schick, D. F., Jr.
Schlack, B.
Schofield, W. R.
Schroder, C. H.
Schwarz, F. W.
Schwendner, A. F.
Scott, C. R., Jr.
Scott, J. B.
Scutt, E. D.
Seaholm, K. E.
Sellers, C., III
Serrell, J. J.
Serrell, M. A.
Severs, E. B.
Shaffer, M. W.
Shaffer, W. G.
Shank, P. R.
Shawcross, J. G.
Shea, T. M.
Sheppard, W. L.
Shermet, R. M.
Sherron, J.
Shoaf, R. G., Jr.
Sibson, H. E.
Simins, A.
Simons, C. W.
Simpson, H. A.
Simpson, R. H.
Singer, M. A.
Sloan, R. A.
Sloan, W. A.
Smith, F. J.
Smith, N. L.
Snyder, H. E.
Snyder, R. P.
Sookasian, G. H.
Sorensen, K. T.
Spellman, C. B.
Spielman, J. R.
Squires, R.
Stapfer, R. D.
Stark, A. C.
Stein, I. M.
Steinfeld, M. E.
Steins, C. K.
Stem, F. B.
Stephenson, W. B.
Stettler, R. H.
Stevens, R. H., Jr.
Stevens, W. J.
Stevenson, G.
Stratton, C. B.
Styri, H.
Swartz, M. D.
Sweigard, J. L.
Sykes, C. S.
Tafel, R. W.
Tanner, H. C.
Tatge, R. J.
Taylor, G. C.
Taylor, H. B.
Taylor, J., Jr.
Terwilliger, H. R.
Theriault, R. J.
Thoenebe, H. S.
Thompson, A. S.
Thompson, C. S., Jr.
Thorson, A. W.
Thress, J. M.
Thumim, C.
Ticotsky, M. E.
Tiesler, J.
Timmis, P.
Torrey, D. F.
Triolo, J. F.
Trout, C. L.
Trumpler, A. L.
Tse, F.-S.
Tucker, D. A.
Unger, L. F.
Vanaman, F. H.
Van Osten, P. G.
Van Valkenburgh, R. M.
Varani, U. A.
Viti, J.
Vogdes, R. T., Jr.
Voigt, L. E.
Vondersmith, E. L.
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Wagner, H. D.
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Wallace, G. A.
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Waner, N. S.
Washburn, F. E.
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Weaver, W. H.
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Weber, W. R.
Weinberg, P. H.
Weir, S. B.
Weiss, A.
Welch, P. J.
Welch, W. E.
Wellons, F. W.
West, D. P.
West, J. E., Jr.
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Lesser, W. H., Jr.
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GUAYAMA

Guayama

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HUMACAO

Humacao

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Parker, J. W.
Parker, R. B.
Picozzi, D. A.
Prager, W.
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Roberts, C. A.
Robinson, E. J.
Roddy, F. M.
Rugh, J. M.
Schafer, T. W. D.
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Shaal, L. F.
Sharpe, H. D.
Sheeran, L. A.
Sheldon, A. N.
Simeon, C. J.
Sizer, H. S.
Tanner, F. C.
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Viall, R.
Vierling, A. J.
Wagner, L. E.
Weimar, H. C.
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Wiley, R. C.
Williams, J. H.
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Woodford, P. S.

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Sherrod, C. A.
Sweeney, G. M.
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Hague, R. W.
Haller, L. G.
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Roberts, W. H.
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Farrar, D. F., Jr.
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Henderson, G. A.
Hughes, P. A.
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Morrison, R. T., Jr.
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Corry, R. T.
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Greene, F. H.
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Hay, B. W.
Holdredge, E.
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Neill, F. H.
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Pohlman, D. F.
Reinhardt, R. G.
Robinson, R. F.
Sacks, M. M.

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Sexton, R. M.
Shuff, W. E.
Sweeney, R. L.
Sweet, H. M.
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Wallace, W. M.
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Wilson, R. C.
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Muller, F. G.
Park, P. S.
Ross, J. H.
Short, B. E.
Smith, M. G.
Svensen, C. L.
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Winters, F. G.
Woolrich, W. R.
Young, D.

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McEwin, J. B.
McKean, C. V.
Mooney, J. P.
Whalen, F.

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Hall, E. E.
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Brown, C. H.

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Faires, V. M.
Files, C. W.
McNew, J. T. L.
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Simmang, C. M.
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Vance, H.
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Holsworth, R. C.
Kocmit, O., Jr.
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Schaaf, G. C.
Wagner, F. C.
Wright, B. W., Jr.

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Beesley, G. W.
Berkley, W. E.
Besio, C. A.
Bickel, L. A.
Blanton, B. C.
Butler, F. A.
Chambers, H. E., Jr.
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Comroe, I. H.
Cowles, C. A.
Downing, F. T.
Drandell, J.
Edmondson, D. E.
Gardner, M. K.
Getz, H. E., Jr.
Gregory, W. B.
Grimes, C. C.
Guerrero, J. E.
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Hallaman, C. G.
Hardy, N. G.
Hart, H. A.
Hill, W. E.
Hughes, H. R., Jr.
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Hurt, J. M.
Hyde, G. C.
Ivey, C. E.

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LaGrange, W. A.
Lake, S. T., Jr.
Lee, J. A.
Leyda, H. L.
Lucas, M. W.
Lundberg, G. A.
Matson, R. McK.
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Moeller, W.
Moore, M. J. P.
Murray, F. F.
Noell, M. J.
Noyes, J. A.
Patterson, S.
Pearson, H. R.
Pfeiffer, D. C.
Poole, J. A.
Rabe, F. W.
Rasmussen, R. A.
Rauch, R. T.
Reagor, A., Jr.
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Schmidt, E. F. E.
Shimer, J. M.
Shumaker, C. H.
Silberman, M.
Staples, C. A.
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Wacker, E. J., Jr.
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Wheeler, L. J.
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Jackson, W. C.
Johnson, H. T.
Munson, L. E.
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Stevens, G. W.
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Behrens, K. F.
Berg, L. E.
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Cameron, H. S.
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Downs, J. B. T.
Eblen, W. F.
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Greenwood, M. H.
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Howard, R. E., Jr.
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Kincade, E. C.
King, J. J.
Kinley, J.
Kinzbach, R. B.
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Koch, W. M.
Kotzebue, M. H.
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Lantau, M., Jr.
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Leverett, W. H.
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McDonald, W. A.
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Moore, J. D.
Moore, M. L.
Moss, E. H., Jr.
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Nagai, G. M.
Neilon, C. R.
Netherwood, J. S.
Neuhaus, R.
Nevill, G. E.
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Olsen, O. LaC.
Patton, J. D.
Patton, J. W.
Payne, A. D.
Pechacek, R. E.
Pitts, C. E.
Power, J. A.
Pratt, B. R.
Rahm, F. D.
Reed, M. V.
Robertson, J. M.
Robertson, W. T.
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Ellis, F. A.
Esdaile, H. M.
Farmer, J. T.
Foster, L. C.
Frey, G. W.
Friedman, F. J.
Frigon, R. A.
Garland, J.
Gaucher, J.
Granger, T. S.
Grier, A. M.
Hamer, I. M.
Higginson, T. H.
Hodgson, R. H. C.
Hutchinson, J. B.
Larkin, A. C.
Laurie, A.
Lemmer, H.
Lindsay, D. L.
MacAfee, R. E.
Madgett, H. H.
McGee, J. J.
Muir, W. P.
Nathanson, M.
Newman, W. A.
Noyes, R. R.
Paton, A. H.
Pradl, G.
Rankin, C. J.
Rankin, R. A.
Raskin, F. J. M.
Reeve, D. D.
Robb, C. A.
Roberts, A. R.
Roberts, J.
Robinson, E. A.
Schell, P. C.
Seton, B. W.
Smith, F. F.
Stadler, J.
Steele, A. W.
Stephens, G. A.
Surveyer, A.
Sweet, W. H.
Van Patter, H. S.
Viberg, E. R.
Warner, E.
Weldon, R. L.
Wells, R. F.
Wiggs, G. L.
Wright, L. A.

NORANDA

Gallagher, E. G.

PLESSISVILLE

Bilocq, G. A.
Boisvert, J. B.
Gamache, J. P.
Hebert, A. J. G.

PORT ALFRED

Goloff, M. J.

SHAWINIGAN FALLS

Cornelius, C. T.

SHERBROOKE

Haight, H. V.
Latulippe, L. J.
Maybank, H. A. G.
Morse, J. S.

THREE RIVERS

Butler, E.

WATERVILLE

Gale, W. D.

WESTMOUNT

Durley, R. J.

WINDSOR MILLS

Allan, C. E.
MacKenzie, F. C.

SASKATCHEWAN

MOOSE JAW

Postin, G. D.

SASKATOON

Fisher, F. P.

NEWFOUNDLAND

ST. JOHN'S

Bannerman, D. K.

MEXICO

CHIHUAHUA
Chihuahua

Fierro, S.

DURANGO
Durango

Zalaha, J.

GUADALAJARA
Jalisco

Collignon, M.

LOS MOCHIS
Sinaloa

Steel, J.

MEXICO, D. F.
(Mexico City)

Avilez, I.
Booth, D. M.
Cabrera y Nevraumont, J. M.
Camp, G. D.
Connon, G. W.
Conway, G. R. G.
de la Macorra, J., Jr.
Donahue, J. F.
Lopez, L. M.
Macías, C.
Mahon, W. J.
Martinez, J. J.
Miller, H. C.
Ossman, E. H.
Sensibar, E.
Siefert, G. C.
Tattersfield, J. P.
Tornel, P. M.
Zilboorg, J. M.
Zoller, L.

MONTERREY
Nuevo Leon

Clark, S. W.

MORELIA
Michoacan

Summerlin, I. W.

TACUBAYA
Mexico

del Paso, A.

TAMPICO
Tamaulipas

Aréchiga, L. E.

CENTRAL AMERICA

CANAL ZONE

See Page 193

COSTA RICA

PUERTO LIMÓN

Ross, R. H.

SAN JOSE

Picado, R. M.
Purdy, H. T.

TURRIALBA

Goode, C. B.

GUATEMALA

GUATEMALA

Melendez, V. E.
Viessman, W.

REPUBLIC OF PANAMA

PANAMA

McKay, J. B.

WEST INDIES

ARUBA

ARUBA

artiss, W. L.
oltane, T. K.
Repetko, E., Jr.
annop, L. G.

BERMUDA

HAMILTON

Watlington, E. H.

CUBA

BANES

Mattson, I. F.
Suarez, L. A.

CAMAGUEY

DeVarona-Aguero, P.

CARDENAS

Arias, E. R.

CENTRAL AMERICA

Fernandez, A.

CENTRAL CESPEDES

Fanjul, H. C.

CENTRAL HERSHEY

Elizondo, H. R.
Michelena, J. L.

CENTRAL JARONU

Gonzalez, E. D.

FRANCISCO

Crawley, G. E.

GUANTANAMO

Santamaria, I. J.

GUANTANAMO BAY

Hartin, F. R.

HAVANA

Dallas, C. F.
Gianelloni, V. J.
Gowling, L. E.
Lanier, H. DuB.
MacFarlane, J.
Miller, E. G.
Riera, P. V.
Romanach, J. A.
Skilton, H. I.
Stuntz, J. E.
Wales, R.

MANACAS

Varona, M. C.

MARIANAO

de Goirigolzarri, M.

MERCEDES

Martinez, F. S.

MIRANDA

Koch, E. G.

NICARO

Fitzhugh, R. R.

PERICO

Higginbotham, O.

PIÑA

Guastella, S. F.

PUNTA SAN JUAN

Bancroft, J.

RODRIGO

Oquendo, R.

SANTIAGO DE CUBA

Martel, F. A.

SENADO

Diaz-Compain, J.

CURAÇAO

Echelson, G.
Tuininga, P.
van Wezel, M.

DOMINICAN REPUBLIC

SAN PEDRO DE MACORIS

Kennedy, D. P.

HAITI

PORT-AU-PRINCE

Hill, A. J.

PUERTO RICO

See Page 226

TRINIDAD

POINTE-A-PIERRE

Hearn, H. E.

VIRGIN ISLANDS

See Page 229

SOUTH AMERICA

ARGENTINA

ALTA GRACIA

Olditch, F. W.

BUENOS AIRES

Ballester, R. E.
Beckwith, B. L.
Galloway, F. M.
Llansó, J. J.
Mellor, O.
Weygandt, A. S.

PARANA

Anderson, E. F.

PUERTO LA PLATA

Fletcher, T. F.

BRAZIL

RECIFE (Pernambuco)

Sidler, E. H.

RIO DE JANEIRO

Christoph, O. K.
Gillespie, F. M.
Heslop, P. L.
Ketchum, K. W.
Ware, J. S.

SÃO PAULO

Billings, A. W. K.
DeGray, W. G.
Haag, P. H.
Morland, J. A.
Sodré, L. N.

BRITISH GUIANA

MACKENZIE

Staples, W. J.

CHILE

ANTOFAGASTA

Garey, G. W.

IQUIQUE

Krug, G. L.

POTRERILLOS

Love, C. P.
Lynch, T. F.

RANCAGUA

Broeker, F. G.

SANTIAGO

Barker, H.
Gamboa, F. R.
Kruger, P. F.

TALCAHUANO

Zapata, J. V.

COLOMBIA

BARRANQUILLA

Rippe, C. C.
Santos, A.

BOGOTÁ

Cortes, J. M.
Laserna, A.
Sanclemente, A. H.

CALI

Galvis, R.
Hernandez A, R. C.
Petura, F. E.

CUCUTA

Walsh, C. Z.

MEDELLIN

Foulds, C. VonH.

ECUADOR

GUAYAQUIL

Bermeo-Cevallos, C. H.

PERU

CALLAO

Bastante, A. M.

LIMA

Dasso, D.
Gallese, J. J.
Grieve, A.

OROYA

Hessellund, R.

URUGUAY

MONTEVIDEO

Melrose, R. G. R.

VENEZUELA

BARCELONA

Austin, H.

CARACAS

Umerez-Blanco, F.

LAGUNILLAS

McSweency, W. T.

MARACAIBO

Andara, J. L.
Brewster, J. T.
Hansen, W. O.
Idenden, F. S.
Neal, W.

SOMBRERO

Tullar, I. V.

AFRICA

EGYPT

ALEXANDRIA

Babikian, H. M.

CAIRO

Johnson, T. S.
Meyer, H. F.

UNION OF SOUTH AFRICA

CAPETOWN
Cape of Good Hope

Benning, V. L.

DURBAN
Natal

Reim, E. P.

GERMISTON
Transvaal

Wade, W. A.
Williams, A. F.

JOHANNESBURG
Transvaal

Allen, M. H. P.
Bateman, E. L.
Boyd, C. O.
Brown, T. C.
Clarke, H. G.
Cotterell, W. J.
Johnston, K. M.
Orr, J.
Reunert, T.

O'OKIEP
Namaqualand

Mohler, R. C.

ASIA

CEYLON

COLOMBO

Pattison, R.

CHINA

CHUNGKING

Ku, Y. C.
Mao, E.
Wang, Z.-F.
Wei, Y. F.
Wong W.
Yang, C. Y.
Yang, L. O.
Yang, S. Y.

SHANGHAI

Feng, T.-P.
Harvey, A. H.
Lem, F. Y.
Tang, T.-Z.

TIENTSIN

Kwang, K. Y.

INDIA

AHMADABAD
Bombay

Babaycon, M. A.
Mehta, M. D.
Thaker, S. H.

AMRITSAR
Punjab

Singh, J.

BANGALORE
Mysore

Penning, C. J. H.
Ramaswami, E. K.

BELAGULA
Mysore

Rao, T. N.

BHADRAVATI
Mysore

Channappa, B. K.

BOMBAY
Bombay

Haskell, M. E.
Master, J. N.
Mukerji, M. L.
Nimbkar, V.

CALCUTTA
Bengal

Bentley, H.
Carnes, P. S.

DELHI
Delhi

Bose, K. K.

HASSAN
Mysore

Krishnamurthy, B. S.

JAMSHEDPUR
Bihar and Orissa

Sinha, J. M.

KARACHI
Sind, Bombay

Bhappu, K. K.
Gabla, N. F.

KHARGPUR
Bengal

Pathak, M. L.

KIRLOSKARVADI
Satara

Kirloskar, S. L.

LAHORE
Punjab

Handa, C. L.

NEW DELHI
Delhi

da Costa, G.
Raju, C. S. N.

PATNA
Bihar

Chatterjee, B.

PHILLAUR
Jullundur, Punjab

Singh, N.

POONA
Bombay

Bilimoria, P .D.
Golwelkar, W. G.

SHILLONG
Assam

Purkayastha, P. B.

TRICHUR
Cochin, Madras

Menon, V. K. A.

UJJAIN
Gwalior, Central India

Ram, R. A., Sr.

IRAN

KERMANSHAH

Faridany, H. P.

JAPAN

KYOTO

Ishimura, L. S.
Kuwada, G.

TOKYO

Kamo, M.

PALESTINE

HAIFA

Kurrein, M.

JERUSALEM

Schocken, M. J.

PERSIAN GULF

BAHREIN ISLAND

Blank, H. A.

SIAM

BANGKOK

Thithan, K.

EUROPE

DENMARK

COPENHAGEN

Bak, A. K.

HOLBAEK

Petersen, P. J.

ENGLAND

ACCRINGTON

Lancs

Kenyon, J. M.

ALTRINCHAM

Cheshire

Robinson, E.

BEXLEY HEATH

Kent

Knowles, D. H.

BIRMINGHAM

Warwicks

Fallon, J.
Helliwell, W.
Kego, R.
MacLaren, J. E.
Orcutt, A. H.
Roby, C. F.

BOLTON

Lancs

Aspinall, A. R.

BRADFORD

Yorks

Donaldson, C. S.

BUCKHURST HILL

Essex

Nevard, M. S.

CANTERBURY

Kent

Banks, S. J. E.

CHELLASTON

Derby

Tresilian, S. S.

CLITHEROE

Lancs

Parris, G. C.

CORBY

Northants

Blair, J. S.

COVENTRY

Warwicks

Voce, J. D.

DERBY

Derby

Andrews, H. I.
McIntyre, D. D.

FILEY

Yorks

King, D. H.

GATESHEAD

Durham

Sigmund, M.

HEXTABLE

Kent

Giller, F. S.

HUDDERSFIELD

Yorks

Thornton, J. A.

ISLEWORTH

Middlesex

Sinclair, H.

LEICESTER

Leics

Ritchie, A. P.
Taylor, M. H.

LIVERPOOL

Lancs

Hewitt, R. W.

LONDON

Allingham, H. W.
Bonstow, T. L.
Brownlie, D.
Bruce, A. K.
Carroll, L. D.
Champion, O. H.
Charlewood, C. B.
Chen, C.-Y.
Dean, E. S.
Dunglinson, B.
Ellenberger, W. J.
Extence, A. B.
Fairhurst, T. W.
Flinn, A. V.
Garratt, E. A.
Glasgow, A. G.
Guy, H. L.
Hague, C. K. F.
Hartley, A. C.
Heenan, J. N. D.
Hoffmann, A. F.
Hopewell, G. H.
Horn, F.
Inman-Emery, J. I.
Kemp, J. T.
Kennedy, G. F.
Lee, E. H.
Lytle, J. E.
McEwen, E. G.
McGregor, A. G.
Mills, E. A.
Montgomrey, J. E.
Mowat, M.
Mullen, J. P.
Murray, G. F. J.
Pearce, S. L.
Roberts, E. D.
Robeson, A. M.
Shannon, W. B.
Southwell, R. V.
Sparks, A. C.
Sparks, C. H.
Spratt, H. P.
Stanier, W. A.
Swan, S. R. B.
Thorpe, W. A. C.
Troup, J. D.
Twinberrow, J. O.
Usher, G. C.
Verity, C. E. H.
Verrall, G. T.
Watson, H. D.
Wauchope, G. A.
Whiteford, J. F.
Zoller, R. E.

LONG EATON

Derby

Schofield, F. B.

LOUGHBOROUGH

Leics

Schlesinger, G. C.

MANCHESTER

Lancs

Fleming, A. P. M.
Pailin, G.
Wood, A. L.

MELKSHAM

Wilts

Bealing, E.

NEWARK-ON-TRENT

Nottingham

Samson, J. B.

NEWCASTLE-ON-TYNE

Northum.

Lyon, A. G.

NORTHFLEET

Kent

Walmsley, S. E.

NORTH SHIELDS-ON-TYNE

Northum.

Beavers, G. R.

NORTHWICH

Cheshire

Thornton, B. M.

PETERBOROUGH

Northants

Ferguson, R.

PUTNEY

Few, E. L.

REIGATE

Surrey

Abercrombie, J. H.

ROMFORD

Essex

Owen, A. S. H. A.

RUGBY

Warwicks

Campbell, G. M.

SHEFFIELD

Yorks

Southern, H.

SMALL DOLE

Sussex

Ricardo, H. R.

STOCKPORT

Cheshire

Day, C. C.
Dearden, B. B.

SUNDERLAND

Durham

James, I. G.

SURBITON

Surrey

Cowan, P. J.

TENTERDEN

Kent

Hudson, W. S.

TUNBRIDGE WELLS

Kent

Case, R. C.

WALLASEY

Cheshire

Caine, J. F. K.

WALLSEND-ON-TYNE

Northum.

Brown, T. W. F.

WALTON-ON-THAMES

Surrey

Robinson, I. V.
Selvey, W. M.

WANTAGE

Berks

Thurston, H. G.

WESTMINSTER

Tritton, J. S.

WOLVERHAMPTON

Staffs

Elliott, G. D.
Plummer, G. A.
Thompson, S. J.

FRANCE

PARIS

Garfield, A. S.
Garnier, A.
Huet, A. P. J.
Mayo, P. H.
Ricard, P. J.

GERMANY

BERLIN

Neuhaus, F.

GREECE

ATHENS

Jacks, I. T.

ICELAND

REYKJAVIK

Halldorsson, G.

ITALY

IVREA

Olivetti, A.

MILAN

Jervis, T. J.

ROME

Perrone, P.

NETHERLANDS

DELFT

Dresden, D.

HAARLEM

de Stoppelaar, L. P.

NORWAY

BREVIK

Holter, A.

OSLO

Firing, W.
Foyn, T.

SKOTBU

Lobben, P.

SCOTLAND

ALLOA

Clackmannon

Macnee, C. M.

BRIDGE OF WEIR

Renfrew

Mellanby, A. L.

CLYDEBANK

Dunbarton

Pigott, S. J.

EDINBURGH

Midlothian

Anderson, A.
Douglas, J.
Partridge, H. E.

GLASGOW

Lanark

Burke, N.
Davies, A. W.
Fleming, J. T.
Orenstein, H.
Semple, D. McH.
Ulander, P. H. N.
Weir, The Rt. Hon. Viscount
Yarrow, H. E.

JOHNSTONE

Lanark

Lang, J. B.

PAISLEY

Renfrew

Bruckmann, H. C.
Gibson, J.
White, T.

SWEDEN

BORÅS

Engblom, A. N.

FINSPÅNG

Wiberg, O. A.

NORRKÖPING

Flater, H.

STOCKHOLM

Carlson, A. F.
Lindhagen, M. T.

VÄSTERÅS

Hansson, A. S.

SWITZERLAND

BADEN

Meyer, A.

BERNE

Weyher, T. A.
Zuberbühler, P.

GENEVA

Schmid, W. E.

WINTERTHUR

Buchi, A. J.

TURKEY

ANKARA

Birlik, E.
Taney, A. R.

ERBAA

Ayasun, N.

ISTANBUL

Capa, G. S.
Dunlap, R. M.
Sungur, S.
Taspinar, A. H.
Uran, N. F.

U. S. S. R.

LENINGRAD

Kousmin, S. I.

MOSCOW

Rybkin, I. Z.

OCEANIA

AUSTRALIA

ADELAIDE

South Australia

Green, W.

BRISBANE

Queensland

Axon, A. E.
Evans, D. E.

CASTLEMAINE

Victoria

Burnell, J. G.
Henry, J. S.
Morton, A. B.
Robertson, R. J.

FOOTSCRAY

Victoria

Saenger, G. W.

ISLINGTON

South Australia

Harrison, F. H.

MELBOURNE

Victoria

Ford, A. S.
McMurrich, R. P.
Mealand, A.
Pullar, W. R.
Rigby, E. J.
Smith, V.
Traill, J. W.

MIDDLE BRIGHTON

Victoria

Field, J.

MIDLAND JUNCTION

Western Australia

Mills, F.

REGENTS PARK

New South Wales

Shirtley, S. L.

SPRINGVALE

Victoria

Lewis, K. P.

SYDNEY

New South Wales

Davey, G. I.
Denne, W. W. F.
Dewar, N. W.
Gibson, W. A.
Hart, L. H.
Makhno, V. V.
Palmer, W. J. D.
Price, N. I.
Ratcliffe, F. R.
Rooste, E. E.
Warner, L. T.

JAVA

BATAVIA

Mans, F. J.

NEW ZEALAND

CHRISTCHURCH

O'Hagan, R. S.
Steele, S.

PHILIPPINE ISLANDS

MANILA

Abaya, G. T.
Ames, A. P.
Hellman, W. M.
Kent, G. C.
Lancaon, J. S.
Strauss, J. G.

ADDRESS UNKNOWN

de Arozarena, R. M.
Greene, I. C.

SUMMARY OF GEOGRAPHICAL LIST

UNITED STATES

Including Territories and Dependencies

Alabama	122
Alaska	1
Arizona	28
Arkansas	17
California	1566
Canal Zone	11
Colorado	112
Connecticut	634
Delaware	147
District of Columbia	398
Florida	128
Georgia	170
Hawaii	31
Idaho	13
Illinois	1338
Indiana	377
Iowa	82
Kansas	94
Kentucky	94
Louisiana	172
Maine	86
Maryland	886
Massachusetts	1085
Michigan	691
Minnesota	122
Mississippi	37
Missouri	844
Montana	10
Nebraska	49
Nevada	5
New Hampshire	40
New Jersey	1412
New Mexico	45
New York	4142
North Carolina	139
North Dakota	8
Ohio	1303
Oklahoma	137
Oregon	78
Pennsylvania	2035
Puerto Rico	29
Rhode Island	146
South Carolina	60
South Dakota	15
Tennessee	213
Texas	413
Utah	44
Vermont	27
Virginia	278
Virgin Islands	1
Washington	242
West Virginia	99
Wisconsin	372
Wyoming	9
Total	19582

OTHER COUNTRIES

North America		
Canada	440	
Mexico	28	
		468
Central America		
Costa Rica	4	
Guatemala	2	
Panama	1	
		7
West Indies		
Aruba	4	
Bermuda	1	
Cuba	34	
Curaçao	3	
Dominican Republic	1	
Haiti	1	
Trinidad	1	
		45
South America		
Argentina	9	
Brazil	11	
British Guiana	1	
South America (*continued*)		
Chile	9	
Colombia	10	
Ecuador	1	
Peru	5	
Uruguay	1	
Venezuela	9	
		56
Africa		
Egypt	3	
Union of South Africa	14	
		17
Asia		
Ceylon	1	
China	13	
India	31	
Iran	1	
Japan	8	
Palestine	2	
Persian Gulf	1	
Siam	1	
		58
Europe		
Denmark	2	
England	115	
France	5	
Germany	1	
Greece	1	
Iceland	1	
Italy	8	
Netherlands	2	
Norway	4	
Scotland	18	
Sweden	6	
Switzerland	5	
Turkey	8	
U.S.S.R.	2	
		173
Oceania		
Australia	31	
Java	1	
New Zealand	2	
Philippine Islands	6	
		40
Total		859

SUMMARY

United States, Territories and Dependencies	19582
Other Countries	859
Address Unknown	2
Total	20443

The Design of Mechanical Auxiliaries for TVA Hydroelectric Plants

By H. J. PETERSEN,[1] KNOXVILLE, TENN.

Sixteen hydroelectric power plants have been designed by the Tennessee Valley Authority during the past ten years. Changes and improvements have been incorporated in each successive design, as a result of experience gained in operation of each plant. This applies particularly to mechanical auxiliaries involved. It is this phase of the design which is treated in the present paper, specific reasons being given for adoption of the particular auxiliaries discussed.

During the past decade the Tennessee Valley Authority has designed sixteen hydroelectric power plants which include almost every type of plant except those using Pelton wheels. These plants were all designed by essentially the same organization, which resulted in a succession of improved designs as experiences gained in the earlier plants were incorporated in the designs of the later plants. This is especially true in the case of the mechanical auxiliaries since considerable latitude is permitted in the design of these systems. Consequently, it has been possible to standardize on a good many features of these auxiliaries which have proved themselves in service.

This paper will discuss the design of the various mechanical auxiliaries for hydroelectric power plants as developed by the Authority, outlining the reasons establishing the designs adopted. It is expected that these designs will be superseded by later and better designs as operating experiences and improved techniques dictate. It is not intended that this paper shall represent a comprehensive treatise on hydroelectric power-plant piping and auxiliaries, as these subjects are well handled in numerous textbooks and manuals; however, the scope of this discussion makes it desirable to complement the specific references to TVA design with a review of certain related principles which the author considers good general piping practice.

In general, the term "mechanical auxiliaries" includes those systems usually consisting of one or more machines connected by piping to the main prime movers to aid them in their proper functioning, or other systems distributed throughout the powerhouse for services such as water, air, drainage, and so forth. Because they occupy relatively small spaces and usually are located in out-of-the-way places in the powerhouse, these systems sometimes do not receive their full share of attention and may be slighted in the design. In almost all cases they are vital to successful and economical operation, and thought given to their design in the preliminary stages will pay dividends in the long run in the trouble-free operation of the plant.

However, the amount of time and effort spent in the design of these features should be justified by the results required. Elaborate calculations and long and complicated empirical and theoretical formulas should be resorted to only when the results prescribed make their use unavoidable. Only a novice will carry out a computation to two or three decimals when a lesser number of significant figures will suffice. An experienced designer is aware of this and has available a number of charts, tables, and curves, from which he is able to pick certain approximate values which facilitate his design. Also, a practical designer will always maintain a close liaison with the department that will operate the plant in order to ascertain its likes and dislikes in the matter of machinery and equipment. Many times that department, because of its operating experiences, can make worth-while contributions to the design.

General Considerations

Before reviewing the design of the individual systems, several features common to all systems and on which generalizations may be made will be discussed.

Piping. The fundamental requirement for any piping system is that it shall be functionally correct. All valves and items of equipment should be in proper relation to each other, so that the system operates in the most efficient manner possible.

The best way to obtain this result is by making a valve-operation diagram. Fig. 1 is such a diagram for a governor and lubricating-oil system at one of the Authority's plants. It will be noted that this diagram outlines the entire system, including all valves, equipment, and machinery. From this it is an easy matter to check the over-all design of the system from a functional standpoint. This diagram, made in sketch form before any drawings are begun, serves as a guide to the draftsman in making the detailed drawings. When made into a final drawing, it has the further purpose of aiding the field erection forces in the installation of the piping, and finally it enables the operators to visualize the system as a unit which promotes more efficient operation. Each valve on each system is marked with an aluminum tag describing its function (see Fig. 2). Tags A, B, and C are attached to the valve handwheels underneath the nut, while tag D is attached to the valve-bonnet bolt, to the valve stem, or to the operating chain. Each valve is numbered according to a standard system, and an experienced operator can tell the relative location of a valve in the piping system by its number. The symbols shown on this drawing for the various kinds of valves and items of equipment are standard for the diagrams for all plants.

During the preliminary stages of the design of the powerhouse, adequate consideration should be given to the physical location of the pipe lines. It is absolutely essential that all the various systems be considered as a group in order to lay out the space requirements properly. Close co-ordination between the piping designer, the architect, and the concrete designer is essential at this stage of the design in order to insure proper space for pipe galleries, tunnels, and riser shafts.

Fig. 3 shows a gallery containing a number of piping systems. It will be noted that the gallery is of ample size, that all the pipes are racked along the wall parallel to the building lines, and that they provide adequate headroom. As a contrast, one of the earlier designs showing the piping around the top of a shaft is illustrated in Fig. 4. Note the pipe at angles to building walls, the inaccessibility of valves, some valves upside down, and the general appearance of having been designed one system at a time instead of as a group. The improvement in the oil piping shown

[1] Head Mechanical Engineer, Tennessee Valley Authority. Mem. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of authors and not of the Society.

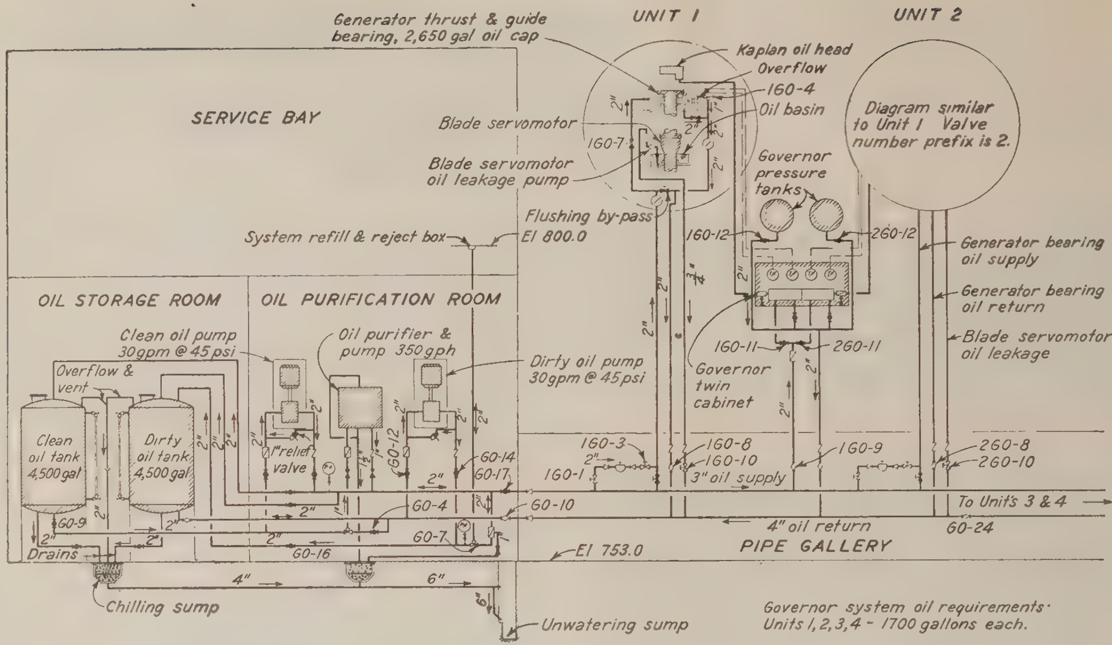

Fig. 1 Valve-Operation Diagram; Governor and Lubricating-Oil System; Fort Loudoun Project
(Units 3 and 4 and some valve numbers omitted for condensation and clarity.)

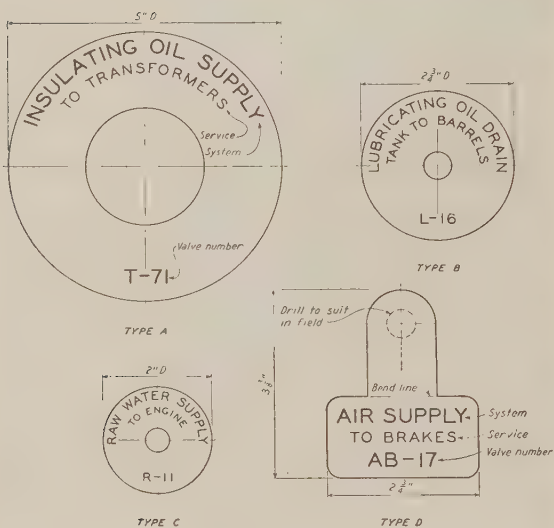

Fig. 2 Valve Marker Tag

Fig 3. Piping in Gallery

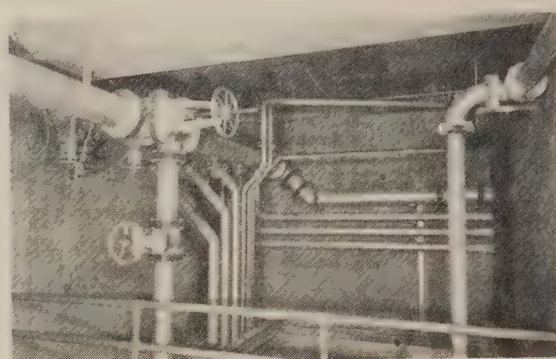

Fig. 4 Piping at Top of Shaft

in Fig. 6, over that shown in Fig. 5, is evident. Fig. 5 has large radii bends combined with welding fittings, pipes crossing each other at angles, pipe in more than one plane, and a multiplicity of valves and piping.

By locating the equipment closer to the walls and by using three-way cocks, as shown in Fig. 6 for a similar system in a later powerhouse, it was possible to eliminate many valves and some of the piping. The use of welding fittings throughout and the uniform spacing of the headers result in a piping design which places the accent on simplicity and good appearance as well as

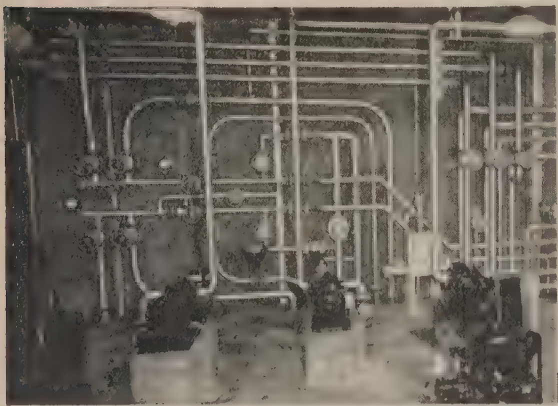

Fig. 5 Piping in Oil-Purification Room in One of Earlier Plants

Fig. 6 Piping in Oil-Purification Room in Later Plant

functionalism. The choice of the operator between these two designs is obvious.

The greatest part of the steel pipe used by the Authority for hydroelectric plants is purchased under A.S.T.M. Specification A 120, standard weight (ASA-B36.10). For certain pipe assemblies where considerable bending or coiling of the pipe is necessary, the pipe is specified as A.S.T.M.-A53, Grade A. Wrought-iron pipe is purchased under A.S.T.M. Specification A72. All cast-iron pipe is bell and spigot centrifugally cast, with class depending upon the pressure. Cast-iron pipe for pressure lines is purchased cement-lined, as it may be had at little or no increase in price, and experience has proved that it is much more resistant to tuberculation. Cast-iron pipe is purchased in 16- or 18-ft lengths, depending upon the supplier. Shorter lengths are seldom used because of the added calking expense.

In some of the larger-size mains, it has proved advantageous to use spiral-welded pipe. This is especially true in drainage lines embedded in concrete where the cost of steel or cast-iron pipe would not be justified. Steel pipe is purchased in random lengths with ends threaded and coupled, or beveled for welding, as the case may be. We find that it is cheaper to weld pipes 4 in. and larger than to use screwed flanges. It also saves time and expense to have complicated welding pipe assemblies shop-welded into subassemblies which are then welded together in the field.

The Authority makes single-line detailed construction drawings for piping systems using pipe 2 in. and smaller and for large-scale yard piping of all sizes. They can be made much faster and, except in some congested areas, they give a satisfactory physical picture. Any congested areas are developed either with standard double-line drawings or by isometric projection. Duplication of views, notations, dimensions, and mark numbers are avoided as they add to the design expense and increase the chances for error.

Where lines lead directly to headwater or tail water, heavier construction is used up to the first valve to guard against possible breakage. A special connection is used where embedded lines emerge from the concrete in the various galleries. In smaller lines a screwed coupling is placed flush with the face of the concrete wall; this permits ready replacement in case of breakage in the line. In larger steel lines a standard cast-iron sleeve is placed flush with the concrete wall, and the exposed pipe is calked therein. Cast-iron lines emerging from concrete are placed with the end of a bell flush with the face of the wall.

Pipes passing through walls and floors are sleeved, with sleeves in floors extending 1 in. above the floor to permit floor washing without water spilling to the room below. Black steel pipe is used for sleeves, except where rust action would prove objectionable, in which case galvanized sleeves are used. Piping systems are provided with drains at the low points to facilitate repairs and to drain water lines which are subject to freezing. A good designer always provides a closure piece in a piping system between two pieces of equipment where there are intervening flanged fittings or valves, instead of designing the system fitting to fitting. This is necessary because fittings and valves seldom are furnished to exact dimensions, and the closure piece takes up the inaccuracies in dimensions. Pipe lines are arranged to provide adequate access to machinery for inspection and repair, and to permit the removal of equipment without the necessity of dismantling large sections of piping. Fig. 6 is a good example of this.

It is of prime importance in the design of a piping system to provide for the maximum ultimate requirements. This is especially true in trunk lines for water-supply mains and plumbing waste lines. Our experience has been that as a project develops, more and more demand is placed upon these two systems. A liberal estimate of ultimate requirements is the best insurance against insufficient capacity in the future.

Next in importance to functionalism in a piping system is that the design shall be economical. This factor involves the proper selection of the material, type, and size of the various pipe, valves, and equipment. It is compounded from a great many elements, some of the more important of which are (1) expected life versus cost, (2) higher pumping costs because of reduced sizes versus smaller initial investment, and (3) increased operating conveniences and reduced maintenance expense versus the added cost for obtaining these features. If these three points are carefully evaluated in the design of each system, it is fairly certain that an economical design will result.

From past experience and from technical data on the subject, the expected life of a pipe or valve may be closely estimated. Obviously, the longer life is more desirable, but it is necessary for the designer to balance the extra years of service against the added cost necessary to obtain it.

The size of each pipe line, once the maximum flow is established, is almost entirely a function of the pressure drop due to friction in the line. It is necessary to investigate each line for head loss, using the friction characteristics for the fluid carried. Sizes of equipment connections should never be construed as indicative of proper pipe size. In most cases the size of pipe mains for water systems is dictated by the flow necessary for fire protection. In evaluating pumping costs and pipe sizes, it must be remembered that station power is cheap in a hydro plant. On the other hand, most of the added cost of a larger-size pipe is the material itself, since the labor of installing each-size line is practically the same. It is advisable not to have too many different

sizes and kinds of pipe in a plant since this requires a larger stock of replacement parts for valves and specialties.

The amount of money that should be spent for operating conveniences and reduced maintenance is largely determined by experience and judgment. A good many operating conveniences may be had without extra cost if the systems are properly designed. Valves should be located so that their purpose is apparent at a glance, and they should be accessible, especially those required in an emergency, using chain wheels where necessary. There should be sufficient unions in screwed lines to facilitate disassembly. All waste and drain lines, especially those buried in concrete, should have cleanouts at strategic points. The practice of burying plumbing or other piping in walls should be avoided; it is much better to provide access corridors behind the fixtures for service and maintenance.

Valves. The function of a valve in a piping system is either to regulate the flow of the fluid in the line or to sectionalize a piece of equipment or part of the system. Globe valves are used for the first function and gate valves for the second. Since most of the valves in hydro plants require only infrequent operation and are seldom used for throttling service, gate valves are used in the majority of cases. Globe valves are used on by-passes around special valves and regulators with a gate valve on either side of the equipment by-passed.

The Authority prefers a rising-stem valve to a nonrising-stem valve because the position of the stem indicates the position of the disk and because the stem threads are not in contact with the fluid. The O.S.&Y. type of rising-stem valve has the further advantage that the stem threads are outside the valve when open, making it possible to examine and lubricate them with the valve in service. The additional headroom which is required for the rising-stem valve over a nonrising-stem valve is of small importance.

Globe valves usually are purchased with plug-type disks because that type of disk gives excellent throttling service, has great resistance to wear, and is easily repaired. The so-called composition disk has not proved as satisfactory for throttling and wears out quickly. Gate valves 2 in. and larger usually are purchased with double disks, and smaller than 2 in. with wedge disks. A double-disk valve gives a tighter shutoff and is more easily repaired in the field. Sometimes a larger-size wedge-disk valve is used when it is impossible to install the valve with the stem up. In this position the disk of the double-disk valve is liable to jam. Gate valves smaller than 2 in. are purchased with wedge disks because, in our opinion, the various parts in a double-disk valve in those sizes are too small.

Valves 2 in. and smaller, both gate and globe, usually are purchased all brass because the extra cost involved is nominal. Larger valves are purchased iron body, bronze-mounted. The use of cleanouts in valve bodies has not been found necessary, nor has it been necessary to specify air-tested valves. Pressures encountered in hydroelectric plants seldom call for by-passes around valves, although at times it has been necessary to provide gearing for the larger-size valves. This, of course, is a function of the pressure on the valve and is investigated for each condition. Larger valves which require fast operation, or are remotely controlled, are fitted with motor operators.

Check valves are used on the discharge of pumps and at all other places where it is imperative to prevent backflow of the fluid. Swing-check valves are preferred, but in some cases lift-check valves are used, especially for upward flow. A balanced-disk or spring-loaded check valve is used to prevent slamming and excessive surges in the higher-head lines.

Bolts and Gaskets. Flange bolts are purchased under A.S.T.M. Specification A 107, with regular unfinished square heads and heavy unfinished hexagon nuts. In rare cases where the bolts are inaccessible and subject to excessive corrosion, such as for fastening inlet gratings to the upstream face of a dam, stainless-steel bolts are used. The thinnest possible gasket consistent with the roughness of the flange, is used, with no gaskets over $^1/_{16}$ in. Ring gaskets are used in preference to full-face gaskets as they make a tighter joint.

Hangers, Supports, and Expansion Joints. In a well-designed piping system the weight of the pipe and its contents are always supported so that no stresses are transmitted to any of the valves or machinery. The stresses introduced in equipment in this manner, coupled with vibration, may eventually break the casting of a pump or valve. Standard commercial hangers are used where possible, as they are cheaper than designing special hangers for individual applications.

While the temperatures encountered in hydro-plant piping systems do not give rise to many expansion problems, the long lengths of the lines in a multiunit plant do call for some expansion design. The Authority uses pipe bends and offsets, where possible, in preference to guided slip joints with packing boxes. Sometimes, where vibration is coupled with expansion in a pipe line, a corrugated type of joint is used.

Painting. All piping systems of the Authority, including all insulation, are painted for appearance, protection, and identification, in accordance with a standard color code. Each system has its own color, and this standard is maintained throughout all the plants. The whole system is first painted with an aluminum paint, and identification is obtained by painting the periphery of flanges or by painting an occasional screwed elbow or tee, as the case may be, with the color of that particular system. If the length of line between these identification points is long, a narrow band is painted around the pipe at intervals. Arrows are sometimes painted on pipe lines where the direction of flow is not evident at a glance.

Control Valves. Control valves are used by the Authority in hydro plants for pressure regulating, temperature regulating, pressure relief, and maintaining liquid level. Pressure-regulating valves are used for reducing the water pressure to generator air and oil coolers, turbine oil coolers and runner seals, air-conditioning equipment, air-compressor jackets and aftercoolers, and general plant services. Where the pressure drop is low direct-operated spring-loaded diaphragm valves are installed, while pilot-operated valves are used for high-pressure drop. The pilot-operated valves are of the hydraulic or pneumatic type. The pneumatic type has proved more satisfactory for use on raw river water, as trouble has been experienced with the hydraulic type owing to fouling of the pilot valves, necessitating the installation of strainers in the pilot lines. We have found that filters installed in the air lines ahead of pneumatic valves are a good investment. Small valves and valves used for dead-end service are single-seated, while large valves are double-seated. A pressure gage is always installed on either side of a reducing station.

The sizing of pressure-regulating valves for hydro plants presents a problem because of varying headwater elevations. Since the valves must pass the required amount of water under any head condition, a valve large enough under low headwater may be too large under maximum headwater levels. Likewise a regulating valve on plant service sometimes must also be used for fire protection. This means that water requirements must be estimated very closely, and valves must be selected which have characteristics best suited for their particular service.

Two types of temperature-regulating valves are used, i.e., solenoid and motor-operated from a thermostatic element. Solenoid valves are used on air-compressor jackets and similar services to control cooling water so as to prevent flow and condensation when the service is intermittent. Motor-operated valves, actuated from thermostatic elements, are used on air-

conditioning equipment and to control the cooling water to generator air coolers in remotely controlled plants.

Both spring-loaded hydraulic relief valves and diaphragm back-pressure valves have been used after pressure-regulating valves. Spring-loaded hydraulic relief valves are used in the small sizes and where the comparatively wide relief range is not objectionable. Diaphragm back-pressure valves have proved more satisfactory for large sizes and where close control is required. Relief valves are sized to pass the total amount of water which the pressure-regulating valve will pass under highest head conditions.

Level controls are used in open tanks and station drainage sumps, and for depressing tail water in the draft tube when units are motored. The types used are: the ordinary balanced float valve to maintain a constant level in an open tank from a pressure main, float-operated mercury switches and cam-type float switches to start and stop pumps, and diaphragm valves actuated by float-operated pilots and compressed air to depress tail water in the draft tube when units are motored for condenser operation.

Motors. All machinery in the hydro plants of the Authority is driven by electric motors because of the availability of a reliable power supply and because of the dependability of the modern motor and its simplicity of control.

Motors driving station auxiliaries are general-purpose, N.E.M.A. standard, with class A insulation. Motors driving constant-speed equipment, such as pumps, compressors, and ventilating fans, are squirrel-cage induction type with normal torque and starting current; those for variable speed and torque, such as cranes and hoists, are wound-rotor slip-ring type. Integral-horsepower-size motors are rated 440 volts, 3 phase, 60 cycles. No auxiliaries require motors large enough to make voltages above 440 economical. Fractional-horsepower sizes are 110 or 220 volt single phase. A few auxiliaries, such as emergency turbine guide-bearing, lubricating-oil pumps, and important valves, are provided with 250-volt d-c motors operated from the station service batteries.

Motors in locations where windings might become damaged by water or oil are provided with moistureproof insulation and splashproof or dripproof frames; others have open frames. In very damp locations it has been found necessary to apply strip-type heaters or to circulate a small amount of current through the motor coils to avoid sweating while the motors are shut down.

Controls are centralized on auxiliary-power switchboards for groups of related equipment. Each motor circuit contains a manually operated disconnecting switch, a set of fuses as protection against short circuit, and a magnetic contactor for full-voltage starting and thermal overload protection. Push-button stations for manual control generally are located at or near the motors. Motors under automatic control also have a push button which provides manual control for inspection. Power supply is sufficiently duplicated to assure continuous operation of auxiliaries during usual maintenance and under any reasonable fault conditions.

Individual Systems

Compressed Air. Compressed air is used in the hydro plants of the Authority for operating pneumatic tools, for generator brakes, for pneumatically operated regulating valves, for depressing tail water below the runners for synchronous-condenser operation in those plants where tail water level is normally above the runner, and in one plant (Kentucky) for preventing the formation of ice on the spillway gates.

Pneumatic tools operate most efficiently using air at 80 to 100 psi, and since the other uses for compressed air require pressures of less than this, 100 psi has been adopted as a standard pressure for the compressors. One stationary 105-cfm compressor has been found to have sufficient capacity for all normal plant needs in all plants except Kentucky. This is supplemented by a portable machine of approximately 100-cfm capacity at each plant which is used when major repair work requires additional air capacity.

The Authority has standardized on a horizontal, crosshead type, double-acting, water-cooled, 325-rpm, V-belt-driven, single-stage compressor for the stationary machine. Its low rotative speed and low bearing loads, together with its all-round rugged construction, make it a most dependable machine, capable of giving continuous service over a period of many years with very little maintenance. Dual control is used with these compressors. This control can be set to operate the compressor continuously when there is a steady demand for air, or it can be set to start and stop the compressor automatically when the demand for air is intermittent. The latter control is the one normally used, with the motor being operated by a pressure switch on the receiving tank. All compressors unload to atmosphere to prevent starting the motors under load.

The portable compressor is an air-cooled, single-acting, 2-stage, motor-driven machine complete with intercooler, air receiver, wheeled truck, and all standard accessories. The motor is controlled by a magnetic starter and push-button station mounted on the machine. The compressor is motor-driven because electrical outlets are provided throughout each plant area which make it possible to use the compressor either in isolated locations, such as up on the roadway crossing the dam, or in the powerhouse to supplement the stationary machine.

A water-cooled aftercooler is always used after single-stage compression to remove the excess moisture in the air before it condenses in the distribution system. An "air through the tubes" type of aftercooler with moisture receiver at one end is installed in the discharge piping. The small amount of moisture and oil from the compressor cylinder carried over to the distribution line is removed by traps. These traps are placed at low points in the line for gravity drainage.

An air receiver is placed in the discharge line after the aftercooler to damp the pulsations of the reciprocating compressor and to provide storage of compressed air when the peak demand is greater than the capacity of the compressor. Since the air cools in a receiver, its dew point is lowered and additional condensation takes place, which is removed by a blowoff or a trap. Because air is compressible, the damage resulting from a ruptured receiver is likely to be severe; consequently, the Authority always specifies that the tank shall be fabricated in accordance with the A.S.M.E. Code for Unfired Pressure Vessels.

Certain features of design are observed in the layout of the compressed-air system; some of these are essential and others are desirable. The air intake is connected to outside air whenever possible and fitted with a filter. When this is not feasible, the compressor is located in the lower levels of the substructure where the air is relatively cool and clean. A valve is never placed in the discharge line between compressor and receiver without an intervening safety valve. All air-hose connections are made at the tops of headers to prevent picking up moisture. Cooling water is discharged through an open funnel so that it may be tested for temperature by feeling. The Authority uses black steel uninsulated pipe for air lines. Internal corrosion from condensate has not evidenced itself up to the present time.

The use of air for depressing tail water below the runner[2] and for protection against ice at spillway gates[3] has been fully de-

[2] "Hydraulic Turbine Practice of the TVA," by H. J. Petersen and J. F. Roberts, *Mechanical Engineering*, vol. 65, 1943, pp. 237–244.

[3] "Kentucky Dam Project of TVA, Part II," by C. L. Norris and R. M. Gardner, *Power Plant Engineering*, vol. 49, March, 1945, pp. 79–81.

scribed in other articles and will be omitted from this paper.

Governor and Lubricating Oil. The Authority uses the same oil for the governor pressure system and for the lubrication of the generator thrust bearing. Some difference of opinion between the governor and generator manufacturers has arisen lately over the viscosity of this oil. The generator manufacturers prefer an oil with a viscosity in excess of 300 S.S.U. at 100 F, while the governor manufacturers prefer this viscosity to be around 250 S.S.U. at 100 F. In order to avoid the necessity of two separate oil systems, oil with a viscosity of 300 S.S.U. at 100 F will be purchased in the future.

We have found that purifying oil by centrifugal force is satisfactory and economical, and we have standardized on that type of machine. A self-contained and enclosed stationary 350-gph unit, complete with centrifuge, motor, pump, electric heaters, temperature control, and all standard accessories, is purchased for each plant. In the early stages of the Tennessee Valley Authority program consideration was given to the use of a portable machine which could be transported to individual plants as required. A central purifying depot, where oil from all plants could be shipped for processing, was also considered. These were both ruled out because of operating inconveniences involved and because the cost of a purifying system is a very small percentage of the total cost of each plant.

Two tanks are provided for each project, one for clean oil and one for dirty oil. Each tank has sufficient storage for the oil in one unit thrust bearing, plus the oil in one governor system, plus a few hundred gallons of extra capacity. When space permits, these tanks are located in the powerhouse adjacent to the oil-purification room. In plants where no space is available the tanks are buried in the fill outside the powerhouse wall. Tanks placed in this manner are designed with an extra $^1/_{16}$ in. of metal to allow for corrosion. In either case the storage tanks are always located so that they may be filled by gravity from the ground level and still provide a positive head on the pumps.

One clean-oil pump and one dirty-oil pump are furnished for each powerhouse. The pumps are standard positive-displacement type with built-in relief valves and are rated 30 gpm when pumping oil at 50 F. A complete piping system is installed at each plant between the units and the purification facilities. The Authority uses black steel, scale-free uninsulated pipe for these oil lines. Experience has proved that it is difficult to prevent leaks in piping using screwed malleable fittings, especially when the oil is hot, as around the centrifuge. Consequently welded joints are used throughout, except at valves and unions, with forged-steel socket welding fittings used for sizes 2 in. and smaller. Oil-resistant gaskets are used when it is necessary to use flanges. No matter how well a pipe line is cleaned, there will always be scale and welding particles carried along in the oil, especially when starting the system. For this reason a filter is installed in the line leading to each unit and a standard Y-type strainer with $^1/_{32}$-in. perforations is used on each pump suction. These filters have 0.005-in. openings through metal slots and are equipped with built-in knife-blade cleaners.

Insulating Oil. Because of its different characteristics from lubricating oil, a separate oil-purification system is installed in each plant for the insulating oil used in the transformers, circuit breakers, and other oil-filled electrical equipment. The same type of machine is used as just described, but because of the larger volumes handled, its capacity is increased to 600 gph. A filter press which raises the purifying capacity to 900 gph is added to this machine to remove the colloidal carbon from the oil.

One clean- and one dirty-oil pump are installed in each plant. They are of the same type as the pumps for the lubricating-oil system, with the capacities increased to 100 gpm each. At some plants it is possible to return the dirty oil from the switchyard to the storage and purification rooms by gravity, and no dirty-oil pump is supplied. One clean-oil tank, one dirty-breaker-oil tank, and two dirty-transformer-oil tanks are provided at each plant. In this way the breaker oil and transformer oil are never mixed after they once have been used. Each of the circuit-breaker-oil tanks (clean and dirty) has storage capacity for one breaker, and the two dirty-oil transformer tanks have a total storage capacity for one transformer.

The same comments apply for the piping as for lubricating oil, except that black wrought iron is used for underground lines in the switchyard. The supply and return lines in the switchyard are not permanently connected to the transformers and breakers; instead, a valve is located in each line near the equipment and a flexible hose is used when filling or draining is necessary. This hose has a specially treated synthetic-rubber lining and is equipped with a flexible ball-joint fitting to facilitate attachment to the service valve. Because the oil is changed at infrequent intervals we find this arrangement entirely satisfactory.

Runner-Hub Oil. In the main river plants which have Kaplan turbines the runner hub is filled with a heavy oil for lubricating the internal blade-operating mechanism. This oil has a viscosity of approximately 1700 S.S.U. at 100 F. Whenever the runner is dismantled for inspection or repair, it is necessary to remove this oil, and since there are approximately 1500 gal of oil in each hub, the Authority salvages it for re-use.

A black steel pipe header is installed in the access gallery located below the units in each plant. Branch headers lead from this line to the draft-tube access doors of each unit, and connection to the hub is made with a flexible hose. An 8-gpm motor-driven screw-type portable pump and two portable storage tanks are provided for transporting and storing the oil during the dismantling of the unit. Because of the infrequency of this operation, the storage tanks and the pump were made portable so that they may be transferred from plant to plant as required. Two storage tanks were supplied to reduce the volume of each tank to facilitate handling.

Raw Water. Raw water is used in the Authority's plants for cooling water for the main generators, for the air compressors, and for the air-conditioning equipment, and in some plants for fire protection, for lubricating the turbine guide bearing, and for supplying make-up for the treated-water system.

In the low-head plants water is taken from the scroll case of each unit and pumped through a multiple basket strainer having $^3/_{16}$-in. openings to a header leading to the station system. One pump is located at each unit to insure a positive cooling-water supply, and a stand-by pump is installed in the service bay of each plant. This latter pump is connected to the forebay by a separate intake and can supply any unit with cooling water in an emergency. All the pumps are connected into a header running the length of the powerhouse so that the chances of any one unit being without cooling water are remote.

The pumps are horizontally split-case, double-suction, single-stage, volute-type centrifugal units. Specifications include enclosed bronze impeller with wearing rings, bronze shaft sleeves through the stuffing box, water-seal piping, and deep stuffing boxes. Experience has proved that this type of pump will give years of service with little maintenance except occasional repacking or lubrication of the bearings. A booster pump is installed in each plant service system using raw water where the forebay pressure is insufficient for a gravity supply. Sometimes the operating requirements of two or more of these systems make it possible to supply them with water from one pump. Quantities involved are usually low, and single-stage single-suction pumps are used. The Authority has found the close-coupled type of pump very satisfactory for this service.

In the high-head plants a separate intake from each unit pen-

stock is connected to a central manifold from which the individual systems are supplied, with pressure-reducing valves installed as required. An air-cushion chamber is installed on the manifold to absorb the water hammer caused by the operation of the system. When the raw-water system supplies the make-up for the treated-water system, intakes at different levels are provided in the dam to enable the operator to take advantage of the varying water temperatures, turbidities, and chemical contents of the lake water at different levels.

The Authority uses black steel pipe for exposed raw-water piping, except that galvanized steel is sometimes used in the smaller sizes. Welded joints are used, except that flanged joints are used around equipment and valves. Gaskets are $^1/_{16}$-in. red rubber. Raw-water lines buried in the ground are cast iron, bell and spigot, in accordance with standard waterworks practice. Black wrought-iron pipe is used for buried lines in sizes $1^1/_2$ in. and smaller, for sprinkler and general yard service.

Lines are insulated in locations where sweating of the cold pipe would prove objectionable. Standard commercial insulating felt molded in 3-ft lengths and covered with 8-oz canvas jacket is used for straight runs of pipe. Fittings and irregular surfaces are covered with a layer of hair felt sealed with a waterproof-tape wrapping and a coating of asphaltic sealing compound. Insulation of 1 in. thickness has been found sufficient to prevent sweating for the water temperatures and air humidities encountered in the low-head hydro plants, but $1^1/_2$ in. of insulation is necessary in the high-head plants because of the colder water. Regranulated cork has been used in some instances to prevent sweating, but unless it is applied very carefully and built up to a thickness of $^1/_4$ in. or more, it is not as satisfactory as hair felt. This is especially true in the high-head plants where water temperatures sometimes go as low as 36 F. Raw-water lines subject to freezing temperatures are insulated with built-up hair felt. The required thickness is a function of the temperatures encountered and the amount of water passing through the line and is investigated for each condition.

Treated Water. Treated water is used in the various hydro plants for all sanitary services, and in some plants for fire protection. Whenever possible the Authority obtains its treated water from a near-by community, even if this involves laying a mile or two of transmission main. We have found it to be more economical to purchase water in this manner than to install a treating system in the plant, requiring the services of an operator. However, it has been necessary to install water-treating systems in most of the projects because of their isolated locations.

At some projects the plant operators live close at hand and use water from the system. This increases the required capacity to the point where a conventional rapid sand gravity filter plant, with mixing chamber, coagulation basin, dry-chemical feeders, and chlorinator is justified. The water treated in this area is such that alum and soda ash or lime are the reagents used. At some of the projects the demand is 10 gpm or less, and pressure filter plants are installed. Originally, pot feeders actuated by differential flow across the orifices were installed for chemical feeding in the pressure filter plants, but our experience with them has not been satisfactory, and they have been replaced with solution-feeder pumps. Pot feeders are difficult to keep regulated, especially when the temperature of the water supply varies considerably as it does with intermittent use.

System pressure is obtained by using elevated storage tanks which usually are transferred from the construction activities at the project. At some of the high-head plants, where such a tank was not available and where raw water for fire protection could be obtained from pond storage, a pneumatic tank was installed for the treated-water system pressure. However, the controls for these tanks require a great deal of attention to keep them in proper working order, and an elevated storage tank is much preferred for this service.

The foregoing comments for pumps, piping, and insulation of the raw-water system apply to the treated-water system also except that in the latter case more galvanized-steel pipe is used.

Fire Protection. Raw water is used for fire protection in all plants where the minimum lake level provides sufficient pressure. The Authority has adopted 40 psi at the hose nozzle as the minimum pressure for this service, but a pressure of 60 to 70 psi is preferred and is used when obtainable. In plants where this pressure is not obtained, fire protection is obtained from the treated-water system or by means of a booster pump.

Fire-hose racks containing 50 ft of $1^1/_2$-in. unlined linen hose with cast-brass nozzles are located throughout each powerhouse at strategic points for inside fire protection. Standard 4-in. A.W.W.A. fire hydrants, fitted with two $2^1/_2$-in. nozzles, are located throughout the plant area and switchyard. A fire-hose cart equipped with a reel and three 50-ft lengths of $2^1/_2$-in. double-jacketed, flat-cured, cotton, rubber-lined hose with rocker-lug couplings is provided for switchyard fire protection. This is supplemented by a wheeled cart containing two 100-lb cylinders of CO_2. National standard fire-hose threads are used for the $2^1/_2$-in. hose and standard-iron-pipe threads for the $1^1/_2$-in. hose. Standard underwriters' play pipes are provided for outdoor fires, except that for electrical fires a nozzle is used which atomizes the water spray to the point where it will not conduct electricity.

Considerations were given to installing built-in fire-protection systems, using either water or CO_2, in the switchyards. These were ruled out because the remote possibility of these fires would not justify the cost of the installation. Sometimes transformers are installed very close to the powerhouse, as on the draft-tube deck. In such cases a fire might cause considerable damage, and built-in water systems, using fog nozzles, are installed. These systems are set off either manually by remote control, automatically by thermostats, or by the tripping of the transformer differential relay which is actuated by a short circuit in the transformer.

Carbon Dioxide. In addition to water fire protection, a complete CO_2 system is provided for each plant. This system protects the main generators, the oil-purification room, and the oil-storage rooms. The CO_2 system is designed in accordance with the standards of the National Board of Fire Underwriters for class A systems, which calls for total flooding of enclosed spaces.

A central supply of 50-lb storage cylinders sufficient to provide the gas concentration for one generator called for by the standards mentioned, is connected to all the generators by a complete piping system. This system is energized either manually by remote control or automatically by thermostats located in each generator. A separate system of cylinders and piping is provided for the oil-purification and storage rooms. It is controlled either manually or by thermostats in the room being protected. When this system trips off, all doors and openings automatically close to confine the gas, and pumps, ventilating fans, and other moving equipment shut down. The gas-discharge nozzles in an oil-purification room for one plant may be seen in Fig. 6. Portable containers of carbon dioxide in the 4- and 15-lb sizes are located at various points throughout the powerhouse to supplement the built-in systems.

Drainage and Unwatering. Roof, deck, and floor drainage in each plant is discharged either to the forebay or the tailrace whenever possible; however, it is necessary to carry some of the drainage to the station sump during periods of high tail-water levels. Galvanized steel pipe usually is used for roof drains, and black steel pipe for deck and floor drains. Screwed cast-iron drainage fittings which have recessed shoulders are used for all

systems. Expansion of roof-drain riser pipes is taken care of just under the roof either by fitting offsets in the pipes or by patented slip joints made for that purpose. Lines are uninsulated, except that deck drains are insulated against sweating when they run above false ceilings in the office areas.

Lead and special-alloy cast-iron lines were originally installed for battery-room drains as a protection against acid. This has been changed to standard steel pipe because the weak concentration of the acid solution does not justify the extra cost of special pipe.

The station sump in each plant is unwatered by two 300-gpm, float-controlled, deep-well-type, vertical-shaft turbine pumps operating in echelon. This type of pump keeps the motor above the wet conditions of an open sump. Very little sand or abrasive matter is present in the water, so enclosed or semienclosed impellers are specified. The pumps are bronze-mounted and oil-lubricated automatically by a solenoid valve energized with the motor circuit.

At times it is necessary to unwater the draft tube of a unit for inspection. Each unit draft tube is connected to the station sump with a large-size cast-iron line containing a sectionalizing valve. The first project of the Authority had the outlet in the bottom of the draft tube, but this soon filled up with sediment and later projects have this outlet about 15 in. from the bottom of the draft-tube floor. Most plants have two deep-well-type vertical unwatering pumps which range in capacity from 2500 to 5000 gpm, according to the volume of the draft tube. They are manually operated and can augment the station sump pumps in an emergency. In those plants where sufficient head is available, unwatering is done by eductors. Since this pressure water comes from the unit penstocks and since it is possible for the penstocks to be empty, the station drainage eductor is supplemented by one motor-driven sump pump. No supplementary pump is installed for draft-tube unwatering as unwatering operations may be scheduled for the times when a penstock is full of water.

Piezometers. The Authority installs piezometers in the various water passages at each plant to check the hydraulic performance of the units, penstocks, and spillways and to operate flowmeters. These lines are $^3/_4$-in. type K, extra-heavy, hard-copper tubing with solder-joint streamline fittings. Lines are installed carefully to avoid pockets and have a minimum pitch of 1 in. in 10 ft from the connection at the water passage to the terminal board.

Plumbing. In general, the design of the plumbing systems in the hydro plants of the Authority follows the "Recommended Minimum Requirements for Plumbing" established by the Subcommittee on Plumbing of the U. S. Department of Commerce.

Water closets and urinals are provided with flush valves, with those on the urinals being foot-operated. All lavatories are equipped with mixing faucets. Hot water is supplied by thermostatic-controlled tanks with immersion-type elements. These tanks vary in size from 10 to 120 gal. Where the hot-water lines are of considerable length, a small circulating pump is installed; however, in some instances it has been found more economical to purchase more than one heater, thus eliminating long runs of piping. A mixing valve is used in the showers to prevent the possibility of scalding. Drinking water is furnished by self-contained electric coolers located throughout each plant. Wall fountains supplied with chilled water from the remote connections on the water coolers are also used. Acid-resisting, enameled, cast-iron sinks have replaced stoneware sinks which were used in the battery rooms of some of the earlier hydro plants.

Type K, extra-heavy, hard-copper tubing with solder-joint fittings is used for all sanitary-water service because its small additional cost is justified by its longer life and its ease of installation. Air chambers 18 in. long are installed as close to the fixtures as possible to absorb the surges caused by quick-closing faucets or valves. Water pressures are maintained at approximately 50 psi, with no pressures exceeding 85 psi at a fixture. Excessive pressure results in leakage, water hammer, and high maintenance costs. Valves are provided at each fixture to permit repairs without shutting off a whole group of fixtures. Vacuum breakers are installed on all fixtures that are directly connected to the water supply to prevent polluted water from being siphoned into the water lines. Although ordinarily the pressure in the water lines prevents the entrance of polluted water, sometimes a partial vacuum occurs which causes siphoning unless vacuum breakers are installed. Floor drains in the toilet rooms have been eliminated in the later plants because they are so infrequently used that water evaporates in the traps and sewer gas enters the room.

All exposed soil and waste lines are standard-weight, galvanized, wrought-iron pipe with black, screwed, cast-iron drainage fittings. Extra-heavy cast-iron soil pipe and fittings are used for embedded lines. Galvanized wrought-iron pipe with black malleable-iron fittings is used for vent lines; where possible, the vents are all connected to the main vent stack to reduce the number of pipes projecting above the roof. A septic tank is installed in each plant with effluent discharged to tail water.

Hot-water piping is insulated with standard molded 3-ft sections of 6-ply preshrunk corrugated asbestos paper, 1 in. thick, and covered with 8-oz canvas pasted on. Fittings in hot-water lines are covered with insulating cement.

Heating, Ventilating, and Air Conditioning. Since the powerhouses are relatively large, with limited occupancy, and the materials of construction are such as to absorb large quantities of heat, the power requirements for general heating to comfort conditions are greater than can be justified. Consequently, only those areas where attendance is required are heated to 72 F, while the rest of the spaces are heated only as required for the relief of dampness or excessive chill.

All heating is by electric-resistance heaters. Except for the air-conditioned areas, heaters, in general, are thermostat-controlled, fan-type, 440-volt, three-phase units, with auxiliary on-off-auto manual switches. In some of the smaller areas 220-volt, single-phase, thermostat-controlled heaters are used. They are either gravity convection heaters recessed in the walls or suspended fan units as space or architectural considerations may require. In addition to these permanent heaters, outlets are provided throughout each powerhouse to which portable electric heaters may be connected as desired. In some of the powerhouses air used for ventilating electrical equipment is recirculated, thus utilizing the heat dissipated. In one plant this reclaimed heat amounted to more than 100 kw.

Ventilation is provided in the several areas of each plant according to the need for human comfort, for the dissipation of heat from electrical equipment and solar radiation on roofs, walls, and decks, and for the relief of dampness. In a number of plants where the arrangement and design of the electrical equipment require, parts of the electrical apparatus, including the main generator leads, are ventilated to prevent damage from overheating. Ventilation is provided in the tunnels and lower galleries for the relief of excessively damp and stagnant conditions.

The scheme of ventilation is a problem which must be solved in each individual plant. The usual practice of the Authority is to introduce air into the generator room and lower parts of the service bay and to exhaust this air through the other spaces to be ventilated by separate fans. In a few plants the fresh air is introduced by gravity, but in most powerhouses a number of supply fans are employed. The arrangement of each powerhouse usually is such that most of the more important spaces adjoin the generator room or are connected to it by open passages. These spaces are grouped together according to location and similarity of ventilating requirements into a number of groups, and each group is

served by a separate exhaust system. In the semienclosed type of hydro plant, where the intake deck is just above the generators, the excess heat from the generators and from solar radiation on the deck above escapes through louvers in the generator hatch covers over the machines. These louvers may be closed in winter, if desired, to conserve heat.

For convenience of maintenance and for economy of space, all the fans in the powerhouse are grouped usually into two separate fan stations. Fig. 7 shows an exhaust-fan station for one plant. It may be seen from this view that both direct-connected and

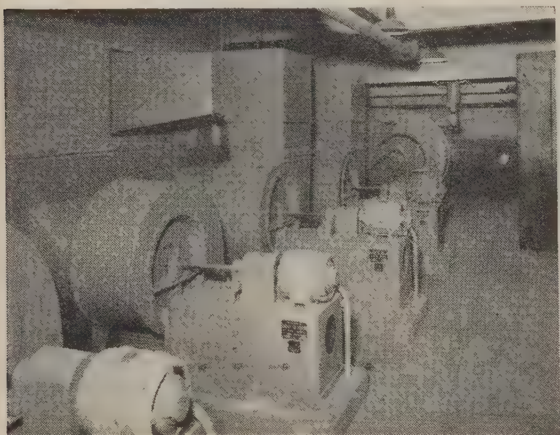

Fig. 7 Exhaust-Fan Room

belt-driven fans are used. When the discharge pressure is fairly constant, fans with forward-curved blades at low rotative speeds are sometimes used. However, most of the fans used by the Authority have backward-curved blades because of their high efficiencies over a wide range of performance and because of the nonoverloading characteristic of that type of blade. It is also better suited to high speeds and hence more adaptable to direct connection. Inlet dampers are used where close control of the discharge pressure or of the delivery of a fan is desired. These dampers are operated by either a pneumatic or an electric motor actuated by the discharge pressure. Resilient bases, usually rubber-mounted, are used under fans when it is necessary to eliminate vibration or reduce noise.

Air conditioning is provided in certain spaces of each powerhouse for the protection of delicate electrical equipment and for human comfort. Because of variations in the heating and cooling requirements of the different parts of the powerhouse, the air-conditioning spaces usually are divided into three systems. One system serves the control room and its related spaces, another system serves the office areas, while the third system serves the public spaces, including the lobby and entrance vestibules. Other spaces, such as the telephone-equipment room, the laboratory, the assembly room, and the first-aid station, are served by one or more of the three systems. The control-room-system areas are grouped together because with their limited outside exposures and heavy heat losses from electrical equipment, they may require cooling at times when the other spaces require heating. The office system areas are served together because of similarity of exposure and thermal load, and the spaces served by the public spaces system are grouped together because of their proximity.

The individual air-conditioning systems may be used for ventilating, heating, and humidifying, or cooling and dehumidifying as required. Electric blast heaters are used for heating, and their operation is thermostatically regulated with auxiliary manual control. Humidification is provided by discharge of live steam generated by an electric immersion heater in an open pan in the plenum of each system. Cooling is by finned-type surface coolers, using chilled water in the coils. In the high-head plants where storage water is available at 55 F or less, lake water is used; in the other plants water is chilled by mechanical refrigeration.

Each of the air-conditioning systems can be placed on the heating, cooling, or ventilating cycles as desired and the control and operation of each is independent, that is, one system may be operating on the heating cycle while the other systems are on the cooling or ventilating cycles. Each system is complete with fan, electric blast heaters, humidifier, cooling coils, and filters, together with the manual, thermostatic, and other automatic control and safety devices necessary for its operation. Fans for air conditioning have two-speed motors or adjustable variable-speed motors, with the low speed used for the heating cycle and the high speed used for the cooling cycle.

The air filters are of the dry-filter type with filtering medium of paper, felted cotton, or felted glass fiber. Cell-type construction is used, and spare cells and filter media are provided at each plant.

In those plants having mechanical refrigeration, one water-chilling system serves all air-conditioning systems. The water-cooling equipment consists of one or more single-acting, multicylinder refrigerant compressors, a shell-and-tube-type water cooler, and a shell-and-tube-type condenser, all piped together and assembled as a unit. Welded joints are preferred for the steel refrigerant piping to minimize leakage, but tongue-and-groove-type flanged joints are used for connections to the compressor and refrigerant specialties. Piping smaller than 1 in. is usually extra-heavy copper tubing with sweated joints. The refrigerant is freon-12 which is nontoxic, noninflammable, and nonexplosive. Lake water, usually supplied by gravity, is used for condensing.

The air-conditioning system in each plant is automatic and in order to prevent excessive starting and stopping of the compressor a separate storage tank of chilled water is provided. Water is circulated from this tank by a small pump which responds to demands for cooling water. The chilled-water piping is insulated with cork, glass fiber, or mineral cork, ice-water thickness, and covered with a sealing tape and compound. Supply ducts are insulated with 1-in. boards of the same material and sealed with an airtight coating. Return-air ducts are not insulated. Fans, tanks, and plenums are insulated as required in the same manner as the ducts.

Machine-Shop Equipment. Each plant has a permanent machine shop, usually located just off the erection space on the generator floor. The size of the machine shops and the amount of equipment in them vary in the various plants according to the proximity of outside repair service. However, even in the isolated plants which have the maximum complement of machine tools, it is not intended that they be able to make all major repairs which may be necessary.

Experience has shown that the following list of machine tools is sufficient to take care of all the normal repair work in any plant:

1—8-in. engine lathe
1—9-in. bench lathe
1—24-in. shaper
1—30-in. upright drill
1—10-in. sensitive drill
1—9 × 9-in. power hack saw
1—12-in. grinder
1—portable pipe-threading machine, 1/2 in. to 2 in.
1—tool-post grinder

The larger machine tools are specified for medium-duty general-purpose precision work. In addition to these machines, each

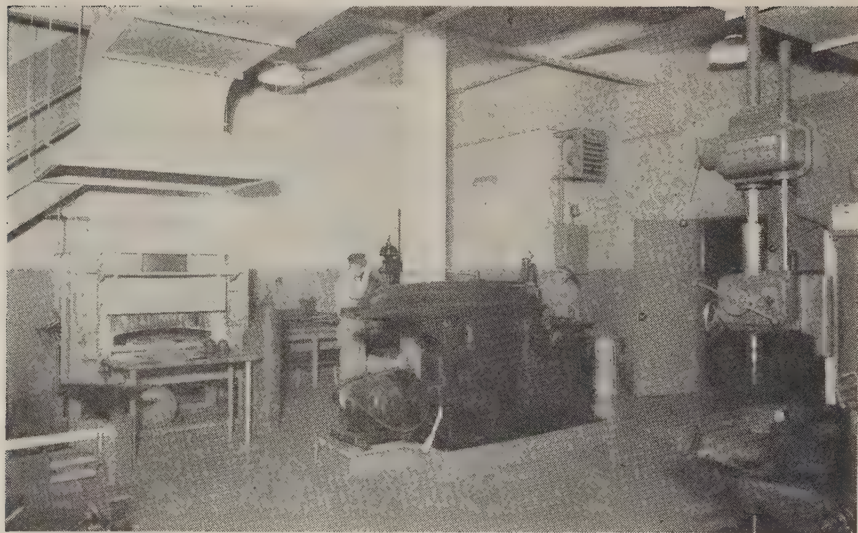

Fig. 8 Typical Machine Shop

plant is provided with a portable forge, blacksmith accessories, and a full complement of small tools. Fig. 8 shows a typical machine shop. The forge in this view has been superseded in later plants by a portable forge.

Acknowledgment

The author acknowledges valued assistance from members of the mechanical-engineering staff in preparing data for parts of this paper and for constructive criticism of the paper as a whole.

Discussion

Sabin Crocker.[4] The paper is principally devoted to describing the several piping systems required in hydroelectric plants and gives an excellent analysis of the basic considerations for good design of such systems. In general, the writer subscribes to and endorses the author's views on simplicity and neatness of arrangement with a view to improving appearance and at the same time making the purpose of the connections and valving readily apparent. Those interested in furthering a better understanding of the requisites of good design are indebted to the author for having taken the trouble to discuss the fundamental principles of piping layouts in a published paper where the information will be available to young engineers interested in improving their piping technique.

There is only one point noticed by the writer which seems to warrant further explanation. This relates to the following observation:

"In some of the larger-size mains, it has proved advantageous to use spiral-welded pipe. This is especially true in drainage lines embedded in concrete where the cost of steel or cast-iron pipe would not be justified."

The writer would like to inquire whether the foregoing statement should be construed to mean that the author considers it desirable to cast concrete solidly around drain pipes or other pipes conveying fluids. This question is prompted by the fact that cracking or settlement of concrete is apt to fracture any pipe solidly embedded therein, thus permitting the contents to escape or seep away to the general detriment of the piping system and plant. As a protection against trouble of this sort it is usual practice to provide a continuous protecting conduit of larger diameter around a pipe buried in concrete so that the outer conduit can fracture from any moderate settlement without damage to the pipe itself. The writer would like to inquire whether the author recommends dispensing with such outer conduit for drain lines and if so, under what conditions he considers this advisable.

[4] Consulting Engineer, Detroit, Mich. Mem. A.S.M.E.

J. B. Cutler.[5] The author's paper is very interesting and should be of value to hydroelectric-power engineers as it gives a general description of the type and arrangement of mechanical auxiliaries that the TVA engineers consider to be the most satisfactory after many years of operating experience in many plants.

It may be of some interest to mention a few of the mechanical features in our Safe Harbor Hydroelectric Station, which was first placed in operation in 1931, that are different or operate on a somewhat different basis from the TVA equipment referred to in the paper.

Piping and Color Code. The piping systems at Safe Harbor are identified by a color code. The greater portions of these lines, where visible, are painted solidly in the selected code color, and it is our opinion that with this arrangement, quick identification of a service line in an emergency is possible. An exception to this rule is made only for a small portion of some of these lines where, for architectural reasons, another color may be desirable.

All of the valves are identified by tags in a manner similar to that used by TVA, but at Safe Harbor only the valve code number is indicated. The service is readily identified by the solid code color of the pipe line, and the code numbers indicate the main power unit which is served by that line and the position of the valve therein. Odd numbers indicate the supply, and even numbers the return lines. A typical identification number for a valve in the raw-water system, serving power unit No. 3 and being the fifth valve in sequence in the supply line, would be 3-RW-5.

Control Valves. At Safe Harbor, when it is desired to operate a unit as a synchronous condenser, the elevation of water in the draft tube is automatically depressed by the admission of compressed air through the head cover of the turbine. An adjustable

[5] Mechanical Engineer, Pennsylvania Water & Power Company, Baltimore, Md. Mem. A.S.M.E.

type of reducing valve is used to maintain a constant pressure below the head cover, and although it is necessary to reset this valve for changes in the tail-water elevation greater than 8 or 10 ft, such changes are very infrequent. This system has operated satisfactorily since the units were first installed and no trouble has been encountered. Inspections are made, however, every 3 or 4 years, and as the result of some of these inspections it has been found desirable to overhaul the valve mechanisms.

The float-operating systems used by TVA would have the advantage of maintaining a constant level of water in the draft tubes regardless of the variations in tail-water elevations. It would be interesting to learn from the author whether any mechanical difficulties have ever been experienced in the operation of these floats.

Governor and Lubricating Oil. Similar to the arrangement used in the TVA plants, one grade of oil is used at Safe Harbor for both the governor and the unit guide and thrust bearings, and this oil is handled by one pipe, purifying and storage system. This arrangement has proved to be entirely satisfactory and the same quality of oil has been used since the date of initial operation of this plant. The viscosity of the oil at Safe Harbor is approximately 250 SSU at 100 F, which is somewhat lower than the viscosity value of the oil used in the TVA systems. The oil at Safe Harbor is purified about once a year using a centrifugal type of purifying unit for this operation. It would be interesting to learn how frequently it has been found desirable to purify the oil now being used for TVA.

Runner-Hub Oil. The oil used in the runner hubs of the Safe Harbor turbines has a viscosity of approximately 1600 SSU at 100 F. At 40 F the viscosity is very much higher or somewhere around 19,000. This grade of oil has proved to be satisfactory for this service both during the summer and winter seasons.

The author states that the oil used for this same purpose in the TVA stations has a viscosity of only about 1700 at 100 F, which, it will be noted, is a very much thinner grade than the oil selected for Safe Harbor. In view of this very wide difference in viscosity characteristics any further information the author may submit in regard to TVA operating experience with their oil should be interesting.

Author's Closure

Mr. Crocker's comment relative to the embedment of pipe in concrete is pertinent. The proper procedure under such circumstances should be determined by consultation between the piping designer and the concrete designer. The concrete designer has no wish to design a structure which will crack or settle, as the resulting damage to the dam or powerhouse holds as much or more risk as the damage which may be caused by a cracked pipe. The TVA has made every effort to design concrete structures with a minimum of cracking, and actual experience in the embedding of pipe solidly in this concrete has proved completely successful. However, if the concrete designer decides that unusual settlements may be expected, or if he must put in special contraction joints to care for possible cracking, then the piping designer must be aware of these facts in order that he may design his piping with such special protection as is necessary. It should be pointed out that if the embedded pipe has any bends in it, it would be very difficult and expensive to provide a continuous conduit of larger diameter around the pipe as Mr. Crocker suggests.

Mr. Cutler's comments on this paper are very interesting as the writer formerly was associated with him and helped develop some of the features he mentions about the Safe Harbor hydroelectric station.

The author believes the TVA method of painting piping systems is an improvement over the Safe Harbor method. The Authority's system, which paints all the various pipes with one color (aluminum) with identifying colors at fittings and flanges, provides a more pleasing appearance. This is obtained without sacrificing quick identification in case of emergency, because all valves are painted with the identifying color, and it is the valve, not the pipe, which controls an emergency.

Regarding the identifying valve tags, anything which aids the operator in establishing the function of a valve is to be desired. The naming of a system on the tag performs that function and adds very little to the cost of the tag. The Authority uses the same system of numbering valves on supply and drain lines as Safe Harbor.

We have found the float-operated valve very satisfactory for controlling the water level below the runner when it is operating as a synchronous condenser. It is preferred over the pressure-operated control in that it is much more sensitive and does not need to be reset. No operating difficulties have been experienced with the float-operated valve.

Under normal operating conditions the oil in the governor and lubricating-oil systems is purified once a year, at the time of the annual inspection.

The oil used in the rubber hub has a viscosity of 1700 at 100 F. The 40 F figure as given in the preprint was an error which slipped through all the checkings of this article.

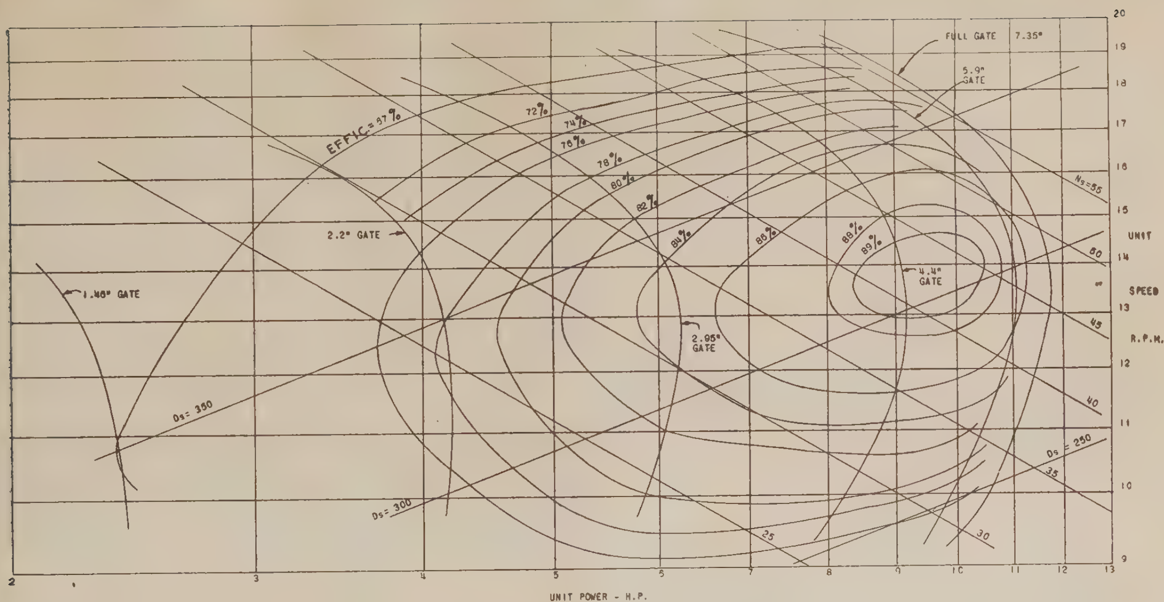

FIG. 1 PERFORMANCE DIAGRAM OF 100-IN-DISCHARGE-DIAM FRANCIS TURBINE
(Efficiencies from model test, not stepped up.)

A Better Method of Representing and Studying Water-Turbine Performance

BY R. A. SUTHERLAND,[1] NEW YORK, N. Y.

The author indicates that the contour form of turbine-efficiency diagram has advantages in giving a clearer picture of performance than the usual efficiency curves. An example illustrates that several different variables can be shown on such diagrams. Formulas are derived for finding the discharge diameters of runners and are illustrated by examples. A simple method of determining the mutual influence of turbine speed and water hammer during transient conditions is illustrated by an example. Finally, the operating stability of a water turbine supplying an independent load is brought forward for discussion.

INTRODUCTION

WATER-turbine performance is usually represented by "performance curves" consisting of a series of power-efficiency curves for different heads; the efficiencies shown may be either "expected" or "guaranteed," and the curves of course apply only to a selected size and speed of runner of a selected type. Such curves are made by "picking off" from test curves which give unit power and efficiency of a given size of runner at various gate openings in terms of "unit speed" or sometimes of "peripheral coefficient." Such curves have been reproduced elsewhere[2] and need not be repeated here.

The performance curves contain the minimum amount of information required by a purchaser and for many purposes are entirely adequate.

Another type of performance curve is in the form of an efficiency contour diagram, examples of which appear in several texts.[3]

The author has found that the contour type of diagram is well adapted for the following purposes:

1 The synthesis of available data on turbine performance.
2 Comparison of different runners.
3 Guide to operation.
4 Study of turbine behavior under transient conditions.

TYPICAL EFFICIENCY DIAGRAM

Fig. 1 represents an efficiency-contour performance diagram for the Norris and Hiwassee turbine model, the data being plotted from the published test diagrams.[2] The size assumed in Fig. 1 is 100 in. discharge diam, to facilitate ready conversion for other sizes. In addition to efficiency, gate openings, and specific speed,

[1] Hydraulic Engineer, Ebasco Services, Inc. Mem. A.S.M.E.
Contributed by the Hydraulic Division and presented at the Spring Meeting, Chattanooga, Tenn., April 1–3, 1946, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[2] "Francis-Turbine Installations of the Norris and Hiwassee Projects," by G. R. Rich and J. F. Roberts, Trans. A.S.M.E., vol. 64, 1942, p. 26.

[3] For example, "Water Power Engineering," third edition, by H. K. Barrows, McGraw-Hill Book Company, Inc., New York, N. Y., 1943, p. 239.

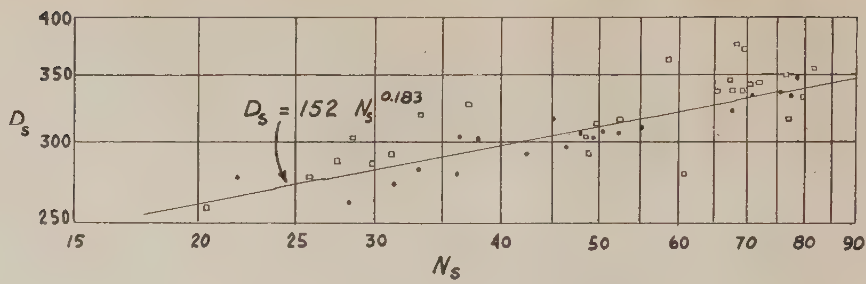

FIG. 2 FRANCIS RUNNERS; DIAMETER NUMBER IN TERMS OF N_s

the diagram also includes some lines representing "diameter number," which is a new concept as far as the author is aware. The expression for D_s is obtained by eliminating head from the basic homologous equations

$$\text{Hp} = A\,D^2 H^{3/2} \quad [1]$$

$$N = B\,\frac{H^{1/2}}{D} \quad [2]$$

where A and B are constants, just as the well-known expression for specific speed is obtained by eliminating D from these equations. By eliminating H from Equations [1] and [2], we obtain

$$D = \left(\frac{B}{A}\right)^{1/5} \frac{\text{Hp}^{1/5}}{N^{3/5}} = D_s\,\frac{\text{Hp}^{1/5}}{N^{3/5}} \quad [3]$$

In the present paper discharge diameters (in inches) are used throughout and values of the diameter number D_s are similarly applicable to discharge diameters (in inches). The symbol D_s has been represented by analogy with the universally used symbol for specific speed, but the author has avoided the use of the word "specific diameter," since D_s is not dimensionally a length, any more than N_s is dimensionally a speed.

The diagram shown in Fig. 1 can readily be converted by the principles of homology to apply to any selected size of runner of the same model type. The values of the efficiency should be stepped up to give an appropriate peak value, as determined by experience or by the use of the Moody formula applied to the original model. The 100-in-discharge-diam runner represented in Fig. 1 would be suitable for use at a normal head of 170 ft, and at a speed of 180 rpm. The unit speed would be 13.8 rpm, and the full-gate unit power at normal head would be 11.8 hp. Allowing a 5 per cent margin in power, the rated horsepower of such a turbine would be 25,000 hp. The D_s value at rated power would be

$$100 \times \frac{180^{3/5}}{25{,}000^{1/5}} = 298$$

SYNTHESIS OF DATA

By tabulating data from a number of turbine tests, expressions can be obtained for various factors in terms of N_s. In the limitations of space only D_s will be considered here. Fig. 2 shows values of D_s for Francis turbines in terms of N_s, each being for rated horsepower, which in the case of models has been assumed 5 per cent less than the full-gate power at rated best speed. The graph can be represented by the equation

$$D_s = 152\,N_s^{0.183} \quad [4]$$

Combining Equations [3] and [4] gives the following expression for discharge diameter

$$D = 152\,N_s^{0.183}\,\frac{\text{Hp}^{1/5}}{N^{3/5}} \quad [5]^4$$

This equation is believed to be more logical as a means of finding the discharge diameter of a Francis runner than others involving head, for the reason that the rated head can always be altered a few feet, while a synchronous speed must be used. The equation can, however, be expressed in these alternative forms

$$D = \frac{152}{N_s^{0.417}} \times \frac{\text{Hp}^{1/2}}{H^{3/4}} \quad [6]$$

$$D = 152\,N_s^{0.583}\,\frac{H^{1/2}}{N} \quad [7]$$

Table 1 shows a comparison of diameters obtained by Equation [5] with actual diameters of a number of turbines, and the results of other methods as noted are given for comparison.

If the specific speed of a Francis turbine be assumed to equal $\frac{632}{H^{1/2}}$, Equation [5] can be simplified (at the cost of some approximation) to the following form

$$D = \frac{1088\ \text{Hp}^{0.139}}{N^{0.722}} \quad [8]$$

Equations [6] and [7] could, of course, be similarly simplified. Equation [8] is represented in Fig. 3.

If a further simplifying assumption is made, namely, that rated load efficiency is 88 per cent, then Equation [8] reduces to the form

$$D = \frac{3.127}{H^{0.0413}}\,Q^{1/2} \quad [9]$$

Values of the coefficient of $Q^{1/2}$ vary only from 2.51 at 600-ft head to 2.81 at 40-ft head, so that Equation [9] leads to a simple "rule of thumb" for finding approximate discharge diameter of a turbine required to use a given amount of water. A simple rule of this type is often useful in preliminary investigation.

For the sake of completeness, equations for the diameters of fixed-blade-propeller and Kaplan runners will be given without description of the supporting data.

For fixed-blade-propeller runners

$$D = 79\,N_s^{0.35}\,\frac{\text{Hp}^{1/5}}{N^{3/5}} \quad [10]$$

For Kaplan runners

$$D = 445\,\frac{\text{Hp}^{1/5}}{N^{3/5}} \quad [11]$$

[4] This equation can be readily solved on a log-log slide rule without resort to tables or charts.

TABLE 1 DISCHARGE DIAMETER OF FRANCIS TURBINE

No.	Name	Head in feet	HP	Speed RPM	Spec. Speed N_s	Disch. dia D_2 inches	Diameter No. D_s	Discharge Diameter. Inches By Eqn. 5	By Creager & Justin	By Kent
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
1	Stevens Creek	27	3 125	75.	68.1	141	376	123.5	131.5	123
2	Amoskeog	46	7 500	112.5	81.5	126	359	119.	130	117
3	Mitchell	70	24 000	100	76.5	166	350	159.5	172.5	157
4	Upper Falls	64	14 250	105.8	69.8	153.5	372	137	145.5	136
5	Sherman Island	66	10 000	150	79.8	104	333	106	115	104.5
6	Muscle Shoals	95	30 000	100	58.4	179.5	362	158.5	160	168
7	Sturgeon Pool	105	6 300	277	65.2	66.7	338	64.5	67.5	64
8	Bridgewater	115	13 200	171.5	52.3	96	315	95	97	96.5
9	Rock Island	142	22 000	164	49.6	108	311	107.5	107.5	108.5
10	Shawinigan	145	49 000	138.3	60.9	126	279	146.5	151	145.5
11	Kern River Canyon	240	12 000	257	29.8	67	286	66	59.5	66.5
12	Queenston	305	63 500	187.5	37.0	129	328	116.5	109	118
13	Kerckhoff	315	15 000	360	33.2	64	320	57.5	53.5	59
14	Davis Bridge	350	20 000	360	31.0	62	293	60.5	53.5	59
15	Pit River No. 1	421.5	45 000	257	28.5	93	304	85.5	77	86.5
16	San Francisquito	515	22 000	428	25.8	56	277	53.5	47	53.5
17	Oak Grove	849	36 200	514	20.4	50	259	51	43.5	49.5
18	Norris	180	81 600	112.5	48.8	165	292	174.5	174	177
19	Bartlett	112	26 700	150	67.2	132	347	125	132.5	124
20	Wilson	92	40 400	100	70.5	179.5	342	174.5	186	173
21	Conowingo	89	54 000	81.8	69.5	213	339	208	221.5	206
22	Boulder	475	115 000	180	27.5	132	289	127.5	112.5	128
23	Hiwassee	190	80 000	120	48.1	165	305	167	166.5	169.5
24	Dnieprostroy	123	100 000	88.2	68.0	230	338	224	238	222
25	Beauharnois	80	53 000	75	72.0	228	345	219.5	234.5	217.5
26	Holtwood	62	20 000	94.7	77.0	149.5	316	159	172	157
27	Waterville	840	57 000	400	21.1	65.4	---	65	---	---
28	Diablo	327	90 700	171.5	37.2	127.5	---	131.5	---	---
29	Fifteen Mile Falls	180	54 200	138.5	48.9	143.5	---	142.5	---	---
30	Shipshaw	208	85 000	128.5	47.3	160	162	162	---	---
31	Chute a Caron	150	65 000	120	58.3	165	---	166	---	---
32	Bellows Falls	57	17 500	85.7	72.4	173	---	163	---	---
33	Parker	80	40 000	94.7	79.0	187	---	184	---	---
34	White Rapids	28	4 200	100	101.0	128	---	118.5	---	---

NOTE: Items 1 to 17 incl. from "Water Power Engineering" by Barrows, McGraw-Hill, NY, 1943, pages 424-7, omitting plants earlier than 1920.
Items 18 to 21 incl. from "Preliminary Selection of Hydraulic Turbines," W.L. Voorduin, T.V.A. 1941, p. 13, Fig. 2
Items 22 to 26 incl. Ditto p. 12, Fig. 1
Items 27 to 34 incl. Supplied by S. Morgan Smith Coy
Items in Col. 9 from "Hydro-electrical Handbook," Creager & Justin, Wiley, N.Y., 1927, p. 604, Fig. 395. An enlarged diagram was used.
Items in Col. 10 from "Mechanical Engineer's Handbook-Power," Kent, Wiley, N.Y., 1936, p.2-46, Fig. 12. An enlarged diagram was used.

Results obtained by these formulas are given in Tables 2 and 3, respectively, the former of which includes also the results of other methods.

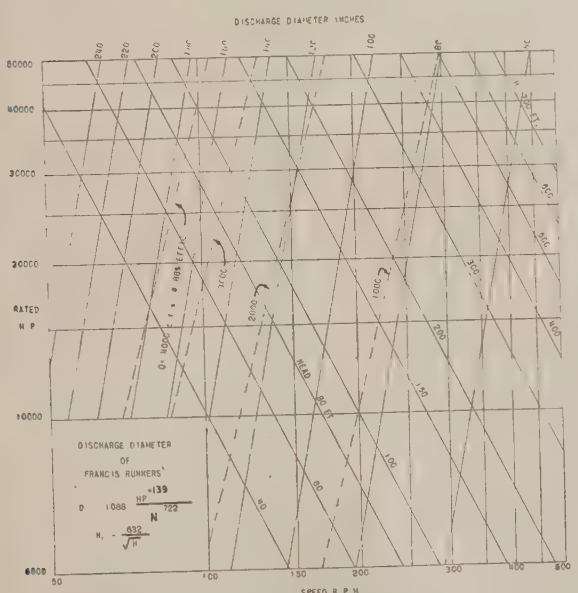

FIG. 3 DISCHARGE DIAMETER OF FRANCIS RUNNERS

The important bearing of sigma value on the best speed of propeller and Kaplan runners militates against obtaining accurate values of diameter for such runners by a formula which does not allow for the effect of the setting.

Equations such as [5], [10], and [11] are useful in the preliminary stages of powerhouse design.

COMPARISON OF DIFFERENT RUNNERS

If diagrams similar to Fig. 1 are available for different runners of approximately the same specific speed, the runners can be compared by superposition of the diagrams. The logarithmic scale enables the "spread" of any efficiency contour (such as 85 per cent or 90 per cent) to be compared as a measure of the percentage of full load over which a given efficiency will be reached or surpassed.

The contour diagram can be used equally as well as the more usual test diagram for "picking off" performance curves of two or more possible runners for given conditions. It can also be used to pick off directly the performance of a turbine at constant output under varying heads, a requirement which may apply to a plant which is used for peaking by being operated a limited number of hours per day at full generator capacity. The last-named pickoff is made by the use of an appropriate D_s line, since these lines represent constant output once the size and speed of the unit have been fixed. It is known that some runners have a better sustained output at low heads than others, and this is at once shown on the contour diagram by the full-gate line continuing to slope upward to the right, instead of bending back to the left. Such a runner might be advantageous in cases of peaking

TABLE 2 DIAMETER OF FIXED-BLADE-PROPELLER RUNNERS

No.	Name	Head in feet	HP	Speed RPM	Spec. Speed N_s	Disch Dia. D_2 Inches	Diameter No D_s	Discharge Diameter. Inches By Equation 10	By Creager & Justin	By Kent	By Voorduin
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
1	London	22.4	6 600	90	150	177	454	178	181	173.5	148
2	Crisp Co. Ga	27.8	5 500	100	116	148	418	148	---	152	79.5
3	Holyoke No. 8	27.8	5 500	128	149	118	381	141	141	135	131
4	Drummondville	30	6 000	138.5	152	133	450	135	137	132	---
5	Beebe Island	32.4	5 500	150	144	118	426	124.5	126.5	122	109
6	Slave Falls	34.3	13 800	94.7	134	187	432	190.5	196	190.5	141.5
7	Lewiston	35	7 000	138.5	136	141	463	134	137	133	---
8	Louisville	37.8	13 800	100	125	180	430	179.5	186	183	134
9	Proser	39.5	4 200	200	130	98	443	96	97	95	76
10	Wheeler	46.7	46 000	81.8	144	264	433	274	279	269	314.5
11	Chat Falls	52.5	27 600	125	145	195	457	192	194	187	254
12	Island Falls	56	14 000	163.6	126	139	439	136	139	137	---
13	Great Falls	56	28 000	138.5	151	189.	471	184	186	179	---
14	Cize Bolozon	57.4	11 400	187	126	121	431	121	123.5	121	---
15	La Gabelle	65.2	36 500	120	124	195	422	197.5	202	200	216
16	Seven Sisters	67	38 200	138.5	141	195	456	191	194	187.5	254
17	Flat Rock	7.5	503	80	144	118	471	113	114	110	---
18	Dixon	8	790	80	167	130	476	130	---	125	---
19	Matte	11.4	305	200	168	63	481	62.5	---	60	---
20	Troy	13.2	2 200	80	151	154	459	154	156.5	150.5	---
21	Wyman	15.5	2 460	107	174	126	437	138	---	134	---
22	Rheinfelden	17.1	2 220	107	144	131	463	127.5	129	124	---
23	Lilla Edet	21.4	11 430	62.5	145	236	436	243.5	248	238.5	---
24	Kachlet	25.2	7 350	75	114	177	398	184	---	191	---
25	Canadian	30.1	3 450	180	151	103	467	101	105	101	---
26	Forshuvud	34	10 050	94	119	167	404	174	---	180.5	---
27	Louisville	37	13 300	100	126	179	424	182	185.5	182	---
28	Manitoba	56.2	27 600	138.5	150	189	471	183.5	186.5	178.5	---
29	La Gabelle	60.2	29 600	120	124	188	425	189	194.5	200	---

NOTE: Items 1 to 16 inclusive from "Preliminary Selection of Hydraulic Turbines," W.L.Voorduin, T.V.A. 1941, pages 42, 43.
Nos. 4, 7, 12, 13, 14 are at rated head. Others in this group are at "critical head" as defined by Voorduin.
Items 17 to 29 inclusive from "Hydraulic Structures," A. Schoklitsch, A.S.M.E., New York, 1937, Vol 2, pages 946-7.
Items in Col. 9: See note on Col. 9, Table I. Items in Col. 10: See note on Col. 10, Table I.
Items in Col. 11: From source noted in second note on Table I, p. 43, Fig. 23.

TABLE 3 DIAMETER OF KAPLAN TURBINE RUNNERS

No.	Name	Head	HP	Speed RPM	Spec. Speed N_s	Disch. Dia. D_2 (inches)	Diameter Number D_s	Disch. diameter, inches by Equation 11
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
1	Klingmau	26.9	20,000	75	173	254	466	242
2	Laholm	28	16,000	83.3	165	216	442	217
3	McIndoes	29	5,100	150	159	120	439	124.5
4	Lanforsen	30.4	13,000	94	152	188	433	193
5	Guntersville	42	38,000	69.2	126	265	419	288
6	Dogern	33	33,000	75	168	276	458	268
7	Chickamauga	35.5	37,300	75	168	264	428	274
8	Aborrforsen	35.8	16,000	107	155	180	428	187
9	Swir	36.1	37,500	75	164	292	472	275
10	Ryburg	37.7	42,000	75	165	269	426	281
11	Pickwick	39.6	43,000	75	156	292	460	282
12	Jonage	42.6	6,000	214	152	101	443	100.5
13	Bonneville Units 1 & 2	60	60,000	75		280		301
14	Bonneville Units 3 to 10	65	74,000	75		280		314
15	Munkfors	50.5	15,000	167	151	147	464	141
16	Safe Harbor 25	55	42,000	100		220		236
17	" " 60	55	42,000	109.1		220		223
18	Cize Bolozon	53.5	10,000	187	130	118	431	122
19	Mixnitz	64.5	12,000	214	129	121	463	116
20	Semo Antschali	65.7	17,000	167	116	146	450	144.5
21	Santee Cooper	70	40,000	120	118	192	408	210
22	Groenvollfoss	75.5	18,400	214	130	133	466	127
23	Wettingen	76.3	10,700	214	98	114	445	114
24	Hojum	102	62,000	136.3	105	197	414	212
25	Shannon	106	33,000	167	89	161	434	165
26	Stromberg	13.8	480	250	226	61	488	56
27	Lanforsen	13.8	590	214	214	67	468	64
28	Kanyakeski	17.1	790	214	172.5	67	442	67.5
29	Siebenbrun	17.8	960	250	212	75	520	64
30	Tolfforsen	21.3	690	300	171	55	456	53.5
31	Lilla Edet	21.3	13,800	62.5	159	228	405	250
32	Gratewein	26.3	3,450	167	164.5	106	449	105
33	Mori	27.8	5,520	150	174.5	128	460	124
34	Hogforsen	33.3	4,030	214	170	94.5	449	94

Note: Items 1 to 12, 15 and 18 to 25, from "Preliminary Selection of Hydraulic Turbines," W. L. Voorduin, T.V.A., 1941, Figs.31 & 34.
Items 26 to 34 incl. from "Hydraulic Structures," A. Schocklitsch, A.S.M.E., New York, 1937, Vol. 2, pages 948, 949.
Items 13, 14, 16 and 17, by S. Morgan Smith Coy.

plants subject to occasional low head, in that a better firm power position could be demonstrated.

In the study of surge-tank stability, the rate of change of efficiency with head (power remaining constant) is one of the factors used.[5] This rate of change may be found directly from the contour curve by use of an appropriate D_s line.

Guide to Operation

Fig. 4 illustrates a typical contour diagram for a Kaplan turbine 132 in. in diam operating at 180 rpm.[6] In addition to efficiency curves, the diagram includes curves for gate opening and blade angle, and the D_s lines have been shown to represent horsepower. It is also readily possible on a large-scale diagram to show discharges and cavitation limits, but they are not included in Fig. 4, to avoid confusion. A diagram of this type is useful as a guide to operating in any desired manner, viz , to obtain highest efficiency, or to maintain a uniform discharge in the river, under varying operating heads.

Study of Turbine Behavior Under Transient Conditions

In so far as the author knows, previous studies of transient conditions in turbines have always been based upon some approximation as regards turbine discharge, and sometimes on a number of approximations. The contour diagram makes it readily possible to study correctly what happens in such cases. To illustrate the procedure, the turbine represented in Fig. 1 is taken.

Steady-load conditions are as follows:

Head, ft	170
Horsepower	25000
Speed, rpm	180
Gate, in.	6.2
Efficiency, per cent	89.3
WR^2, psf	8×10^6

[5] See "Hydraulic Stability," by A. W. F. McQueen, *The Engineering Journal*, vol. 16, 1933, p. 13.

[6] This diagram has actually been falsified from a turbine test and is thus illustrative only.

The effect of gate closure in 4 sec is studied in the following cases:

Case 1. Pressure rise neglected. This case is not attainable in practice, but might simulate conditions if a large surge tank were installed immediately upstream of the turbine.

Case 2. Turbine supplied by a penstock such that the effective characteristics of the water conduit are as follows:

Length, ft	300
Diameter, ft	13
Velocity of propagation, fps	3000

The unit is tripped off the line by relay operation.

Case 3. Similar to Case 2, but the load is reduced by load-limit control, the unit remaining on the line. (It may be noted in passing that according to previous methods the discharge history for this condition would be the same as in Case 2.)

In each case the effect of generator windage has been ignored for the sake of simplicity. It can readily be taken into account by adding another column to Table 4.

Fig. 5 shows the efficiency contours on a network of "unit speed" and "unit power" lines, and the gate lines shown are in accordance with the gate-time curve shown at the left in Fig. 6. The several data given have been selected for simplicity, but illustrate the procedure equally well as though an actual installation were considered. The complexity of the conditions that can be studied are limited only by the patience of the computer.

Pressure changes are determined by the graphical method, which has been described in many published articles.[7] Speed changes are determined by the formula

$$N_2^2 - N_1^2 = \frac{3{,}238{,}000 \times \text{Hp sec during interval}}{WR^2}$$

[7] One of the most recent being "Water-Hammer Problems in Connection With the Design of Hydroelectric Plants," by E. B. Strowger, Trans. A.S.M.E., vol. 67, 1945, p. 377.

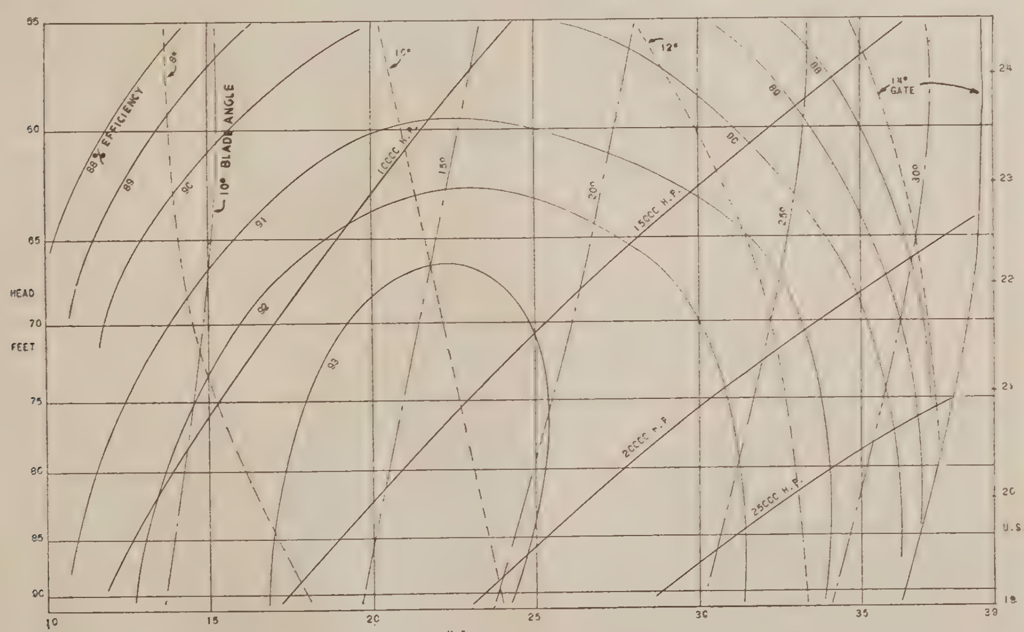

Fig. 4 Typical Performance Diagram of Kaplan Runner

TABLE 4 PART OF COMPUTATION OF CASE 2

Time	Gate	Trial Head	Trial Speed	Unit Speed	Unit Power	H.P.	Ave. excess H.P.	$N_2^2 - N_1^2$	N_2^2	N_2	Effic. %	Discharge required for H.P.	Discharge required for pressure	Remarks
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
0.00	6.2"	170	180	13.8	11.28	25,000			32,400	180	89.3	1450	1450	
0.20	6.0	175	186	14.06	11.13	25,780	25,390	2052	34,452	185.5	90.0	1439	1442	Speed wrong
		176	185.6	13.98	11.14	26,000	25,500	2062	34,462	185.6	90.0	1445	1442	O.K.
0.40	5.8	182	190.8	14.13	10.92	26,820	26,410	2138	36,600	191.3	90.7	1431	1416	Speed wrong Q wrong
		179	191.2	14.28	10.9	26,100	26,050	2108	36,570	191.2	90.6	1417	1420	O.K.
0.60	5.55	182	196.2	14.53	10.62	26,100	26,100	2110	38,680	196.6	91.1	1384	1390	Speed wrong
		183	196.6	14.53	10.62	26,300	26,200	2120	38,690	196.6	91.1	1390	1389	O.K.
0.80	5.25	186	202.0	14.80	10.26	26,020	26,160	2116	40,806	202.0	91.4	1348	1348	O.K.

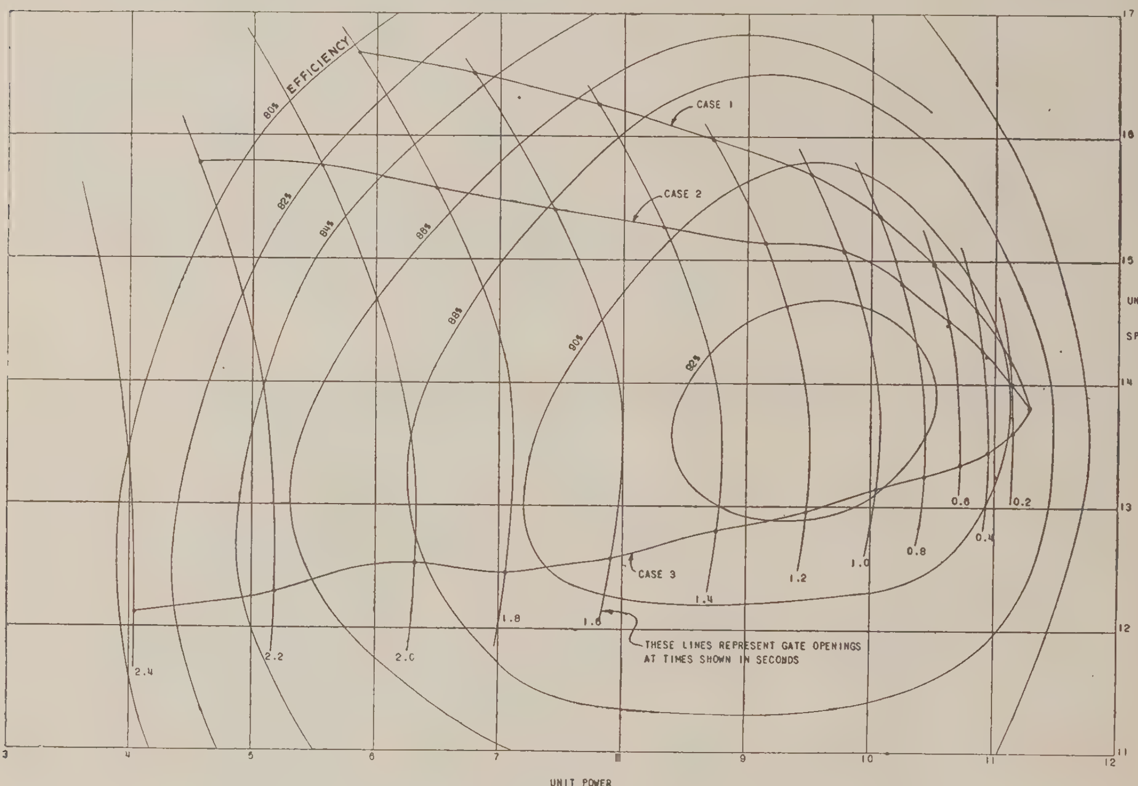

FIG. 5 PERFORMANCE DIAGRAM OF FRANCIS RUNNER FOR TRANSIENT CONDITIONS

Using the interval of 0.2 sec, and inserting the value given for WR^2, this expression reduces to

$$N_2^2 - N_1^2 = \frac{\text{Mean excess hp during interval}}{12.36}$$

The procedure used is a trial-and-error one, illustrated by Table 4, which shows part of the calculations for Case 2.

At each step a head and speed are assumed, and the discharges called for by the turbine diagram and by the pressure diagram are computed. If these discharges are not identical, the head and speed are modified in a direction which will be obvious from the first trial until substantial or exact agreement is obtained.

The results of the three cases cited as examples are shown in Fig. 7. The computation of pressure and discharge has been carried only as far as 1.8, 2.2, and 2.4 sec, respectively, for cases 1, 2, and 3, as indicated in Fig. 5. The remainder of the horsepower curves beyond the computed values can, however, be drawn in (shown broken in Fig. 7) with only slight error and the total speed rise thus determined.

Unfortunately, this method cannot generally be used for Kaplan turbines, because the blade movement does not in general keep time exactly with the gate movement, so that the efficiency-contour diagram (such as Fig. 4) does not hold for transient conditions.

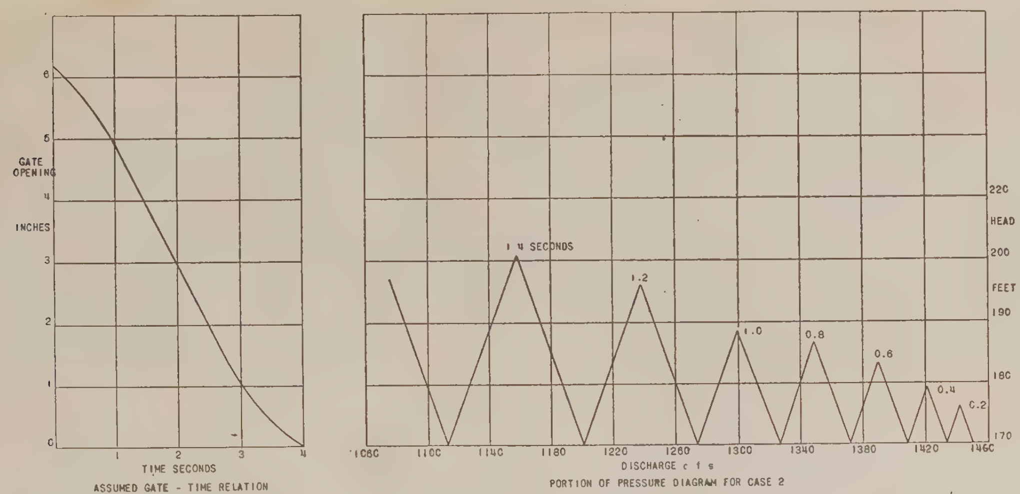

Fig. 6 Gate Relation and Typical Pressure Diagram for Francis Turbine Gate Closure

This method of studying transient conditions is obviously more refined than is generally necessary, but has been of inestimable value in studying one important and unusual installation, in which not only were numerous closure operations studied, but also governing stability of the unit when supplying independent loads. The computed behavior was closely checked by prototype tests.

Governing Stability

Operating stability of a hydroelectric plant generally requires the provision of an amount of WR^2 in the generator considerably in excess of that normally built in by the generator manufacturer. The leading generator makers have tables which show the "normal WR^2" for various sizes and speeds, and Fig. 8 has been prepared by the author to show the resulting regulation constant C. This diagram is useful as a preliminary guide to the amount of excess WR^2 required to give a desired value of C. The regulation of a hydroelectric unit connected to a system has been ably treated by numerous writers. In the exceptional case of a unit supplying an independent load, the stability may require study.

The moving water column contained in the penstock and draft tube represents a considerable amount of kinetic energy, which must be reduced or increased when the turbine gates close or open, respectively, as a result of any governor action, however slight. In the case of gate closure, part of the kinetic energy in the column is in effect transferred to the unit, which is already burdened with the excess energy which gave rise to the speed and gate change.

In the case of the gate-opening movement, the unit needs a quick supply of energy, which the water column is unable to furnish, being burdened with the necessity of increasing its own energy. Thus the water column always acts for an appreciable time in opposition to the governor, and it is easily seen that instability may occur if the water-column kinetic energy becomes too large in proportion to that of the unit.

The author has been able to observe only one case of this kind in practice, in which a unit was supplying a rheostat load and operated stably at small outputs, but became violently unstable as the output was increased. The ratio of the kinetic energy of

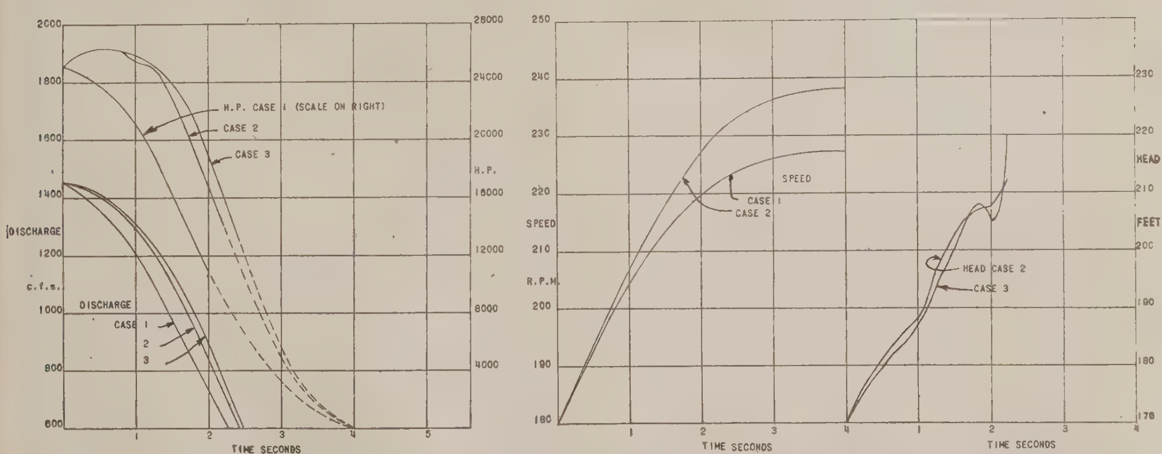

Fig. 7 Discharge, Horsepower, Speed, and Head During Gate Closure Under Different Conditions

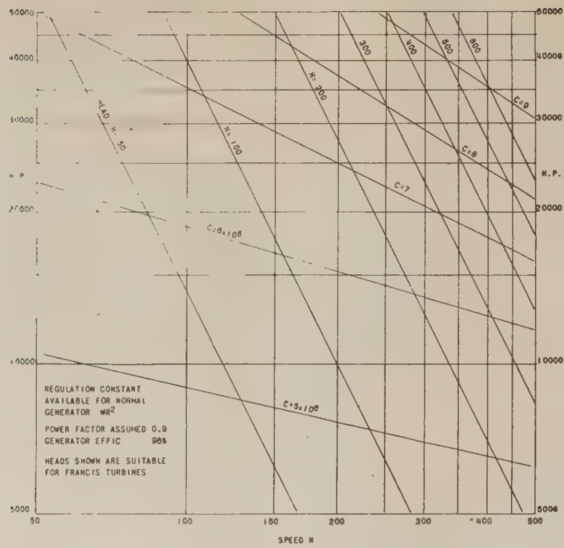

Fig. 8 Regulation Constant Available for Normal WR^2 of Generators

the water column to that of the unit, when instability occurred, was about 0.2. This ratio is obviously not the only factor involved, but is believed to be an important one, and the author suspects, from a study of other plants, that not a few units which now operate very sweetly on the line would be unstable if supplying a rheostat load.

The problem is analogous to but more complicated than the stability of surge tanks; and the author wishes at this time to bring the phenomenon into the open for discussion.

ACKNOWLEDGMENT

The author's thanks are due to Mr. L. F. Harza, who greatly stimulated his interest in turbine performance, and to Mr. George A. Jessop, for suggestions and data.

Discussion

L. F. Harza.[8] The author has urged the use of a valuable form of diagram in which to present hydraulic-turbine test data designed to display the characteristics of the turbine vividly and clearly. This form of turbine characteristic curves was used extensively in the first and subsequent editions of a text by Daniel W. Mead,[9] first published in 1908, on which the writer collaborated, except with a natural instead of logarithmic horizontal scale. The logarithmic scale adds little, if anything, to the diagram except the N_s lines become straight instead of curved.

Many years ago the writer developed standard platting paper for this type of curve with a sufficient range of horizontal and vertical scales so that any turbine then being built could be platted on this paper.[10] The N_s lines were curved because of natural scale. The position of the set of turbine curves on the sheet indicated the type of turbine. Thus a turbine of high unit speed and unit power would be platted, in the author's form of diagram, in the upper right-hand portion of the sheet, and a high head and therefore low-speed turbine in the lower left-hand portion of the sheet.

The author has added one new feature, namely, his D_s curve, whose usefulness the writer is not as yet prepared to evaluate without more experience in using it. The author refers to N_s and D_s characteristics, the former generally known as "specific speed," as not actually dimensional constants. In this the writer differs; N_s is the speed of a 1-hp turbine under 1 ft head, the turbine being of design homologous to the large turbine. Likewise D_s would appear to be the diameter of a homologous turbine which would run at 1 rpm and develop 1 hp. There would appear to be less value, however, in the mental conception of the meaning of this constant as compared with N_s.

The use of the relation curve, Fig. 2 of the paper, between N_s and D_s is not clear except as to the preliminary studies in laying out a hydroelectric station. It cannot be more than a general indication as to the diameter at the outlet of a turbine of an assumed N_s, sufficiently close for a first layout or assumption. It will be noted that the points from which this curve is projected are scattered widely. For example, at N_s = 30 the D_s may vary from about 260 to 300 because of the great difference in dimensions and turbine characteristics of different manufacturers, or the different types of turbines of essentially the same N_s of the same manufacture. For other than the first preliminary studies it is always necessary to use the dimensions of the individual type and size of turbine offered by the manufacturer. The D_s constant might be useful for preliminary studies before this more detailed information is available.

The author offers his Fig. 4 type of diagram as one suitable as a guide for operation, but in the opinion of the writer there is a much better form of diagram available for that purpose.[11] After the turbine is installed and its characteristics known by extrapolation or field test, there is no value in continuing the use of "unit speed" and "unit power" as scales of the operating diagram. The curve mentioned uses the operating head and actual horsepower of the prototype as co-ordinates. Thus at a glance

[8] President, Harza Engineering Company, Chicago, Ill. Mem. A.S.M.E.

[9] "Water Power Engineering," by Daniel W. Mead, second edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1915.

[10] "Further Means Suggested for Interpretation of Water-Turbine Test Data," by L. F. Harza, *Engineering News Record*, vol. 72, October 30, 1915, pp. 542–544.

[11] Ibid., Fig. 5, p. 544.

the operator can pick off the best horsepower at which to run the unit at each head. This should be carried one step further using gross head horizontally and kilowatts vertically, because these are the figures which the operator reads and which he can use without any conversion such as deduction for hydraulic losses to obtain net head, or converting horsepower to kilowatts.

The author presents some applications of his curves to the study of transient phenomena and governing stability which are well worth careful study and application.

Manufacturers generally do not put their test data into the "efficiency-contour" form of curves, or at least if they do, the customer never sees them. The information displayed to the engineer by this form of platting is so far superior to that of the usual form of characteristic curves submitted by a manufacturer, that the extra work of platting is well justified on the part of the purchaser's engineer.

J. D. McLaughlin.[12] The method given for studying the behavior of hydraulic turbine - generator units under transient conditions makes it possible as the author has outlined, to estimate with reasonable accuracy the power, discharge, speed, or head versus time relationships for unusual or unconventional hydroelectric installations.

The behavior under transient conditions of the hydroelectric unit installed at the Fort Peck Project during the war emergency required special analytical consideration in the manner outlined due to the fact that it was installed without the surge tank contemplated in the original design. Prediction of the behavior of the unit under anticipated operating conditions made possible the forecasting of operating regulations and the installation of special accessory features as means of adapting the unit to prevent excessive pressure surges in the penstock. The pressure-surge limitation was met by restricting the wicket-gate closure time to a minimum of approximately 42 sec. Overspeed of the unit, resulting from heavy load rejection at the line circuit breaker, was controlled within reasonable limits by automatic transfer of the generator output to a water rheostat which was preset to dissipate approximately 75 per cent of the load rejected.

The writer, being familiar with the installation, initial tests, and test conditions referred to under "Governing Stability," desires to elaborate upon the causes of instability of the Fort Peck unit when supplying a rheostat load, for the conditions that applied during the tests witnessed by the author. Observations upon which these comments are based were made during the operation of the unit without surge tanks over a period of approximately 3 years, during which time several series of tests were run to obtain specific data for use in designing a continuous-rating water rheostat. Throughout this period the behavior of the unit was one important consideration.

The resistivity of the river water, used in all rheostat tests, is relatively low, being approximately 600 ohms per cu cm. This factor had the effect of making a water rheostat located in the powerhouse tailrace extremely sensitive to fluctuations of the water surface.

The temporary, and what proved upon analysis to be unsatisfactory location of the water rheostat in the powerhouse tailrace during the initial test period undoubtedly intensified and perhaps initiated the instability noted in the operation of the unit supplying an independent load provided by the water rheostat. When operating in this manner, the load on the generator is determined by the depth of the electrode immersion. Surges and surface waves in the tailrace, whether due to the discharge from the adjoining tunnels or from the powerhouse draft tube, changed the depth of electrode immersion, and thus varied the generator load, by changing the water elevation of the tailrace.

An increase in tail-water elevation of 6 in. would produce an increase of approximately 22 per cent in a generator load of 20,000 kw, and would decrease, by the above increment, the head available to the unit. The former would increase the demand for kinetic energy from the water column, while the latter would decrease the energy available in the water column. Conversely, a decrease in tail-water elevation of 6 in. would decrease the load on the generator by approximately 28 per cent and would increase the energy available in the water column. In either case the governor would respond with a gate movement tending to restore the speed to normal, and in both cases would tend to "overadjust" the gate. Since the surging is periodic, it is believed that should the period achieve the proper magnitude with respect to the speed of the governor, serious operating conditions would result from this cause.

C. B. Spellman.[13] This paper arrives at certain new formulas for approximating the discharge diameters of various types of reaction-turbine runners. It appears to the writer that the application of these formulas for diameter is somewhat awkward, because they involve functions other than the basic engineering conditions and on this account the diameter number or the discharge diameter for a given project cannot be determined until one has first determined the specific speed or the speed in rpm. Also, it is suggested that a somewhat more definite idea of the runner size would be obtained if it were expressed as "throat diameter" rather than discharge diameter. The throat diameter seems more definitely fixed by considerations of flow and head, whereas the discharge diameter of a runner having a given throat diameter may vary depending upon the peculiarities of individual runner design. The ratio of discharge diameter to throat diameter might vary from as much as 1.20 down to even less than unity for very low-specific-speed runners.

For purposes of quick approximation of the throat diameter for a given set of basic conditions, the formula

$$\text{Diameter of throat} = \sqrt{\frac{68 \times \text{Horsepower}}{\text{Head}}}$$

has been developed. It will be noted that this formula involves only the two fundamental variables of horsepower and head.

We are indebted to R. B. Willi of the I. P. Morris Department for a very useful formula, involving only these same basic variables, for determining approximately the operating speed of vertical-shaft Francis or Kaplan turbines. This formula is

$$\text{Rpm} = \frac{C \times H^{5/4}}{\sqrt{\text{hp}}}$$

where C is equal to 632 for Francis turbines and 950 for Kaplan turbines.

As a check on the accuracy of these formulas, the writer has tabulated a comparison of the approximate diameter and speed with the actual values, for many of the I. P. Morris installations of both the Francis and Kaplan type (Tables 5 and 6 of this discussion). The tabulation indicates that the error in the formula for throat diameter averages less than 1 per cent and for speed less than 4 per cent. It must be admitted that in some few cases errors of the order of 20 to 25 per cent are encountered, but this occurs only when, for some reason, the specific speed selected for the design was at variance with the permissible value indicated by the "experience curve" of N_s versus head. When more accurate

[12] Engineer (Electrical), Hydroelectric Branch, Engineering Division, U. S. Engineer Office, Fort Peck, Mont.

[13] Assistant Supervising Engineer, I. P. Morris Department, Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

TABLE 5 I. P. MORRIS FRANCIS TURBINES; SIZE AND SPEED

Plant	Head	Hp	Rpm	$\frac{632H^{3/4}}{\sqrt{hp}}$	Rpm, per cent error	D_{th}	$\sqrt{\frac{68 hp}{H}}$	D_{th}, per cent error
Berlin, N. H.	83	20500	128.6	121.9	—5.2	126.0	129.7	+2.9
Sauzal	387	33600	300.0	303.5	+1.0	76.0	76.9	+1.1
Ixtapantongo	1028	39000	600.0	582.0	—3.0	48.6	50.75	+4.3
Boulder Dam	510	115000	180.0	201.2	+11.8	130.0	123.8	—4.9
Apalachia	360	53000	225.0	228.0	+1.1	103.0	100.1	—2.6
Yuba Narrows	235	13500	300.0	326.0	+8.8	66.0	62.5	—5.2
Marshall Ford	120	27000	144.0	140.0	—2.7	128.5	124.0	—3.6
Columbus	112	18000	150.0	162.5	+8.2	108.0	104.5	—3.1
New Kanawha	157	35000	150.0	150.0	0.0	125.0	123.0	—1.5
Hardy Dam	99	14800	163.6	163.6	0.0	103.5	101.0	—2.5
Carpenter	89	39500	94.7	97.8	+3.1	173.0	174.0	+0.4
Cobble Mountain	420	19200	450.0	426.0	—5.3	54.0	55.7	+3.2
Cobble Mountain	420	8550	600.0	638.0	+6.1	36.0	37.2	+3.4
Morony	82.5	31000	81.8	98.5	+20.5	173.0	160.0	+7.6
Lower Tallassee	88	36000	100.0	96.1	—4.9	165.2	166.8	+0.9
Colton	270	15700	360.0	338.0	—6.2	62.5	63.0	+0.8
Cadyville	156	4300	450.0	426.0	—5.4	40.0	43.3	+8.2
Upper Tallassee	55	25000	80.0	81.6	+2.0	175.0	176.0	+0.5
Lake Chelan	377	34000	300.0	295.0	—1.6	75.5	78.2	+3.6
Conowingo	89	54000	81.8	78.9	—3.6	194.0	203.0	+4.6
Chapala	233	8000	360.0	424.0	+17.6	51.5	48.3	—4.1
Norwood	70	31100	90.0	86.5	—3.9	170.5	174.0	+2.0
Cherokee Bluff	145	45000	120.0	124.7	+3.9	147.0	145.3	—1.1
Cheat Haven	81.5	18000	133.3	127.1	—4.6	121.0	122.8	+1.3
Rumford Falls	97	10000	200.0	195.5	—2.2	85.0	83.6	—1.4
Cutler	124	21500	150.0	159.7	+6.3	111.0	108.6	—2.1
Exchequer Dam	300	24500	257.0	292.0	+13.5	73.6	74.5	+1.2
Wallenpaupack	330	28500	300.0	292.0	—2.6	73.6	76.6	+4.1
Soft Maple	121.5	10500	225.0	227.5	+1.0	78.6	76.5	—2.6
Lighthouse Hill	62	5250	150.0	192.8	+28.5	83.5	76.0	—9.0
Dix River	220	9850	300.0	369.0	+23.0	58.8	55.1	—6.1
St. Croix	48	4550	150.0	172.8	+15.1	82.0	80.2	—2.2
Amoskeag	46	7500	112.5	130.0	+15.2	110.0	91.1	—17.1
Holtwood	62	20000	94.7	99.0	+4.4	149.5	148.2	—0.9
				Avg	+3.7		Avg	—0.56

TABLE 6 I. P. MORRIS KAPLAN TURBINES; SIZE AND SPEED

Plant	Head	Hp	Rpm	$\frac{950 H^{3/4}}{\sqrt{hp}}$	Rpm, per cent error	D_{th}	$\sqrt{\frac{68 hp}{H}}$	D_{th}, per cent error
Fort Loudoun	65	50000	105.8	97.5	—7.6	222.0	229.0	+3.0
Watts Bar	52	42000	94.7	90.1	—4.8	234.0	234.5	+0.2
Safe Harbor	55	42500	100.0	93.9	—6.1	220.5	229.5	+4.0
High Point	70	10000	225.0	229.0	+1.8	101.0	98.5	—2.5
Chickamauga	36	36000	75.0	74.1	—1.3	264.0	261.0	—1.2
Buzzard's Roost	60	7400	240.0	241.0	+0.6	92.0	91.5	—0.5
Mattaceunk	39	5900	171.4	195.1	+13.9	109.5	101.5	—7.3
				Avg	—0.5		Avg	—0.61

data are required, manufacturers' recommendations, of course, should be obtained.

The principal application which I. P. Morris has found for the contour type of diagram has been in the Kaplan field, where it greatly facilitates the preparation of expected-performance curves for proposed prototype turbines. Without the contour diagram the preparation of such curves is very laborious, as the basic test data comprise a series of tests for various fixed-blade positions. Thus a performance curve has to be derived for each blade position, and the envelope of these individual performance curves drawn to indicate the performance obtainable with automatic adjustability of the runner blades. The reason for this is that each different blade position, in effect, makes a different runner is so far as performance is concerned. This, of course, does not hold true in the Francis field where the blades are immovable. Hence the Francis performance curve is readily obtainable from a single set of model test readings. For this reason there has not been the same incentive to use contour diagrams for the Francis models as there has been for the Kaplan models.

E. B. Strowger.[14] In discussing turbine behavior under transient conditions, the author makes use of the contour form of turbine-efficiency diagram for showing the performance relations in terms of power, speed, and gate opening. On these diagrams the speed is shown in terms of unit speed and the power in terms of unit power rather than in terms of rpm and power output of the unit under normal head conditions. The writer prefers to show the transient performance in terms of the normal head performance rather than reduce the data to 1 ft head. In this connection he wishes to call attention to the joint paper he wrote with S. L. Kerr in 1926, on this subject,[15] where the step-by-step or arithmetic integration method was first applied to the problem and the solution was first made by a rational method rather than by rough approximation. He also wishes to call attention to a discussion of Arnold Pfau's paper which was published in 1930.[16] The author's method is the same as that shown in these publications except that he has substituted the contour diagram for the runner ϕ curves, and the graphical method of water-hammer determination for the arithmetic-integration method. These substitutions do not change the basic character of the method of computation.

Case 3, cited by the author, assumes that during load rejection the unit remains on the line. Consequently, with the unit connected to a large system, the speed of the unit will remain sensibly constant. However, because the head is increased, due to water hammer, the unit speed decreases as shown by the curve, marked Case 3 in the author's Fig. 5. The author states that according to previous methods the discharge history for this condition would be the same as in Case 2. The writer believes that the method shown in the references cited would give a discharge history which would not be the same as the author's Case No. 2 but would agree with the author's results for Case 3.

It should be pointed out perhaps that the time-horsepower curves for Cases 1 and 2, as shown in the author's Fig. 7, should be modified in the dotted portions to show that the power output becomes negative near the end of the governor stroke due to the overspeed, forcing the governor to close the gates below the speed-no-load point.

[14] Hydraulic Engineer, Buffalo Niagara Electric Corporation, Buffalo, N. Y. Mem. A.S.M.E.

[15] "Speed Changes of Hydraulic Turbines for Sudden Changes of Load," by E. B. Strowger and S. L. Kerr, Trans. A.S.M.E., 1926.

[16] "Mechanics of Hydraulic Turbine Pressure Regulation," by Arnold Pfau, Trans. A.S.M.E., vol. 52, 1930; discussion, HYD-52-4.

Author's Closure

Mr. Harza questions the value of logarithmic plotting of turbine test data. The main advantage, in the author's experience, is the ease of transferring from one size of unit to another. In such case the curves need not be replotted but only the scales need be shifted. Disadvantages are that the portion of the diagram near full gate has the smallest divisions and that the diagram cannot well be extended down to small gate openings. Mr. Harza's definition of "specific speed" is frequently used but nevertheless is believed to be inaccurate, for the insertion of dimensional values will show that it is not a speed. One at least of the makers recognizes this by using the more correct term "characteristic speed number." Similarly the author's term D_s could be called the characteristic diameter number. Its value, as Mr. Harza indicates, lies in giving a first approximation to the size of runner to meet given conditions, just as the N_s parameter is used in finding a suitable speed, and it is believed that the author's Tables 1, 2, and 3 indicate that it serves this purpose. The author's formulas were developed in self-defense against the insistent demand of the structural men for dimensions, in the early planning of hydroelectric projects. One of the engineers of a large public body has devoted a whole book to this and allied questions (Table 1, second footnote). It is true as Mr. Harza points out, that there is quite a spread in the experience values of D_s, but it is believed that this is due largely to the turbine makers' use of developed models for heads or speeds for which they are not ideally suited. It is the author's belief, which unfortunately may never be proved or disproved, that if all turbines were specially designed for optimum operation at their rated head and speed, the spread would be very small.

Mr. McLaughlin's discussion of the Fort Peck installation is of particular interest to the author, who took part in the tests made prior to regular operation. It is hoped that Mr. McLaughlin can be prevailed upon to publish the results of the later tests which he indicates have been made, for this wartime installation was of unusual interest from an hydraulic point of view and reflects considerable initiative and courage on the part of the U. S. Engineer Department. The author is unable to agree with Mr. McLaughlin's suggestion that "the location of the water rheostat in the tailrace perhaps initiated the instability noted." He will, however, agree that it complicated the phenomenon. In a certain report[17] are shown oscillograms obtained during operation both with a tank rheostat (Plate 14, Test D-1) and with the tailrace rheostat (Plate 16, Test H-1) and each shows the same characteristic instability, the period being of the order of 14 seconds. Actually the rate of pressure rise in the case of the tank rheostat appears to be appreciably sharper than in the case of the tailrace rheostat. Instability of this type is comparatively little known in the United States, due to interconnection of plants, but appears to be more familiar to European engineers, and in fact has been ably discussed by Daniel Gaden in a recent text.[18]

Mr. Spellman's contribution is distinctly useful and practical and his formulas have the advantage of being easily remembered. His suggestion that "a more definite idea of the runner size would be obtained if it were expressed as throat diameter rather than as discharge diameter" is no doubt valid, and for a maker's engineer the throat diameter is no doubt the more fundamental dimension. The author's use of his formulas was mainly for the limited purpose of determining in the first place the dimensions of draft tubes, etc., as a means of fixing approximate powerhouse dimensions. This he did by means of various empirical relationships referred to discharge diameter and specific speed. The same purpose could be served if such data were available in terms of throat diameter, which could then be obtained by Mr. Spellman's formulas. The author is interested to note that Mr. Spellman's company values the facility of the contour type of diagram, in the case of Kaplan turbines. It is a fact that the contour diagram is not only capable of containing a surprising amount of information in one diagram without confusion, but is very revealing as to the suitability of a runner for a given head and speed and for this reason has been given the apt name, by Mr. Harza, of the "goldfish bowl" diagram.

Mr. Strowger's eminence in his field makes his contribution especially appreciated by the author, who is glad to state that he regards the first paper cited by Mr. Strowger[15] as the original "gospel" on the subject. Mr. Strowger believes that the methods shown in the references cited by him would give results which would agree with the author's Case 2 and Case 3. Mr. Strowger's Fig. 5 (loc. cit.) gives the gate-time relationship and one turbine efficiency curve, which is presumably taken at the rated head and speed, and therefore at a constant unit speed. It is not apparent to the author that efficiencies taken at a constant unit speed could correctly be applied to the behavior of a turbine whose actual unit speed, on account of the pressure and speed changes, is varying throughout the phenomenon, although no doubt the error involved might be negligible for many purposes. The second reference cited by Mr. Strowger[16] contains a very excellent exposition of his procedure of determining the change in head and speed during the phenomenon of a change in load. It takes into account both the changes in head and speed during the transient as shown by his Tables 2, 3, and 4. Two complete sets of computations are required, the second applying trial and error principles to the first.

The author's procedure seems to be more direct in that each point in the computation is arrived at by trial and error to satisfy simultaneously two sets of conditions, viz., the water-hammer conditions and the turbine characteristics.The accuracy is limited only by the correctness of the physical data and the patience of the computer. The author feels considerable confidence in this method for it was successfully used to predict pressures in the Fort Peck installation under various conditions, and was found to give quite appreciable differences in pressure with the same gate closure, depending on the amount of water rheostat load thrown on the generator during the gate-closing operation. In general, the pressure rise increased as the rheostat load used decreased, although this would not necessarily hold true for other operating heads.

Referring to the author's Fig. 5, if the operating head were different, the starting point of Cases 1, 2, and 3 would likewise be different from that shown, and the lines marked Case 1, Case 2, etc., would cut across the efficiency contours in a different manner and other results than those just mentioned might occur. The Fort Peck tests referred to by Mr. McLaughlin required turbine closures with a water conduit about a mile long and no surge tank, and the element of risk involved warranted the most painstaking investigation prior to embarking on such tests. The author's method also successfully predicted hydraulic instability on rheostat load. This was possible because the slow governor time considered rendered the behavior of the unit amenable to computation by reason of the governor compensation effect being negligible. The author believes that an additional advantage of his procedure lies in its giving a graphical picture at each instant, and this is particularly apparent in studying stability, where what might be called the "history line" (like the lines marked Case 1, etc., on the author's Fig. 5) wanders around the diagram somewhat in the form of a lemniscate.

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[17] Field Pressure Measurement, Fort Peck Powerhouse Penstock, Technical Memorandum no. 206–1, U. S. Waterways Experiment Station, May 15, 1944.

[18] "Considérations sur le Problème de la Stabilité—Contribution à l'Etude des Régulateurs de Vitesse," Lausanne, Editions La Concorde, 1945.

Mr. Strowger correctly points out that the "tail end" of the author's horsepower curve, Fig. 7, should dip down to a negative value. The excuse for not doing so is that a good runner of the specific speed illustrated would run at speed-no-load with a very small gate (in the case of Fort Peck about 0.04), so that this negative dip would be negligible, and secondly, that makers generally fail to test runners at very small gate openings so that data would ordinarily be unavailable to compute this negative dip, even though it is known to exist. In certain cases, however, it is recognized that this negative power may be of considerable importance, and in any case the existence of the negative portion of the horsepower-time curve would make the speed curve peak earlier, viz., at the instant the horsepower changes from positive to negative.

FIG. 1 GENERAL VIEW OF THE NANTAHALA POWERHOUSE

Nantahala Turbine

BY J. P. GROWDON,[1] R. V. TERRY,[2] AND H. H. GNUSE, JR.[3]

The Nantahala plant of the Nantahala Power and Light Company is located near Franklin, N. C., and was designed by the Aluminum Company of America. The 60,000-hp hydraulic turbine operates under 925 ft rated net head at 450 rpm. The static head on the plant of 1008 ft is the highest for any reaction-type turbine installed in the United States. The design and construction of this turbine and butterfly valve are described, followed by results of the field tests. Operation and maintenance work undertaken since the unit was installed are frankly discussed. The field tests showed an efficiency of 93.7 per cent. Considerable stainless steel was used in the original installation and its use has been greatly extended since the unit was placed in operation.

INTRODUCTION

THE Nantahala hydroelectric development is owned and operated by the Nantahala Power and Light Company, a subsidiary of the Aluminum Company of America. It is situated on the Nantahala River in southwestern North Carolina, where the steep slopes permit a high-head development in a relatively short reach of river. It was completed in July, 1942, and has been in successful operation since that date.

The acquisition and design of this development has been carried on by the owner and its parent company since 1911. In this period of 35 years, many different types of development have been proposed and studied. As finally designed and built, the Nantahala hydroelectric development consists of an earth-faced rock-fill dam, 250 ft high, creating a reservoir with 125,000 acre-ft of usable storage. The water from the reservoir is taken to the powerhouse through an unlined pressure tunnel, approximately 5.3 miles long, and a penstock 1391 ft long. A differential surge tank is located 1991 ft from the turbine. The powerhouse, shown in Fig. 1, contains a single hydroelectric unit, consisting of a 60,000-hp reaction turbine, operating under a maximum static head of 1008 ft, directly connected to a 3-phase 60-cycle generator, with the necessary transformers and auxiliary equipment. The power output is fed into a transmission system so that it is not required to operate the unit when for any reason transmission is interrupted.

A great deal of study was given to the problem of selecting a turbine which would combine the following characteristics: Low cost per horsepower; maximum efficiency; satisfactory and reliable operation.

At this head, either an impulse or reaction turbine appeared feasible. For the total capacity at least two impulse turbines would be required, with an expected efficiency of 88 per cent.

[1] Chief Hydraulic Engineer, Aluminum Company of America, Pittsburgh, Pa. Mem. A.S.M.E.

[2] Assistant Chief Engineer, Newport News Shipbuilding & Dry Dock Company, Newport News, Va. Fellow A.S.M.E.

[3] Chief Electrical Engineer, Nantahala Power & Light Co., Franklin, N. C.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of authors and not of the Society.

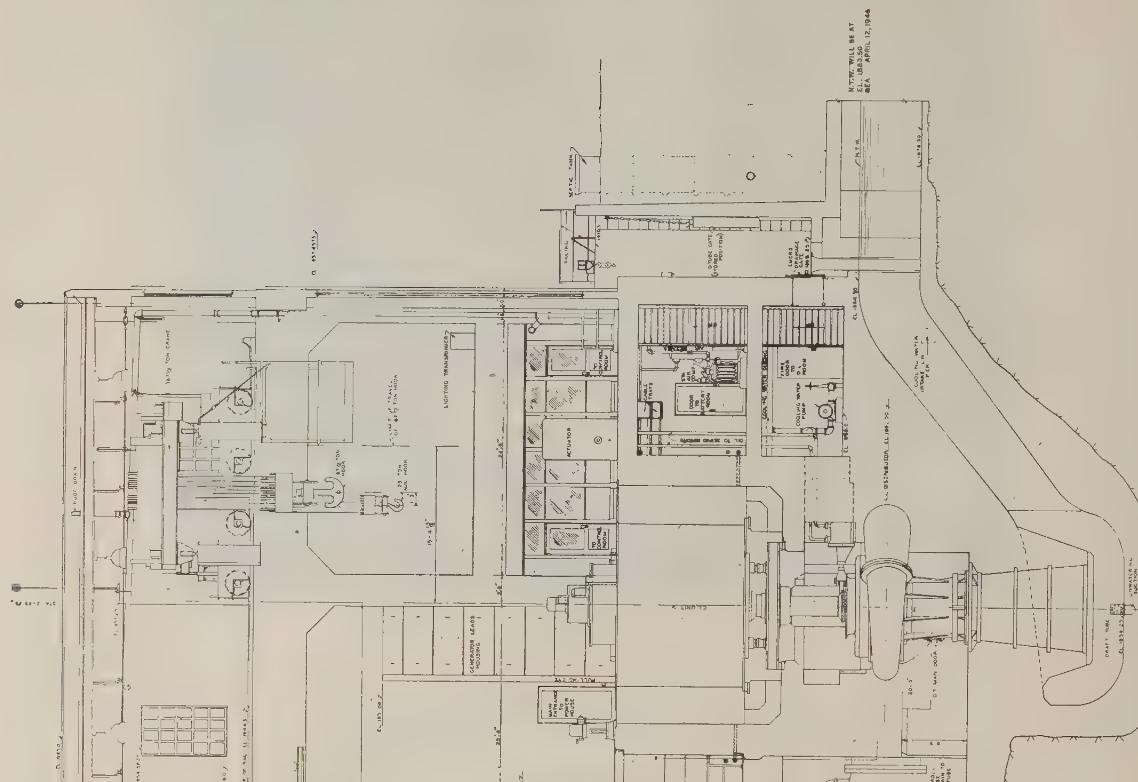

FIG. 2 LONGITUDINAL SECTION THROUGH POWERHOUSE

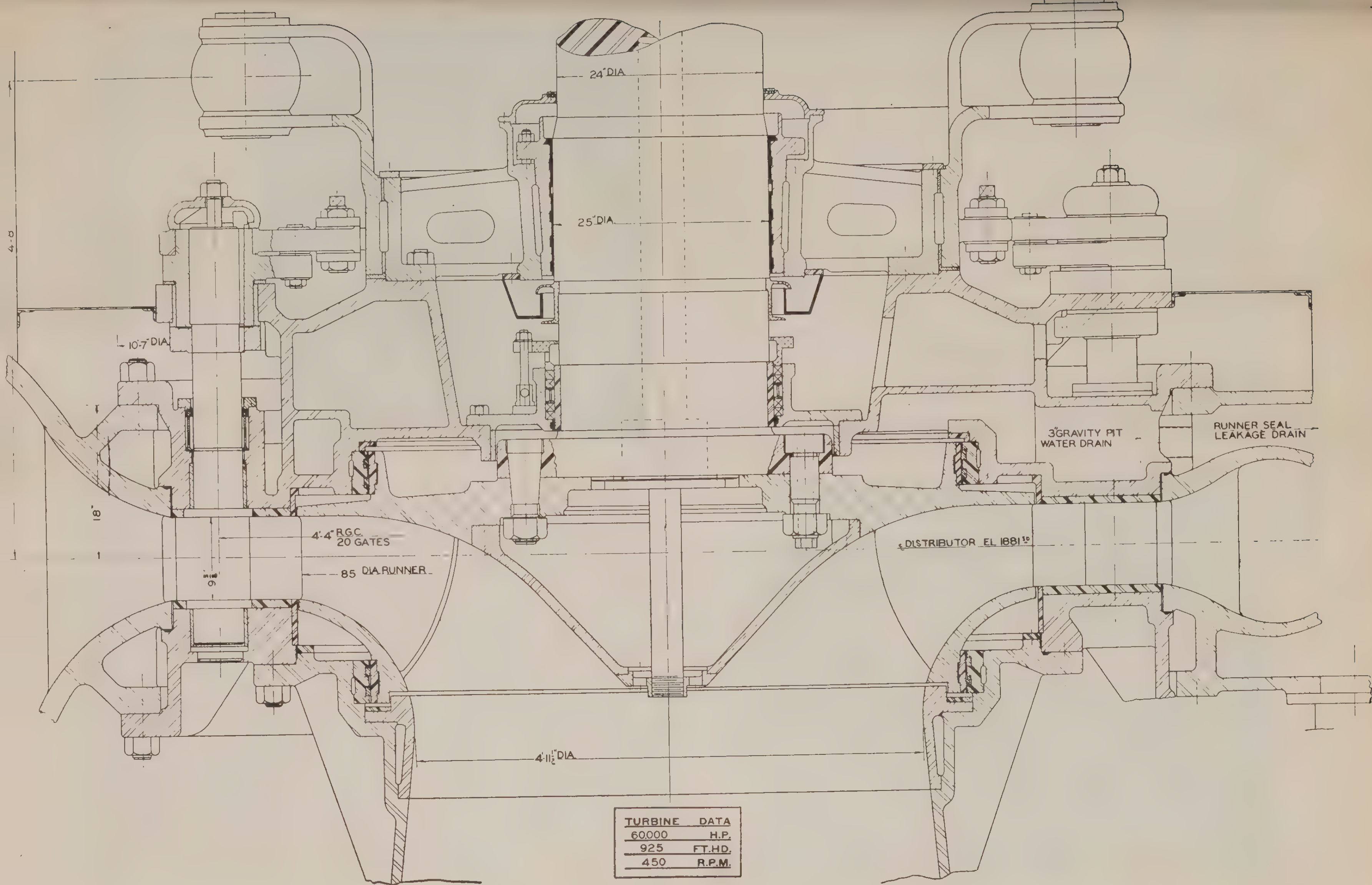

Fig. 3 Sectional View of the Turbine

Since the power requirements did not necessitate operation at low loads, and since the large reservoir provides an opportunity for periodic inspection and repairs, without wasting water, the two units are no more useful than a single unit and would cost considerably more.

A reaction turbine of this capacity, having an efficiency of more than 90 per cent, appeared to be feasible. Due to the permissible lower turbine setting, it would regain a few feet of head which the impulse turbine could not utilize. A single-unit reaction turbine, with its direct-connected generator, would cost considerably less than the two impulse units previously mentioned. It was realized that the head is higher than any other reaction turbine in America, and that reaction turbines under somewhat less head had developed many operating difficulties.

The parent company had had some experience with high-head reaction turbines and believed that as a result of this experience and the increased knowledge of the turbine designers, it would be possible to design and build a reaction turbine which would meet the requirements of the Nantahala hydroelectric development better than any other type and which would be free from serious operating difficulties.

Several turbine manufacturers agreed with this conclusion, and after consideration of all turbine builders' proposals, the contract was awarded to the Newport News Shipbuilding & Dry Dock Company for the design and manufacture of a 60,000-hp (925-ft head), 450-rpm vertical reaction turbine, with all essential auxiliaries, operating under a maximum static head of 1008 ft, and an 8-ft-diam butterfly valve to be located immediately above the scroll case.

Nantahala Turbine

The turbine selected for the Nantahala plant is of the vertical-shaft single-runner Francis type, rated 60,000 hp under a net head of 925 ft, operating at a speed of 450 rpm. This gives a specific speed of 21.6. It is designed for a maximum net head of 1000 ft, the maximum static head on the plant being 1008 ft. This is believed to be the highest-head reaction-type turbine installed in the United States and to be the highest-powered extremely high-head reaction turbine in the world.

The Francis type was selected in preference to an impulse type after giving due consideration to several factors. The efficiency of the reaction-type turbine is several per cent higher (about 4.5 per cent) than that of the impulse type. The speed is considerably higher, resulting in a much smaller and less expensive generator. The unit is sufficiently large so that reasonable clearances could be employed and yet maintain relatively small leakage losses. The disadvantages are that load cannot be changed as rapidly as with an impulse turbine and that there is some element of danger in operating the unit unwatered as a synchronous condenser.

Fig. 2 shows a longitudinal section through the powerhouse. The turbine setting includes a hydraulically operated butterfly valve between the turbine-casing inlet pipe and the penstock, and a cast-steel spiral casing in 3 sections with 20 vanes, one opposite each wicket gate. The vertical part of the draft tube is of the straight conical type, the upper part being of cast steel and the lower part of steel plate. The upper part includes a removable section just below the runner. This permits removing the runner and throat ring from below and gives access for repair or renewal of the seal rings and other work on the turbine without disturbing the crown plate and other parts supported by the crown plate. Downstream of the conical part of the draft tube there is a collecting chamber which discharges diagonally upward and then around a bend to the tailrace. There is one pier in the diagonal portion and bend of the discharge passage. The discharge passage is vented upstream of the bend and near the wall of the powerhouse.

Runner and Seal Rings. Fig. 3 shows a sectional view of the turbine. The runner is of the conventional Francis type of cast steel in one piece, 85 in. nominal diam, with 21 vanes, arranged for a bolted connection to the main shaft. The bolted connection is arranged for easy removal of the runner. There are ten tapered bolts used primarily to take shear and four loose-fitting straight bolts to carry the hydraulic thrust and weight of the runner.

The entrance edges of the vanes are sloped circumferentially and a large radial gap of about 6 in. is left between the runner vanes and wicket gates so as to result in smooth flow from the gates into the runner. This arrangement has proved highly satisfactory and cannot be too greatly emphasized.

The runner carries two straight steel seal rings shrunk on, one on the crown and one on the band. The two rings are of equal diameter to provide good hydraulic balance and at the same time are interchangeable. The rings are placed at as small a diameter as possible so that a small running clearance could be used and the leakage area reduced. A radial clearance of 0.015 in. is employed. The clearances at the periphery of the runner are purposely left large to preserve hydraulic balance. The seal rings are supplied with 80 gpm of clear water through a check valve, piping, and radial holes between the inserts in the stationary seal rings.

Provision is made for checking the seal clearances periodically from above and below the runner, with the turbine shut down. Leakage through the top seal is carried out through the crown plate and a pipe to a point above tail water where the amount of leakage may be observed at any time during operation.

The runner was dynamically balanced and the shaft-connection holes drilled and reamed before shipment to the generator manufacturer. A cast-steel fairwater is bolted to the bottom of the runner. Fig. 4 shows a shop view of the runner and spare runner.

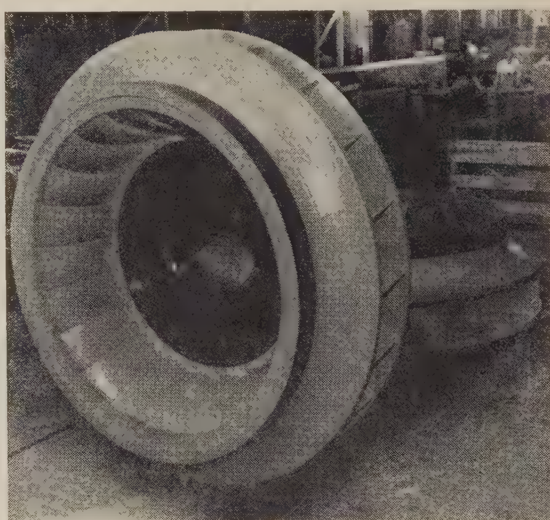

Fig. 4 Shop View of Runner and Spare Runner

Shaft. The usually separate turbine and generator shafts were combined and furnished in one piece by the generator manufacturer. This is practicable since the runner is removed from below. The shaft has a nominal diameter of 24 in. The flange for connection to the runner was drilled and reamed by the generator manufacturer using templates and drilling jigs furnished by the turbine manufacturer who had previously used them in drilling

and reaming the runners. A rather elaborate set of jigs was required owing to the necessity for having the runner, spare runner, and any future replacements interchangeable. The generator builder fitted each runner to the shaft and checked alignments before shipping to the field. The shaft is provided with a stainless-steel sleeve in halves, opposite the turbine stuffing box. A turbine manufacturer's representative supervised the installation of the sleeve and witnessed the shaft and runner alignment.

As usual on large forgings of this kind, a hole was provided in the shaft. This hole was used for certain generator leads and to admit a small amount of venting air to the draft tube through a pipe in the runner cone or fairwater. The hole was also used at first for passing a cable for handling the runner but this was later dispensed with in favor of a jacking arrangement.

Main Bearing and Lubrication. The unit has three bearings, a turbine bearing, a generator-guide bearing below the rotor, and a spherical Kingsbury guide-and-thrust bearing above the rotor. The turbine bearing is of the babbitted oil-lubricated type, 25 in. diam, rather short, being only 16 in. long. The shell is in halves, machine-grooved, tinned, babbitted, bored, and scraped to a clearance of 0.010 in. on diam.

The oil sump is of 74-gal capacity placed in a sector of the crown plate. Oil is supplied to the bearing through a cooler by an a-c motor-driven pump with a d-c motor-driven pump as a stand-by arranged to cut in upon loss of pressure. The pumps, cooler, and gage board are located in an alcove in the turbine pit. The oil supply enters the bearing in an annular groove about one third down from the top, flowing both ways to lubricate the bearing. A reservoir of oil is left at the top of the bearing to act as a supply when changing from a-c to d-c pump or other temporary lack of supply. Excess oil overflows through a pipe to the sump.

Crown and Curb Plates. The crown and curb plates are of cast steel in one piece and of usual construction except that the curb plate is removable from below, after removal of the draft-tube section. This gives access for removal of the wicket gates from below and for maintenance of certain wearing plates and rings under the crown plate.

Wicket Gates and Operating Mechanism. The 20 wicket gates are of forged steel with integral stems. The wearing surfaces at each end of the body of the gates were overlaid by welding with one layer of stainless steel, using a 25 chromium - 20 nickel rod. A total clearance of 0.020 in. was provided.

Each gate has three bearings, one below and two above the body. The lower bearings are greased from below as ample space is available from the passageway around the removable section of the draft tube. Cup leather packing instead of the usual type of square packing is used in the upper gate-stem packing boxes to seal against the high pressures involved. A thrust collar on each gate stem carries the upward thrust against the crown plate.

Each gate lever is made in two sections connected by the usual safety shear pin.

The operating ring and two servomotors, etc., are of usual construction designed for operation with oil at 250 to 300 psi, the displacement of the servomotors being 18.3 gal. In order to suit the design of penstock and surge tank, positive stops were installed on the servomotors to limit the stroke to about 90 per cent, corresponding to a discharge of 635 cfs and about 60,000 hp at 935 ft head.

Governor. The governor is of the Woodward cabinet actuator type with 3-in. pipe connections and rated about 85,000 ft-lb at 250 psi; the governor being designed for a maximum pressure of 300 psi. The minimum time for full gate stroke was specified as 4 sec. However, due to the pipe-line conditions, the closing stroke is limited to 12 sec by $^{17}/_{32}$-in. orifices in the port piping to the servomotors, and the opening stroke to 22 sec by a special small relay valve with limited stroke.

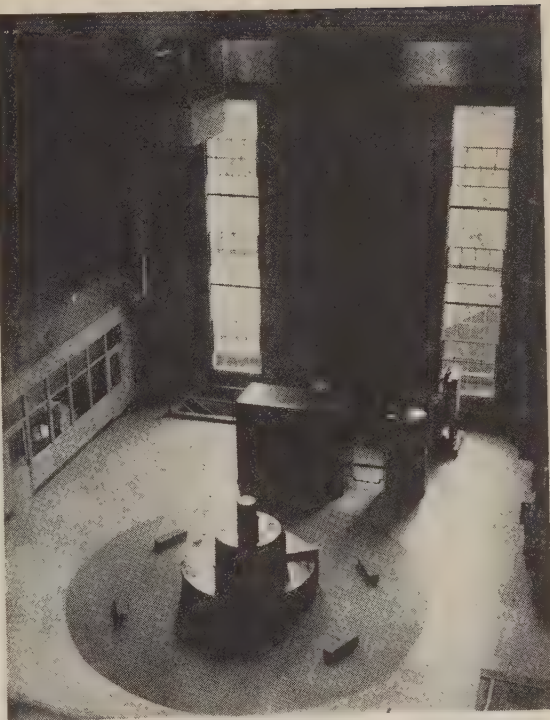

Fig. 5 Interior View of Powerhouse, Including the Governor

The flyballs are driven by a permanent-magnet generator mounted on the generator. Oil pressure is supplied by two 50-gpm motor-driven pumps with echelon control.

The *WR* square of the generator is about 4 000,000 ft^2 lb, giving a regulating constant, WR^2N^2 divided by horsepower, of 13,500,000.

Fig. 5 shows an interior view of the powerhouse, including the governor.

Butterfly Valve. Fig. 6 shows a sectional arrangement of the butterfly valve and operating gear. The valve has a nominal diameter of 8 ft at inlet, reducing to 6 ft 6 in. diam at discharge. Its center line was placed about 20 ft upstream from the center line of the turbine so that there would be no question about the valve disk creating a disturbed flow that would affect the efficiency of the turbine. Body and disk are each in one piece made of cast steel. The shaft is of forged steel keyed to the disk. An 8-in. by-pass is provided for filling the turbine casing before opening the valve, although the valve is designed for opening or closing under full static head of 1008 ft. A 3-in. Crispin air-and-vacuum valve permits the air in the casing to escape and automatically closes when the casing is filled with water. It also admits air when the casing is being drained.

Special care was devoted to the design of the seal rings to reduce the leakage to a minimum and to prevent erosion when the valve was left closed for long periods with the penstock under pressure. Body and disk seal rings are both made of stainless steel of different grades to prevent the possibility of galling. The body seal ring was bored after installation in the body. The disk ring is in halves, secured in its groove, but free to float like a piston ring so that it will adjust itself properly when the valve is closed. This arrangement has proved very satisfactory and there has been little increase in leakage during the 4 years since installation. The choice of stainless steel for the seal rings has proved to be wise.

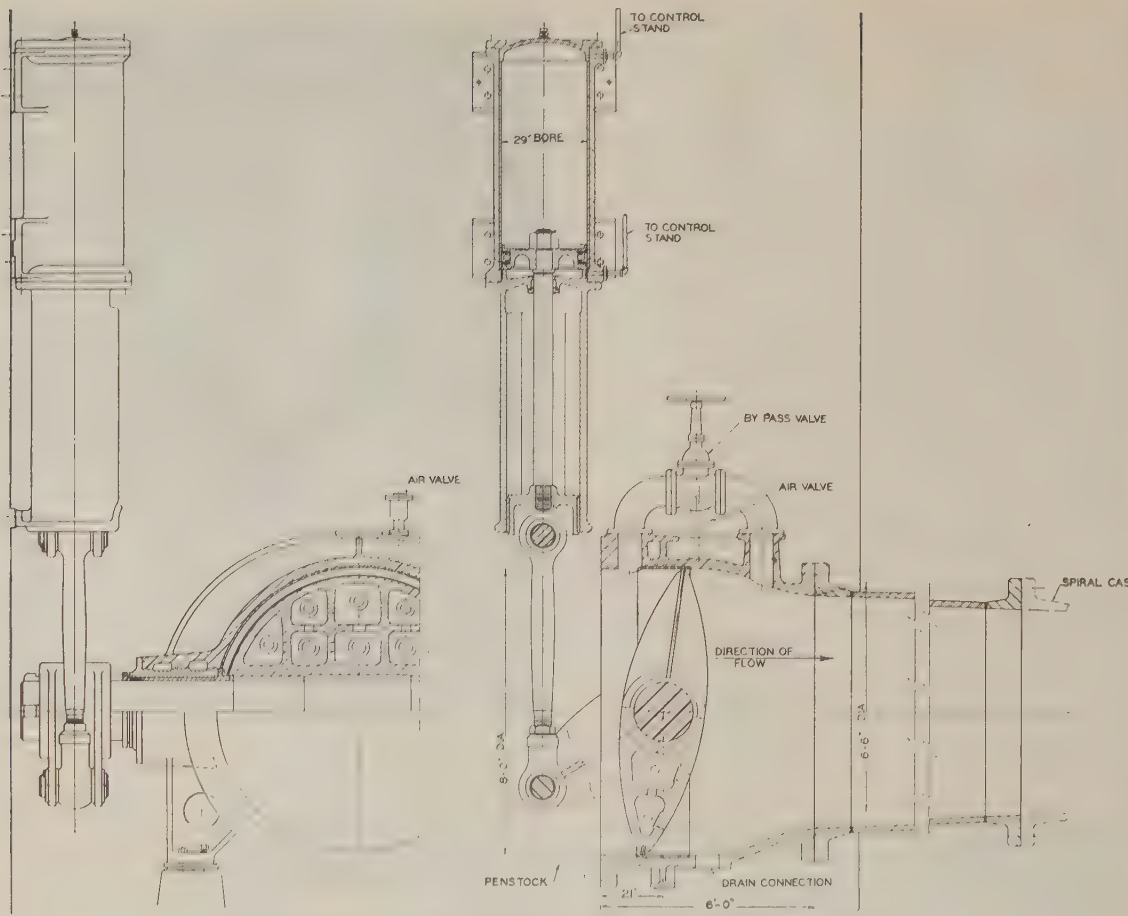

Fig. 6 Sectional Arrangement View of Butterfly Valve and Operating Gear

The valve is arranged to be opened or closed in about 2 min from a control stand on the powerhouse floor, by means of penstock pressure in a bronze-lined cylinder 29 in. diam by about 5-ft stroke. The opening and closing time are limited by the size of the control piping.

Weights. The total weight of the turbine, not including shaft and spares but including governor equipment, was about 300,000 lb. The weight of the complete butterfly valve was about 150,000 lb.

Shop Erection and Tests

The turbine spiral casing, inlet section, and butterfly valve were erected and tested in the shop under a hydrostatic pressure of 700 psi, for 2 hr. Fig. 7 shows these parts assembled with a cast-steel test head bolted to the upstream end of the valve housing. The crown plate and curb plate were in place, and a test ring was fitted between the crown and curb plates opposite where the periphery of the runner would be in the field. The wicket-gate stem holes were temporarily plugged for this test and the butterfly-valve disk was in the open position.

The space between the upstream face of the valve disk and the test head was given a separate test at 700 psi, after closing the disk with a torque of about 80,000 ft-lb. The valve leakage was measured at a pressure of 435 psi, corresponding to full head in the field, and found to be 0.135 cfs. The leakage in the field was much less for then the closing moment was about 10 times as much or 800,000 ft-lb.

A tightness test of 500 psi was placed on the casing and butterfly-valve body in the field before pouring concrete.

Turbine Model and Field Tests

A homologous model was built with a 22-in-diam runner and tested in the manufacturer's hydraulic laboratory before construction of the turbine was started. The homologous parts consisted of all parts which affected the water passages from the inlet to the butterfly valve to the discharge end of the conical draft tube. The model tests were made under a reduced head of about 30 ft. The best efficiency obtained was 89.5 per cent. These tests were made both with and without the butterfly disk in place. No differences in results could be detected.

A complete field test was made in accordance with the A.S.M.E. Test Code for Hydraulic Prime Movers, the water being measured by the Gibson method. Fig. 8 shows the results of the field test, the maximum efficiency being 93.7 per cent.

Sixty-two thousand hp was obtained at 87.5 per cent gate opening which is the maximum gate opening at which the turbine is operated. Above 85.5 per cent gate opening or 60,000 hp, a cavitation noise starts in the draft tube.

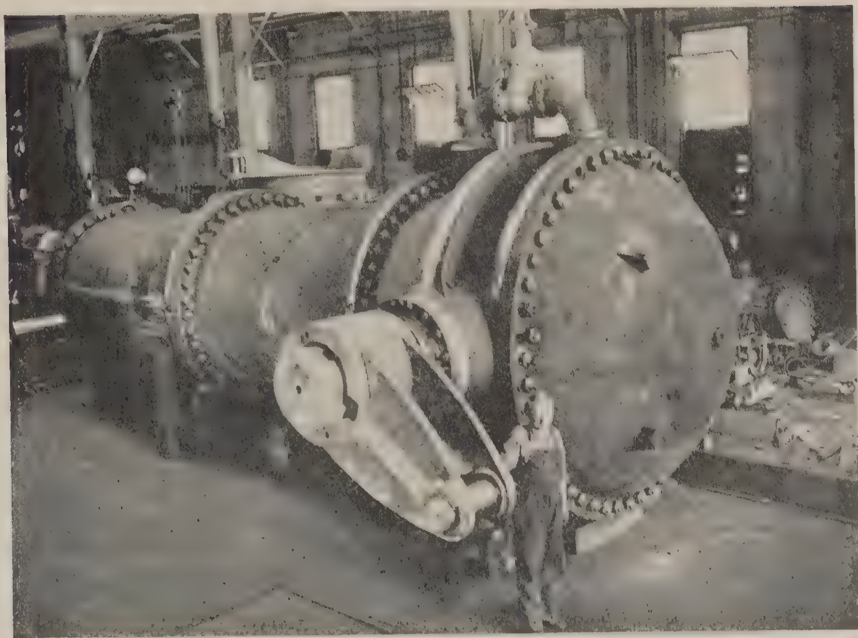

FIG. 7 ASSEMBLY VIEW OF TURBINE-SPIRAL CASING, INLET SECTION, AND BUTTERFLY VALVE AS ERECTED IN THE SHOP FOR TESTING

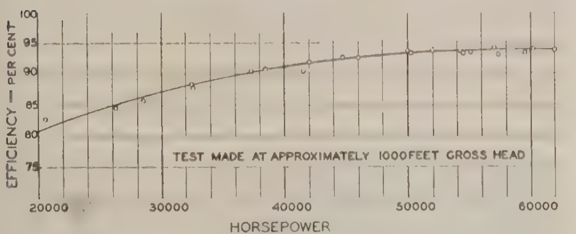

FIG. 8 RESULTS OF FIELD TESTS; MAXIMUM EFFICIENCY 93.7 PER CENT

Other results of field tests are as follows:

Gate opening required to bring unit up to speed, per cent	6.0
Pressure rise for full load off—148 ft or (per cent)	16.0
Speed rise for full load off, per cent	33.0
Power required to motor, runner submerged, kw	1750
Power required to motor, runner unwatered, kw	560
Wicket-gate leakage under full head, cfs	8.2
Oil pressure required to move gates, unwatered, psi	12
Oil pressure required to open gates, full head, psi	175

Gate leakage is sufficient to keep the unit running at 135 rpm before closing the butterfly valve but not sufficient to start unit from rest.

Following is a comparison of guaranteed, model, and field test efficiencies at 925 ft net head:

Per cent load	Efficiencies, per cent		
	Guaranteed	Model test	Field test
100	87.0	89.0	93.7
Best	89.5	89.5	93.7
75	88.5	86.6	93.0
50	84.0	79.0	88.0

Water Velocities. The following tabulation shows the approximate water velocities from the entrance to the butterfly valve to the discharge end of the draft tube, based upon the field test discharge of 610 cfs at rated load and head:

Location	Velocity, fps
Penstock	12.1
Butterfly valve	19.3
Casing entrance	23.5
Casing	29.8
Speed-ring discharge	49.5
Wicket gates	118.0
Entrance to runner, absolute	165.0
Entrance to runner, relative	38.0
Discharge from runner, absolute	37.0
Discharge from runner, relative	115.0
Draft tube, throat	31.2
Draft tube, bottom of cone	10.7
Draft tube, discharge passage	4.3

OPERATION

The turbine was placed in operation July 2, 1942. The field tests were made in August, 1942.

The operation of the Nantahala turbine is not unusual except for several operating limitations that are imposed due to certain design features that resulted from the high head. Some of the original limitations have been mitigated by slight changes in design and by the use of more suitable materials when repairs have been made. The remaining limitations as will be indicated do not impose any serious restrictions on the operation of this unit, in view of the fact that the Nantahala turbine is always operated as part of a large co-ordinated system in which it can normally serve as a base load plant.

The physical location of the powerhouse made a relatively long pipe line necessary between the surge tank and the unit. In order to protect this pipe line against excessive surge pressures it was felt desirable to provide relatively slow governor time. This slow governor timing makes it impossible to use the Nantahala unit for any tie-line load regulation which requires quick changes of load. The slow closing speed also results in an overspeed upon sudden loss of full load of approximately 33 per cent. Since the small load which would normally remain connected to this unit upon the loss of tie line can usually be taken care of by generation at smaller plants, it was considered very desirable to separate the Nantahala unit completely from the remainder of the system upon the sudden loss of load. To accomplish this an overspeed "inertia" relay was interlocked through a watt relay and con-

nected to trip the generator breaker immediately upon the occurrence of a fast rise in frequency, which would indicate loss of a major portion of the load. This relay installation effectively removes the unit from the system within $^1/_2$ sec and prevents excessive overspeeds on the small local system.

The runner of the Nantahala unit is partially submerged by the normal tail water, even when the unit is not in operation. This center line elevation was purposely selected in an effort to minimize cavitation of the runner during normal operation. This results in a positive pressure under the runner for loads up to approximately 57,500 hp at rated head, at which point the velocity head of the water and other disturbances create a negative pressure at the periphery of the draft-tube throat. It was found that when this negative pressure exceeds approximately 14 psi, cavitation noises start in the draft tube. Since the load at which this disturbance occurs varies with the gross head, a table has been prepared showing "maximum gate limit setting versus head," so that the wicket gates will not be opened beyond this critical setting. It has been found that the output varies from approximately 55,500 hp at the minimum head to 65,000 hp at the maximum head before these cavitation noises occur.

The setting of the runner below normal tail water makes it impossible to use the unit as a synchronous condenser without providing special features, such as compressed air to lower the tail water and to provide additional equipment to assure water lubrication to the seal rings during extended periods of operation as a synchronous condenser. After considerable discussion and investigation, this equipment was not considered justified due to the fact that the reactive kva can always be supplied from other units which are located closer to the load center.

A further limitation which was originally imposed on the operation of the machine was a precaution against operating for extended periods below one half rated load due to the excessive wear which would result from operation at light loads. This consideration is no longer as critical as it was originally due to the use of stainless steel in the repair of various of the wearing parts.

Maintenance

It was generally agreed, even before the turbine was installed, that considerable maintenance, particularly of the runner, would be required owing to the high head and high water velocities which exist in this unit. However, some additional problems were experienced which required some redesign as well as some repair.

As originally designed, an oil sump of 74-gal capacity was placed in a sector of the crown plate with the thought that sufficient cooling would be provided by contact of the underside of the oil sump with the water discharged through the turbine. This oil sump provided lubrication through one set of oil pumps to both the generator and turbine guide bearings. It was soon found that there was not sufficient heat transfer through the oil sump to cool the oil adequately, so a heat exchanger was inserted in the oil line between the oil pumps and the bearings.

The first difficulty with this system was experienced due to water spray from the shaft packing gland entering the oil system through the oil-collector pan underneath the turbine guide bearing. This water caused considerable rusting and pitting of the turbine-generator shaft just above the guide bearings, and also a rough rusty appearance on the actual bearing surfaces of the shaft. This condition was corrected by using a rust-inhibiting oil which was very effective in eliminating the rusting, pitting, and sludging which had taken place in the original oil. Admission of water into the system from the packing gland was later effectively eliminated by adding a water deflector to the shaft just above the packing gland, but moisture continued to be present due to condensation on the cool surfaces of the sump walls.

Although the location of the oil sump in the crown plate conserves space in the structural design of the powerhouse, it results in a very unsatisfactory location from a maintenance standpoint. It is practically impossible to clean out completely the sump and to remove all traces of moisture and sediment. In order to remove moisture of condensation effectively, a continuous small blotting-paper filter has been installed in series with the oil line.

It was also found that the one oil-pumping system would not satisfactorily lubricate both the turbine and the generator bearings. This was due to the fact that the generator bearing is located approximately $7^1/_2$ ft above the turbine bearing, and also to the fact that the generator bearing was supplied through a small orifice, whereas the turbine bearing was supplied through a $^3/_4$-in. pipe. Any small or slight obstruction in the generator piping orifice would merely force the oil into the turbine bearing without resulting in an appreciable rise in pressure to remove the obstruction. This combination system was separated to provide an independent forced oil lubrication system with separate oil sump and heat exchanger for the generator guide bearing.

Provision was made in the original design of the turbine for the runner to be removed by means of a cable through a hole in the generator shaft. The cable and runner could then be handled by means of the main powerhouse crane. However, the upper portion of the hole through the shaft was also used for the generator-field leads which extend from the field poles to the collector rings. Early experience indicated that considerable time and effort would be required to remove and reinstall the generator leads every time the runner was removed. It would also be difficult to keep the runner centered when it was being lifted by the cable due to the close seal-ring clearances. It was therefore decided to remove the runner by means of jack bolts and a special frame so that all of the work could be carried on from below the scroll case where the entire operation could be observed and closely checked by the maintenance force. It has been found that by use of this device, a runner can be removed and completely reinstalled by a six-man crew in five 8-hr work shifts.

Another small detail which developed was the fact that under certain conditions when filling the draft tube and scroll case, a water hammer was created which forced water up through the hole in the shaft and out into the generator rotor, and also into the main thrust-bearing oil sump. Since it was felt that the hole served a useful purpose of admitting air to the draft tube, this problem was corrected by installing a check valve at the lower end of the shaft in such a manner that air would be admitted through the shaft to the draft tube but water would be checked and prevented from rising up the shaft.

After the completion of the original installation it was noticed that the wicket gates would not completely pinch when closed against full water pressure. Upon release of the water pressure from the scroll case, the gates would move to the completely closed position. It was found that due to an unbalanced pressure under the gate stem, amounting to an upward thrust of several tons per gate, the upper flange of the crown plate against which the thrust collars pressed deflected sufficiently to allow the top of the gates to rub the upper wearing plate. This condition was corrected by readjusting the thrust collars on the stems and by installing lugs and additional bolts to minimize the deflection of this flange. Whereas it was originally necessary to close the butterfly valve in order to bring the unit completely to a stop, the unit can now be brought to a stop by merely closing the wicket gates.

The major portion of the maintenance and repair of the unit results from pitting and erosion that are caused by the high water velocities and pressures. Realizing that considerable maintenance would be necessary, frequent inspections were made of the unit after it was placed in service. After about 6 months of operation, the wicket gates and the back side of the nonoverlapping parts of the runner vanes began to show signs of erosion or pitting.

After a year's operation, the wearing plates opposite the top and bottom surfaces of the gates had started to erode, the seal-ring clearances had increased from 0.015 to 0.020 in., and considerable wiredrawing could be observed on the closure contact surfaces of the gates, despite the fact that the butterfly valve was always closed and pressure removed from the scroll case within a few minutes after the wicket gates were closed.

After two seasons of operation the Nantahala unit was dismantled with the intention of making minor adjustments and repairs, such as the welding in the field of small areas of the wicket gates, wearing plates and runner that showed wear from an inspection of the completely assembled machine. However, upon removal of the runner the complete extent of the wear could be observed, and it was evident that a major overhaul would be required. It was decided that the following parts should be renewed or repaired: runner and seal rings, throat ring below runner, wicket gates, curb and crown-plate wearing plates and rings.

The stainless steel originally applied to the ends of the wicket gates and the stainless seals used for the butterfly-valve seals showed no sign whatever of erosion or other damage. Fig. 9 shows a typical wicket gate before being repaired. The curb and crown-plate wearing rings and the seal rings on the runner had originally been made of S.A.E. 1045, a carbon-manganese steel, heat-treated to a Brinell hardness of 200 to 235. It was thought that this material would be suitable for resisting the erosion from the high velocities, but this was not the case. The stationary seal rings were made of cast steel, each with two bronze inserts held in place with calking strips. The bronze inserts eroded faster than the carbon-steel rings on the runner.

The parts to be restored were shipped back to the turbine manufacturer and were treated as follows:

The nonoverlapping back sides of the runner vanes and the inside of the runner band from the vanes to the bottom edge were recessed about $^1/_8$ in., overlaid with 25-20 stainless welding and refinished. The runner wearing rings were replaced with steel rings overlaid with 25-20 stainless-steel welding. It would have been less expensive to use stainless bar material for this purpose but necessary early delivery could not be obtained during the war.

Fig. 9 Typical Wicket Gate Before Being Repaired

The bore of the cast-steel throat ring for a distance of 6 in. from the top was similarly treated with a stainless overlay.

The bronze inserts in the stationary seal rings were replaced with inserts made of a 13 chrome-stainless steel. This gave two different grades of stainless steel working opposite each other so that even if they should accidentally touch in operation, galling would not result.

The entire downstream side and one third of the upstream side of the body of each forged-steel wicket gate were overlaid with 25-20 stainless welding and refinished. The intermediate and lower wicket-gate stem bearings which had pitted somewhat from corrosion were restored with a stainless metal-spray finish, and a collar was shrunk on the lower gate stem adjacent to the gate to help retain grease on the lower bearing surface.

The curb and crown-plate wearing plates and rings opposite the runner periphery were renewed with mild steel overlaid with 25-20 stainless welding. At the same time total clearance at the end of the wicket gates was reduced from 0.020 to 0.015 in.

In order to make the foregoing repairs the unit was taken out of service on March 6, 1944, and by working continuously it was put back into operation on May 2, 1944. No water was lost from the reservoir during the 56 days the unit was down.

As a result of the changes the power output was increased about 5 per cent, 3000 hp, because the runner-discharge openings were increased. The leakage through the runner seals was even less than when the unit was first installed. With the smaller wicket-gate clearance and complete closure of the gates, the unit would stop of its own accord with the gates closed and the butterfly valve left open, whereas previously it would not.

After two additional seasons of operation the runner was again removed for a detailed inspection of the unit. All of the stainless steel which had been properly applied was still in perfect condition. Even the tool and the file marks still showed, indicating that very little if any metal had been removed due to erosion or pitting. The closure contact surfaces of the wicket gates were also in perfect condition. A number of spots were found on which the stainless-steel weld had been almost completely removed when the piece was remachined and those spots were pitting or eroding appreciably. These small areas were chipped out and filled in with 18-8 stainless-steel rod; and subsequent inspection indicates that this work was very effective.

Evidence of the effectiveness of stainless steel was clearly demonstrated by surfaces of the stationary wearing ring which is installed in the crown plate opposite the periphery of the runner. Through an oversight in the repair work, an inside vertical surface of this ring was coated with $^1/_8$-in. stainless-steel overlay but the lower horizontal surface which adjoins the wearing plate was not coated except for the $^1/_8$ in. vertical overlay that extended to this lower horizontal surface. After two seasons of operation the mild steel between the wearing plate and the $^1/_8$-in. stainless-steel overlay was eroded or pitted to a depth of $^1/_{16}$ to $^3/_{32}$ in. This area was repaired in place by chipping out and filling in with stainless-steel welding rod.

After the runner was removed for this inspection it was decided to install the spare runner and to perform additional stainless-steel welding on the original runner to cover areas which were not repaired when the unit was returned to the factory. These areas consist primarily of the top and bottom edges of the periphery of the runner and a portion of the working face of the vanes where the water impinges upon starting. Considerable

erosion occurs on these faces because the runner acts as an impulse rather than a reaction wheel at small gate openings.

It was gratifying to note that even the seal rings did not show any wear. The manufacturer showed good judgment in applying different grades of stainless steel to the stationary and movable seal rings, since it was found that they had actually touched in a few spots without appreciable damage. It was found that the contact between the two surfaces was due to warpage of the inserts, which probably resulted from internal stresses which had not been relieved. This was corrected by grinding the stationary seal-ring inserts to a true circle by means of a field-constructed jig, and by buffing the movable rings. All of the surfaces were trued up and refinished without changing the original clearances appreciably.

The use of stainless steel has greatly reduced the amount of maintenance work which would have been required to keep the unit in a safe and efficient operating condition. By maintaining smooth surfaces it undoubtedly greatly increases the average long-term efficiency of the unit and removes some of the previous limitations that had been imposed to reduce wear on the unit.

Conclusion

During the more than 3 years' operation of this unit, all the difficulties encountered have been corrected. It is believed that the moderate initial cost (as compared with other types), the very high efficiency, and the absence of serious operating difficulties have amply justified the selection of the Francis type for the conditions under which this turbine operates, and that it will have a long life, with only moderate renewals and repairs.

Discussion

G. D. Johnson.[4] The use of Francis instead of impulse turbines under head of 1000 ft and over has again been vindicated by the Nantahala turbine.

Because of the high efficiencies attainable and the economies in first cost due to high-speed equipment with small space requirements for the amount of power developed, there is a natural inclination to use Francis turbines for higher and higher heads. However, because of operating difficulties experienced with previous high-head installations, there are today only a few Francis turbines in the entire world operating under heads approximating 1000 ft. Apparently use of stainless steel and improved welding techniques will now assure satisfactory, reliable operation of units of this type without excessive maintenance.

The writer was privileged to witness a vivid illustration of this modern trend, when in November, 1945, he visited the Lages Plant of the Rio de Janeiro Tramway, Light, and Power Company (The Light) in Brazil. At present this plant contains eight four-jet vertical-shaft impulse turbines and two vertical-shaft Francis turbines, all of European manufacture, operating under a rated head of 310 m (1018 ft). The small space occupied by the 50,000-hp 600-rpm Francis units, as compared with the size of the 9000-hp and 19,000-hp 300-rpm impulse units, was striking. Moreover, the maintenance on the Francis wheels, installed in 1940, and 1942, respectively, and equipped with special alloy-(stainless-) steel runners, has consisted mainly of stainless-steel welding on the gates and is no more than is required on the runners, needles, and nozzles of the impulse wheels. No erosion or cavitation of the runners was apparent after 3 to 5 years of service. This is due, of course, to the use of special material in the castings, but it is apparent that, as in the case of the Nantahala turbine, the liberal use of stainless steel results in units requiring a minimum of maintenance.

The writer congratulates the authors on the courage and skill demonstrated in the conception and design of the Nantahala turbine and on the success which they have achieved in its operation, as well as on the manner in which they have presented the complete story in their excellent paper. Their generous discussion of design considerations and operating experiences will undoubtedly promote further worth-while developments in the application of hydraulic turbines to high-head power generation.

S. J. Needs.[5] The paper was particularly interesting to the writer, who happened to be at Nantahala on several occasions while the machine was being installed. It is also timely in that it discusses a most unusual machine after it has been proved successful in service. Comparison with larger machines makes it difficult to realize that such a small turbine can develop so much power. Its high output is due, of course, to the exceedingly high head and this is one of the features which make the installation so outstanding.

The authors mention the spherical thrust bearing selected for this machine, and since the bearing is also quite unusual and interesting, a few words about it are in order. Instead of the flat thrust and cylindrical upper guide bearing, common in vertical machines of this type, a spherical thrust bearing is used. This bearing is capable of supporting radial as well as vertical loads, hence it eliminates the upper generator guide bearing.

The spherical thrust bearing for the Nantahala unit is shown in Fig. 10 of this discussion. It is located at the top of the machine and it runs in an oil bath with a cooler like most Kingsbury bearings in hydroelectric service. It is 51 in. diam and to date is the largest spherical bearing that has been built. The rubbing surface of the runner A is a zone of a sphere, the center of which is on the shaft axis. Six segmental spherical shoes B are fitted to the runner and pivoted in the usual Kingsbury manner. The bearing angle of the shoes is 39 deg with respect to the shaft center line. Total vertical thrust load is 575,000 lb or about 96,000 lb per shoe. Due to the bearing angle the normal load per shoe is about 123,000 lb and its horizontal component is approximately 78,000 lb. The shaft is held concentrically by the horizontal components of the thrust load acting radially at each shoe. The babbitt surface area of each shoe is 261 sq in. and loading per unit area is 472 psi.

At the normal operating speed of 450 rpm the average minimum film thickness along the trailing edges of the shoes is 0.0026 in. under the loading cited. When a horizontal load comes on the bearing the shaft can move sidewise due to the slight decrease in oil-film thickness at the more heavily loaded shoes. A side load of 10 per cent of the vertical thrust will decrease the minimum film thickness 0.0004 in., or 15 per cent. Twenty per cent side load will decrease the minimum film thickness 0.0006 in. or 24 per cent, and 40 per cent side load will cause the normal minimum film thickness to decrease somewhat less than 0.001 in. or about 36 per cent. Sidewise movement of the shaft is approximately the same as the change in minimum film thickness, from which it follows that the shaft is held very close to a central running position, even under appreciable side loads.

Total frictional power loss at full load and speed due to oil-film shear is 175 hp. This is about the same as the combined frictions of the corresponding flat thrust and cylindrical guide bearings. The frictional heat is removed by a cooling coil consisting of approximately 1000 ft of $1^1/_8$-in-diam copper tubing. Oil is circulated around the cooler by means of a centrifugal pump consisting merely of two opposite holes drilled in the runner at an angle. At 450 rpm the peripheral speed of the

[4] Hydraulic Engineer, S. Morgan Smith Co., York, Pa. Mem. A.S.M.E.

[5] Service Manager, Kingsbury Machine Works, Philadelphia, Pa. Mem. A.S.M.E.

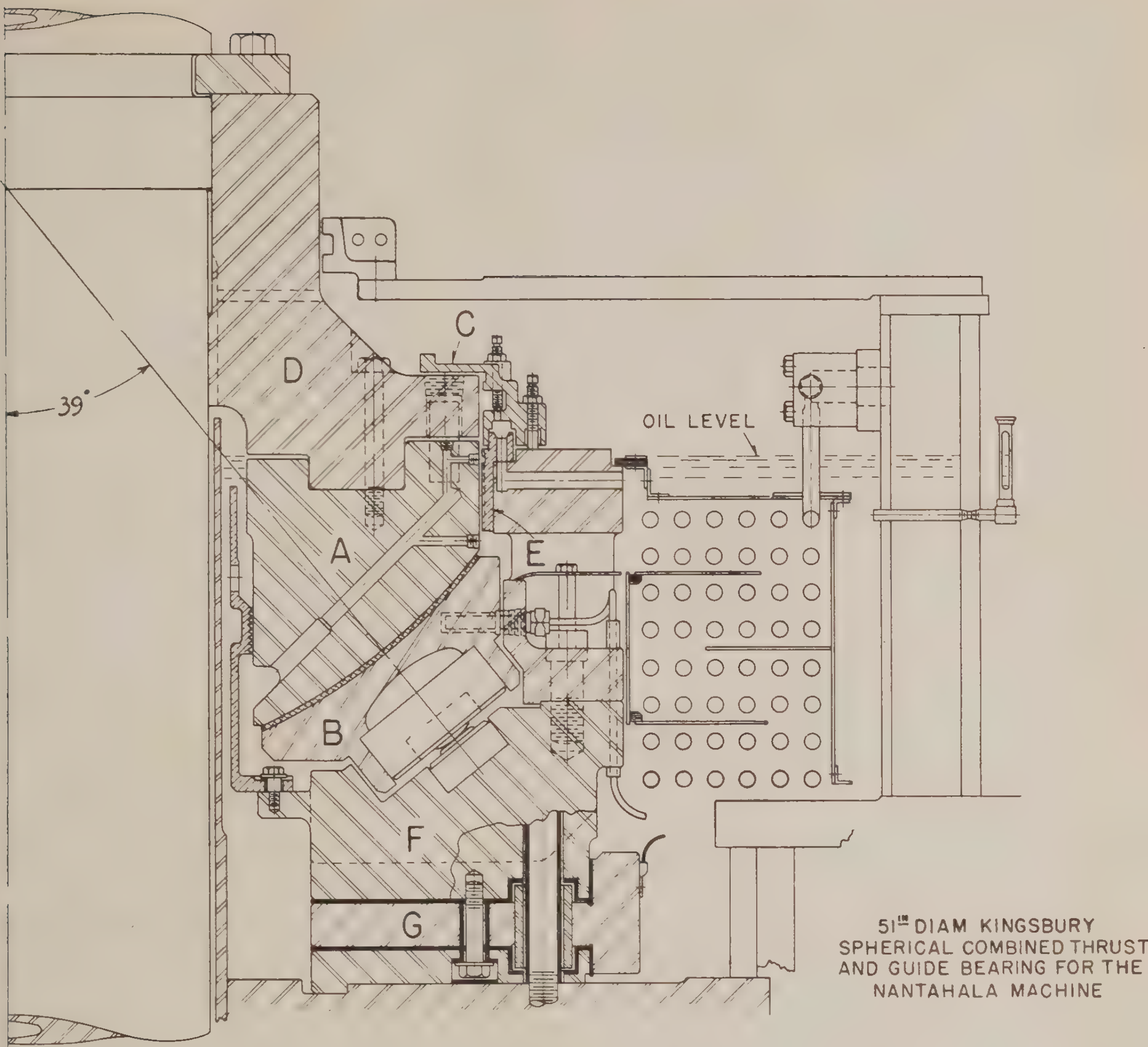

Fig. 10 Spherical Combined Thrust and Guide Bearing for the Nantahala Machine; 51 In. Diam

runner is 101 fps or about 69 mph. Static pressure at the discharge ends of the pumping holes is 43 psi and about 35 gpm is circulated around the cooler tubes. This flow is increased by the centrifugal action of the runner surface on the oil in the spaces between the shoes.

Clearance between the air-seal ring *C* and the removable thrust block *D* is so adjusted that oil is maintained in this space, thus sealing the bath against entrance of air. The inner air-seal ring at the runner bore prevents air from entering the oil at that point. Due to these seal rings the top of the bath is clear and practically motionless.

In the unlikely event that the shaft should ever rise, a cylindrical bronze guide bearing *E* is provided. Radial clearance between the outside diameter of the thrust runner and the guide bearing is $^1/_{16}$ in., hence this emergency bearing is practically frictionless in normal operation.

Beneath the solid base ring *F* the bearing is provided with double insulation against shaft currents. A lead from the interbase *G* makes it possible to test the insulation at any time.

The principal advantages claimed for the spherical thrust bearing are as follows:

1 The shaft is held in a concentric running position as if in a guide bearing of zero clearance.

2 Any inaccuracies of the thrust block and ring key are compensated for by the permissible rolling action of the spherical runner. Movement so induced at right angles to the direction of motion would be of small amplitude and practically unresisted by the fluid oil film. Hence galling between thrust block and shaft and shaft throwout at the turbine bearing due to thrust-block inaccuracies are eliminated.

3 The spherical bearing simplifies design of the thrust bracket by eliminating the cylindrical upper generator guide bearing and the special oil pump and piping usually required for its lubrication.

4 The spherical bearing is so constructed that at zero speed the babbitt faces of the shoes are segments of a true spherical zone and will bear equally against the runner. Thus equal shoe

loading is inherent, no field adjustments of loading are necessary.

5 The vertical center lines of bearing and shaft will intersect at the center of the runner sphere, but it is not essential that these center lines coincide. Hence equal shoe loading will obtain despite shifting of foundations and appreciable departure of the shaft from plumb. To obtain this self-alignment with a flat bearing requires the use of leveling-plate equalizing construction.

J. F. Roberts.[6] This 60,000-hp Francis-type turbine, operating under 925 ft net head, has set a new mark in American practice for high-head reaction-type turbines.

One point which strikes us as particularly important is the marked improvement which stainless steel shows as compared with either cast steel or rolled steel in resisting cavitation and scour erosion due to the high velocities under these relatively high heads. This further confirms data gathered in regard to cavitation of Kaplan and propeller-type wheels. Possibly the Europeans are right in their conclusions that it is well worth while to make the runners and some other vital parts entirely of stainless steel. In this particular case, had the runner, guide vanes, and facing plates been made of stainless steel, possibly all of the repairs required at the end of 2 years of operation might have been avoided. Until recently it has been impossible to secure a stainless-steel runner in this country, but during the war several of the steel foundries have enlarged their stainless-steel casting capacity and several stainless-steel runners are now being manufactured. It will be interesting to see whether or not these outlast their carbon-steel predecessors.

The fact that stainless steel was used in the butterfly valve both as the seal ring on the disk and seat in the housing with excellent results further confirms the data regarding its greater resistance to wear and erosion. While discussing the butterfly valve, the excellent streamlining on the downstream side of the disk so as to maintain practically uniform velocity as the water flows from the valve into the spiral casing deserves mention. The housing contracts rapidly so that there is no decrease in velocity as the water leaves the area surrounding the disk, which in this case probably occupies from 25 to 30 per cent of the effective area in the valve.

The company with which the writer is connected built a 30,000-hp impulse turbine of the double-overhung type for this same power company at about the time the Nantahala turbine was built. The difference in efficiency of an impulse turbine as compared with the reaction turbine was more than expected. While the impulse turbine gave about 1 per cent higher than the expected 88 per cent, or about 89 per cent, it still was over $4^1/_2$ per cent below the 93.7 per cent obtained on the Nantahala turbine. Mr. Growdon and the other officials of the Nantahala Power Company, in discussing this difference, agree with the writer that if we had had as much experience with high-head reaction turbines at the time of building these plants, both plants, that is, Glenville as well as Nantahala would have been of the high-head reaction type.

The tabulation showing the comparison in efficiency between the 22-in. model and the 85-in. Nantahala turbine deserves consideration, especially the increase of 9 per cent at half load from the model to the full-sized unit in the field. This is also one of the rare cases where the actual field tests showed a greater improvement in the plant than is indicated by the Moody formula, as the Moody formula would show only about 92.5 per cent efficiency as compared with the 93.7 per cent actually obtained.

While the Nantahala turbine is set with the center line of the spiral casing about 4 ft below the average tail water, the fact that serious cavitation noises occurred at loads slightly below rated capacity indicates the desirability of setting this type of turbine even lower in order to decrease such cavitation. Had this turbine been set some 4 or 5 ft lower than it actually was, we feel that the cavitation limit would have been materially improved, that is, larger capacity would be possible without cavitation noises and, undoubtedly, considerably less pitting and erosion would have shown up on the runner.

The authors are to be complimented on the frankness of their discussion in placing this valuable information before the Society. There is no doubt that this paper will be beneficial to many other prospective users of this specialized type of hydraulic equipment.

F. H. Rogers.[7] This paper is of great interest to both the operating companies and the turbine manufacturers as the use of Francis turbines for heads of about 1000 ft is exceptional. The authors are to be congratulated on the frank description of the original difficulties experienced and the corrective methods used. The maximum efficiency obtained of 93.7 per cent is indeed a splendid record.

As pointed out in the paper, the successful use of a Francis-type unit instead of the impulse type results in higher efficiency, higher speed, and lower cost. The difficulties to be guarded against are unstable operation, pitting due to cavitation, and wear of internal parts, caused by the extremely high velocities and abrasive materials in the water, and loss of the original efficiencies due to wear at the runner seals.

An interesting comparison with the Nantahala turbine is the 39,000-hp unit built by the writer's company for the Ixtapantongo Development of the Comision Federal de Electricidad in Mexico. This unit was designed for a net head of 1028 ft at a speed of 600 rpm. The specific speed is 20.3, only slightly less than the value of 21.6 given in the paper.

In the Nantahala unit a radial clearance of about 6 in. was allowed between the wicket gates and entrance to the runner vanes to result in smooth flow from the gates into the runner. For the Ixtapantongo unit the corresponding clearance was about $2^1/_2$ in. It is our experience that so large clearance is not necessary if wicket gates are so streamlined as to produce smooth flow. This smaller clearance results in decreased size of unit and lower cost.

The authors describe how the heavy upward hydraulic thrust on the bottom of the gate stems caused a deflection of the upper flange of the crown plate, which permitted the top of the gates to rub the upper wearing plate, and this friction prevented sufficient gate closure to stop the turbine completely.

This difficulty was avoided in the Ixtapantongo unit by extending the lower gate stems through stuffing boxes in the lower cover, thus eliminating all upward thrust.

The repairs made after 2 years of operation are of particular interest. It is noted that considerable wear occurred on the runner, throat ring, gates, curb, and crown-plate wearing rings and seal rings, and that these parts were repaired by welding the surfaces with 25-20 stainless-steel rods.

In the case of the Ixtapantongo unit, the same carbon-manganese steel (S.A.E. 1045) was used for the wearing rings and seal rings, and the same bronze inserts in the stationary seal rings. This unit, however, has been in operation only about 18 months, but the power company reported that an inspection made after the first year of operation showed no appreciable pitting or wear of the internal parts.

The authors speak of erosion or mechanical wear rather than pitting and the writer would like to know if this wear was caused by erosive action from sand or other abrasive materials in the water. It is noted that water is carried through an unlined pres-

[6] Manager, Hydraulic Department, Allis Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

[7] Manager, Hydraulic Turbine Sales, Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

sure tunnel, which might result in foreign materials being present in the water.

It has been our practice to use 18-8 stainless steel on the runner vanes, gates, wearing rings- and seal rings to resist pitting rather than the higher chrome-nickel materials. We understand that recent tests made at the Massachusetts Institute of Technology indicated that a 17-7 stainless steel showed greater resistance to pitting than the higher combination of chrome and nickel.

It is quite possible that the 25-20 stainless steel used at Nantahala has a higher Brinell hardness and will stand up better where sand is present under these extremely high velocities. We would like to have the authors' opinions on this matter.

The efficiencies obtained on the field test as compared to the model test are unusual, as at best efficiency the prototype shows 93.7 per cent and the model 89.5 per cent; a gain of 4.2 per cent. The well-known Moody formula[8] would show for the runner sizes and heads stated, a gain of $3^1/_4$ per cent for a coefficient of $n = 0.25$, and a gain of $2^3/_4$ per cent for a coefficient of $n = 0.20$. The gain at part loads of 6 to 9 per cent over the model efficiencies is even more unusual, as such gains are rarely higher than at the maximum efficiency. Our company is of the opinion that the exponent $n = 0.25$ results in too high an efficiency for the prototype and is now using the value $n = 0.20$.

R. E. B. Sharp.[9] An efficiency of 93.7 per cent with a specific speed of 21.6 represents an important advance in hydraulic-turbine performance. Surrounding conditions, justifying large units of the Nantahala characteristics are rare, and unfortunately the conducting of field tests of such units is even rarer.

It is gratifying for those interested that the Aluminum Company has seen fit to test not only the Nantahala unit, but also four of the Shipshaw units as well, three Allis-Chalmers, and one S. Morgan Smith, with water measurements by the Gibson method and all with excellent efficiencies. The Smith unit attained a maximum efficiency of 93.6 per cent and was one of four built by S. Morgan Smith, Canada, Ltd.

The Ixtapantongo unit, which was designed under the writer's direction and built by The Baldwin Locomotive Works for the Comision Federal Electricidad of Mexico, has, regrettably, not been field-tested. This unit is also of the Francis type, with a rating of 39,000 hp under a net head of 1028 feet, with a specific speed at rated power of 21.8. It was put into operation during the summer of 1944, and therefore does not have as long a case history as the Nantahala unit.

In an effort to predict, on the basis of the Nantahala performance, the maximum efficiency which might be attained from a runner of still lower specific speed, the writer, in Table 1 of this discussion, has made a comparison of those estimated losses which vary most greatly with specific speed.

The disk losses include only the external runner surfaces, both of the crown and band. The Unwin formula

$$2.07 \times 10^{-10} \times \text{rpm}^3 \times \text{diam in ft}^5$$

was used. For the surfaces which depart from a disk shape a

TABLE 1 COMPARISON OF ESTIMATED PERFORMANCE OF VARIOUS HYDRAULIC-TURBINE RUNNERS

	Shipshaw	Nantahala	Hypothetical unit
Specific speed at best efficiency	44.3	21.6	15
Speed, rpm	128.6	450	450
Head, ft	208.0	925.0	925.0
Horsepower at best efficiency	74000	60000	29000
Diameter of runner, entrance, in.	154	85	85
Diameter of top draft tube, in.	159.75	59.5	46
Radial clearance at seals, in.	0.0625	0.015	0.015
Disk loss, per cent	0.785	1.340	2.780
Clearance loss, per cent	0.578	0.650	1.007
Draft-tube loss, per cent	0.875	0.327	0.214
Shock loss at entrance, per cent	0.426	0.219	0.146
	2.664	2.536	4.147

summation of small divisions was made. The clearance losses were obtained by determining the velocity head at the seals and deducting this from the head acting, using a coefficient of 0.5 through the area and allowing for the reduced head on the seals due to the centrifugal effect between the runner periphery and the seal location. The authors may be able to state the measured Nantahala clearance loss. The draft-tube losses were calculated assuming a draft-tube efficiency in all cases of 80 per cent. The shock losses are on the assumption that one tenth of the radial velocity head entering the runner is lost. All losses were calculated at best efficiency. While the aggregate losses not included in this comparison amount to more than the sums of those included, it is believed that this comparison gives a fair indication of what efficiency might be expected from a Francis turbine of still lower specific speed for possible use with an appreciably greater head than 1000 ft, that is, something over 92 per cent for such conditions would be in order with a runner of about 85 in. diam. An account[10] has been given of a unit of Escher Wyss design which attained from field test an efficiency of 91.4 per cent when developing 16,000 hp under a head of 918 ft, at a specific speed of 15.07. The diameter of this runner was only 70 in.

The introduction of 80 gpm of clear water into the seals is an interesting feature, but the small amount used leads the writer to question the advantage gained thereby as a means of protecting the seal rings from the passage of foreign matter. Possibly the original reason for the introduction of this water was in connection with synchronous-condenser operation.

One of the principal advantages in obtaining well-conducted field tests, to the turbine designer, is the comparison thereof with the model performance, as this is the yardstick for use in future guarantees and designs. The Nantahala field test is unusual in that the step up in efficiency is so great. The Moody formula on the basis of the losses varying inversely with the one-fourth root of the runner diameters would result in a field efficiency of 92.5 per cent. A field efficiency of 93.7 per cent corresponds with an exponent of about $\frac{1}{2.6}$. Possibly one reason for the large increase might be in the relatively large model seal clearance. For this value to be homologous on the model it would be

$$0.015 \times \frac{22}{85} = 0.0039 \text{ in.}$$

which is an impracticably small value.

On the lower portion of Fig. 11 of this discussion is shown the model curve as compared with the field, as plotted from the data in the authors' paper. It is noted that the guarantees at part load are disproportionately higher than the model curve which falls off quite rapidly. Further information in regard to this point would be of interest.

On the upper part of Fig. 11 a comparison is given of the Shipshaw field and model tests of the Smith unit tested. In this

[8] Moody formula:

$$E_p = 100 - (100 - E_m)\left(\frac{D_m}{D_p}\right)^n \times \left(\frac{H_m}{H_p}\right)^{0.01}$$

where

E_p = efficiency of prototype
E_m = efficiency of model
D_p = runner diameter prototype
D_m = runner diameter model
H_p = effective head on prototype
H_m = effective head on model

[9] Consulting Engineer, S. Morgan Smith Company, York, Pa. Mem. A.S.M.E.

[10] "Vemork Reaction Turbine Operates Under 918-Ft Head," Escher Wyss & Company, *Power*, vol. 73, Jan.-June, 1931, p. 630.

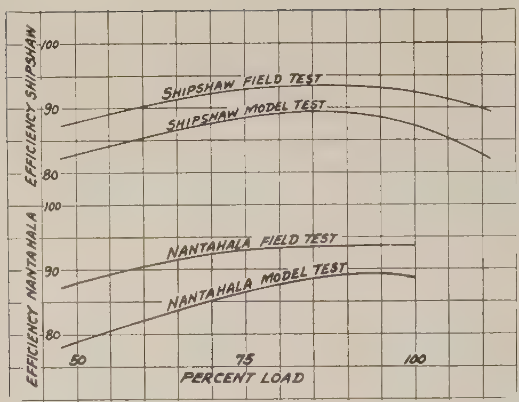

Fig. 11 Efficiency Curves of Model and Field Tests of Nantahala and Shipshaw Turbines

instance the 19.5-in-diam model attained 89.6 per cent efficiency. The Moody formula steps this up to 93.8 per cent field efficiency, which agrees quite closely with the actual value of 93.6 per cent. Both the Shipshaw and Nantahala field tests show satisfactorily high part-load efficiencies, with the Nantahala values somewhat higher if they are both brought to the same percentage of load for maximum efficiency.

The authors make no mention of presence or absence of foreign matter in the water, but it would appear that there is some abrasive matter present, and possibly some degree of acid content. The writer had occasion to examine the seals of Units N5 or N6 at Boulder after about a year of operation under a head above 500 ft, and found the clearances on the average slightly less than when the units were installed, due to the deposit, in some places, of verdigris on the white brass inserts in the stationary rings. The Boulder water was, during this period, exceptionally free from abrasive matter.

The substitution on the Nantahala turbine, of stainless steel on the stationary wearing ring for bronze, while showing gratifying results from the standpoint of maintained low leakage loss, appears questionable to the writer from the standpoint of safety during possible runaway speed. Assuming a runaway speed of 725 rpm, which is believed to be conservative, calculations made indicate an expansion at this speed of about 0.008 in. on the radius, leaving 0.008 in. as the clearance under that condition or a possibility of this being reduced to 0.003 in. if the shaft swings to its extreme position in the bearing. Even though the rotating and stationary seal surfaces are not subject to galling, the presence of two broad surfaces of hard metals under this admittedly abnormal condition might result in violent seizure due to heating after contact. In view of the excellent resistance of stainless steel for these surfaces, as brought out by the authors, labyrinth seals of this metal along the lines used in steam-turbine design might be a good solution from all standpoints.

On the Ixtapantongo unit mentioned the lower gate stems extended through stuffing boxes in the bottom, or curb plate, and were of the same diameter as the upper stems. This was for the purpose of avoiding the heavy upward thrust on the stems, and also of avoiding the loss of grease due to water passing under the lower ends of the stems and forcing the grease upward on the sides of the stems nearest the runner.

The writer wishes to congratulate Mr. Terry on the excellent design and performance of the Nantahala turbine, and all of the authors on the frank manner in which they discuss the difficulties encountered. This attitude is of distinct long-range benefit both to manufacturers and users.

Authors' Closure

The extent and quality of each of the several discussions is greatly appreciated by the authors. These discussions add greatly to the value of the paper.

While the spherical Kingsbury thrust bearing is installed on the generator, it is of course an essential part of the turbine as well. Mr. Needs's description of the design and construction of this bearing is most interesting and timely. So far, the bearing has functioned perfectly and we would definitely favor its use for hydroelectric units.

Mr. Roberts has misinterpreted the statement in the paper with respect to cavitation noises in the draft tube, since these noises only occur at loads above rather than slightly below the rated load for the various heads. The turbine operates regularly at between 45,000 and 46,000 kw under a net head of 951 ft without cavitation noises in the draft tube.

Several of the discussers raised questions relative to the presence of abrasive materials or acid in the water. The tunnel is driven through hard insoluble Arkose rock. It is unlined except near the portals. It was thoroughly cleaned and washed out before being filled. The washing removed most of the rock dust and small gritty particles. Some rock dust remained and was taken out by the water during the first few hours of operation. The water velocity in the unlined section of the tunnel at full load is approximately 4.75 fps, which is insufficient to transport loose material through the tunnel. Practically no rocks or sand have been found in the rock catcher, which is located at the end of the unlined tunnel just before the water enters the steel penstock. Except for the first few hours of operation, we believe that no sand or rock dust has passed through the turbine.

The drainage area is generally heavily wooded, with oak trees predominating. The water from this drainage area is therefore slightly acid. It does not affect the rate of corrosion.

The runner seal-ring clearances for the turbine model were 0.008 in., or relatively twice as large as for the main unit, which partly accounts for the large step up in efficiency, particularly at the lower percentages of output.

While the bulk of the welding with stainless was done with a 25-20 rod, some repair work was done in the field using 18–8 and 17–7 rods. These three grades of stainless rods have apparently stood up equally well in the field. 25–20 was selected because of the manufacturer's experience with the application of this grade. The resulting deposit, is, of course, diluted by the parent metal in any case, dilution varying somewhat with welding technique. The 25–20 rod is quite readily applied, and the resulting weld is somewhat more ductile than the other two.

The 80 gpm of cooling water supply to the runner seal rings was intended for use if the unit was operated as a synchronous condenser and not for the exclusion of any foreign matter. The unit is not, however, used as a synchronous condenser but the operators make a practice of turning on the seal water just before starting the machine. The supply valves are left open as long as the unit is in operation. The actual necessity for this procedure is questionable but it is certainly on the safe side. The seal supply pressure is limited. Consequently, the supply line is provided with a check valve to keep the water from backing up in the line as the turbine wicket gates are opened and the pressure builds up in the seal spaces.

The Ixtapantongo unit described by Mr. Rogers is of considerable interest. Experience with that unit for the first year closely parallels that of the Nantahala Unit. It was only after two years of operation at Nantahala that the extent of the wear with the carbon steel was appreciated. In other words, carbon steel stands up satisfactorily under this head for a limited time but it is the authors' opinion that stainless is needed for long-time trouble-free service.

New Developments in Combination Controls

By J. W. Kelly,[1] Burbank, Calif.

This paper deals with combination controls, which present advantages of weight and cost saving which in recent years have been greatly enhanced by the dependability and ruggedness which can be built into them. Small electric motors, developed for aircraft-accessory operation, have permitted the construction of motor-operated valves. Both solenoid- and motor-operated valves now give the aircraft designer a wide choice of control characteristics. New methods and improvements in older types of follow-up systems for hydraulic controls are discussed in the paper.

Introduction

COMPETITION between mechanical, hydraulic, and electrical modes of actuation has, through the last few years in the aircraft industry, produced certain results that would probably not have been obtainable otherwise.

After maximum efficiency with minimum weight had been obtained from each separately, it became obvious that combinations of the mechanical, hydraulic, and electrical systems, utilizing the best of each, would produce results heretofore unobtainable.

Hydronic Units

Realizing this condition, the author's company undertook a program several years ago of developing what we term "hydronic" controls (hydraulic-electrical combinations) and various mechanical-hydraulic combinations.

Solenoid Valves. The first of the hydronic units (the electrical-hydraulic combination) to be developed were the solenoid-operated selector valves. The field of available solenoids was thoroughly covered for possible standard units usable to operate selector valves. Unfortunately, none was found that came within our weight expectations; therefore a series of developmental solenoids were made. The results of tests on these solenoids set up the general type of solenoid to be used for valve actuation. One of the first solenoids put into production is the one shown in Fig. 1, which, with a weight of 21 oz, gives a pull of 60 lb over a stroke of $^1/_8$ in. at 18 volts. This solenoid was designed for intermittent operation and would reach a maximum temperature of 250 F after several hours of continuous operation.

Still not satisfied with the original solenoids, attempts were made to reduce the weight further. Finally a double-coil arrangement was worked out with a primary and secondary coil. When the solenoid is first energized, both coils contribute to the pull. The plunger, upon reaching the end of the stroke, operates a switch as shown in Fig. 2, interrupting the current to the primary coil, allowing the secondary coil to apply a holding pull. Thus an instantaneous high-amperage input operates the valve; and the secondary coil, with low-amperage input, taking advantage of the principle that pull increases inversely to the air gap, holds the valve in the desired position. This type of solenoid gives an exceedingly high pull with a minimum of weight.

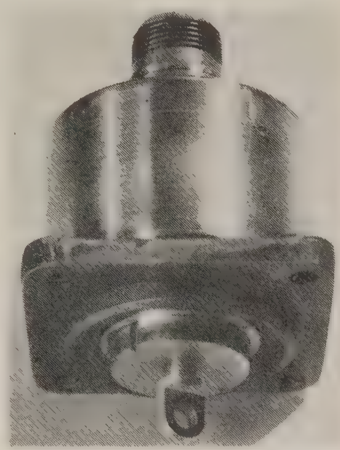

Fig. 1 First Solenoid Design; Weight 21 Oz

Fig. 2 Improved Solenoid Design; Weight 15 Oz

Having developed a suitable solenoid, a valve had to be designed to utilize the high-pull and short-stroke characteristics of the solenoid. One of the first solenoid valves built by the company used a rocker arm but did not require a neutral position. A 30-lb spring loads the rocker arm in one direction; the solenoid pulls 60 lb, thus 30 lb is available in both directions.

Fig. 3 indicates a valve having a rocker arm operated by dual

[1] Chief Engineer, Adel Precision Products Corporation.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

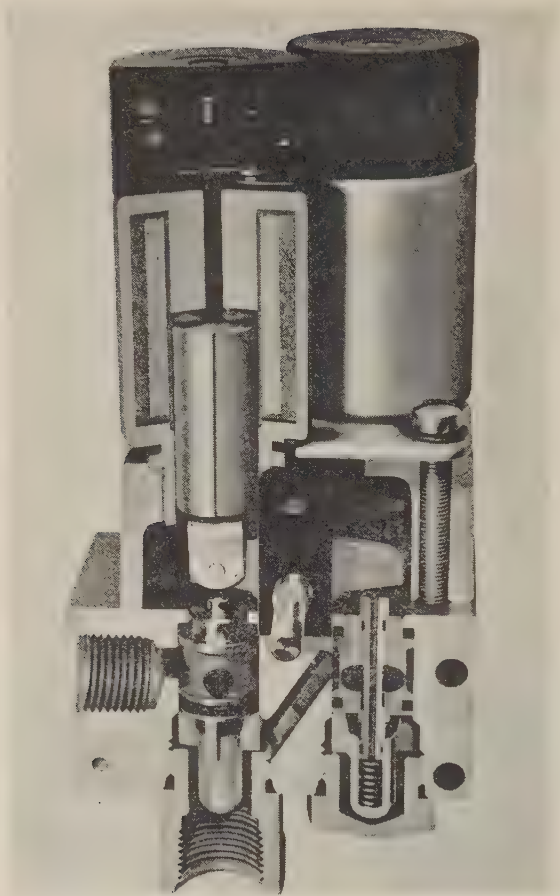

Fig. 3 Solenoid-Operated Selector Valve, Incorporating Rocker Arm

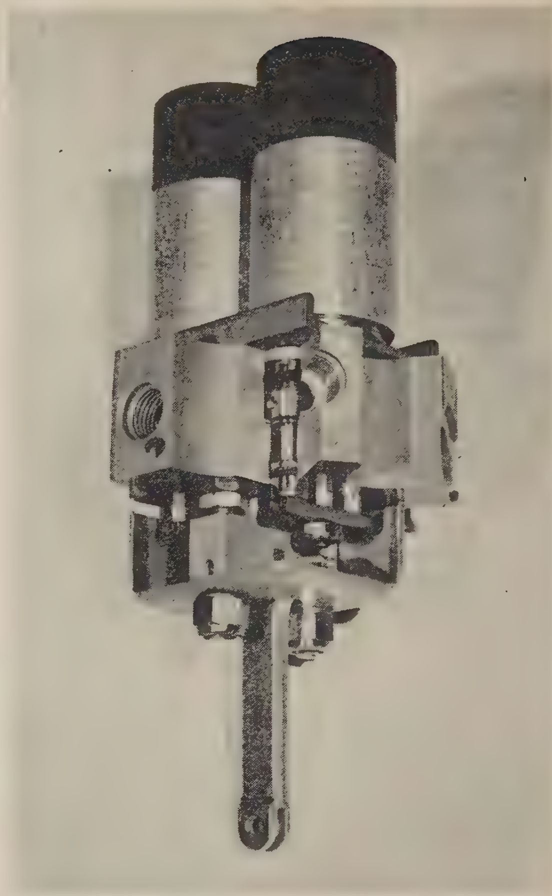

Fig. 4 Solenoid-Operated Selector Valve; Crossbar Replaces Rocker Arm

solenoids. This valve requires a neutral position; therefore two of the double-coil solenoids are utilized with one solenoid being energized for one position and the other solenoid for the other position. When neither is energized the valve is in neutral. On this valve an emergency override lever is added to allow for mechanical actuation in case of electrical failure.

In order to simplify design, over-all cost, and weight, a third type of actuation, as in Fig. 4, was developed which eliminated the need for a rocker arm. Because of the pressure and return-poppet arrangement it is possible to get a 4-way-valve action by lifting two adjacent poppets at a time; thus as shown in this illustration, a crossbar is put between these two adjacent poppets with the solenoid pulling on this crossbar. On this type of design manual actuation is easily provided.

Various types of valves for operating pressures of 1500 and in some cases 3000 psi, including the shutoff, 3-way and 4-way, have been designed for solenoid actuation having single solenoids, utilizing spring returns or double solenoids with and without rocker arm.

Electric-Motor-Actuated Valves. The advent of small inexpensive electric motors allows motor-operated valves to become competitive with solenoid-operated valves. Although the cost of the motor-operated valve is slightly higher at present, proper production engineering on either motor or solenoid valves may give either the cost advantage. Over and above cost consideration, performance requirements may dictate the use of either mode of actuation in that both have advantageous characteristics for particular applications.

Solenoid actuation has the advantage over motor actuation in conventional selector valves requiring a neutral position. Limit switches are required to obtain a neutral position in motor-operated valves; whereas, it is a natural function with solenoid-operated valves. Obviously, this advantage is lost to solenoid actuation on valves requiring no neutral position.

Emergency manual override is easier to obtain on the solenoid-operated valves, due to the free movement allowed in a solenoid when the current is not energized. A manual override on a motor valve leads to complications in design. Solenoid valves have the advantage where fast actuation is required, operating in a fraction of a second, whereas the time of motor-operated valves is usually measured in seconds. This applies, however, to direct-connected conventional poppet-design solenoid valves and not to pilot-operated valves which are inclined to be somewhat slow and erratic in operation.

Extremely high starting loads, usually obtained in the larger valves, are handled more effectively by motors which give out high initial torque as against the low initial torque of a solenoid.

A motor-operated valve can be readily designed to stay in

either extreme position without being energized, whereas a solenoid requires continuous energizing to hold its position, unless locking mechanism is provided.

In the weight-comparison graph shown in Fig. 5, solenoid valves are seen to be lighter in the smaller sizes than motor-operated valves, with the lines crossing slightly above the $^1/_2$-in-size valves. Above $^1/_2$-in-line size, the motor-operated valves have definite weight advantages.

Power Package. The power package, a combination of electrical and hydraulic units, has provided the industry with another unit of extreme versatility with weight saving. Several years ago, when the need presented itself for a valve actuator required to hold the valve in extreme positions for a protracted period of time, the device shown in Fig. 6 was developed. An electrical-mechanical actuator, originally contemplated, presented overheating problems when held in a stalled position. By utilizing a small electric-driven gear pump, taking fluid from the return side of the valve, a simple reversible pump-and-cylinder combination was worked out, providing easy actuation of the valve with a minimum of weight. No overheating problems were present due to the built-in leakage in the gear pump and the cylinder.

This device led to the development of a power package for actuation of landing gears and flaps for light aircraft. This unit combines a motor-driven gear pump, submerged in the hydraulic reservoir; the reservoir is surrounded by a glass tube, allowing visual inspection; an integral selector valve allows positioning; by-pass valve avoids any possibility of overflowing the reservoir; and a built-in relief provides adequate thermal protection. This unit weighing $4^3/_4$ lb, dry, has an output of $^1/_2$ gpm at 300 psi operating pressure.

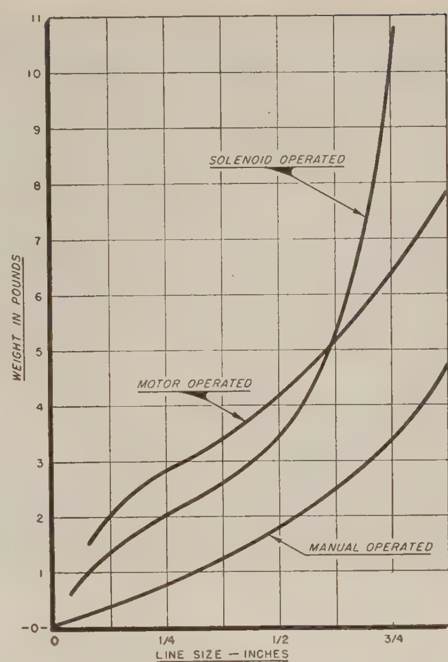

Fig. 5 Relative Weights
(Manual, solenoid- and motor-operated 4-way selector valves.)

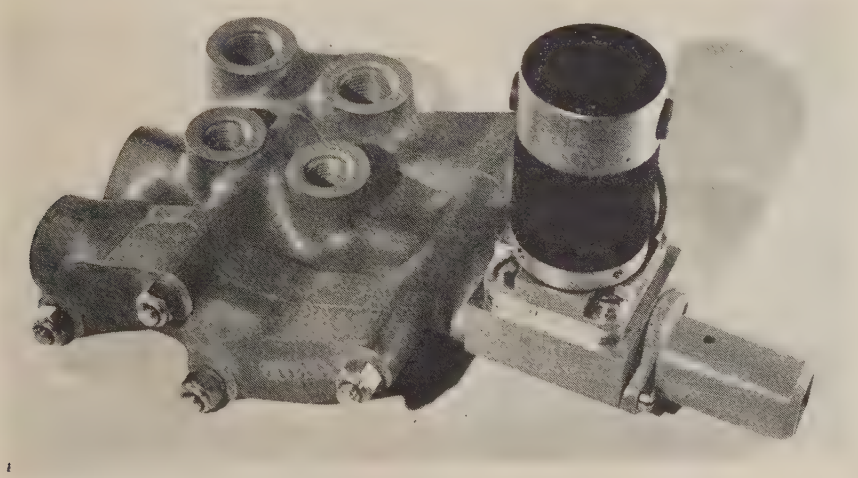

Fig. 6 Electrical-Hydraulic Actuator

Later requirements for a heavier-duty unit dictated a slightly sturdier design as shown in Fig. 7. Designed primarily for emergency power stand-by, it incorporates a motor, pump, reservoir, relief, and pressure and return ports. The glass-reservoir cover is replaced by a casting, with fluid level being easily determined by the use of a dip stick. Provision is made for the adaptation of the Adel Stacking Midget[2] (a manifold-type selector valve), thus a package unit can be obtained incorporating a motor, pump, reservoir, relief valve, auxiliary pressure and return ports, and/or 1, 2, 3 or more selector valves. This unit gives 800-psi output at 0.7 gpm with $^1/_3$ gal capacity for 11 lb weight, dry.

Another problem, somewhat different from the others, led to the development of the pump-and-tank combination which forms the basis of a power package. Difficulties in pumping oil at 35,000 to 40,000 ft altitude led to this particular pump design, which shows no cavitation effects at high altitude under high- or low-temperature conditions. The unit has a pump, motor, reservoir, with a built-in relief; selector valve, and other hydraulic components can be added to produce a power package which gives 6-gpm flow at 1000 psi with a weight of 37 lb. The

[2] "Hydraulic Control Standardization," by J. W. Kelly, *Aero Digest*, vol. 41, Dec., 1942, pp. 211–212, 215–216, 218, 276, and 279.

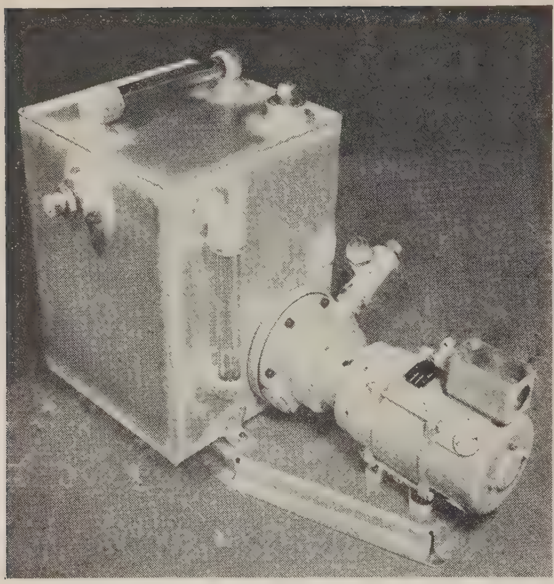

Fig. 7 High-Altitude Pump

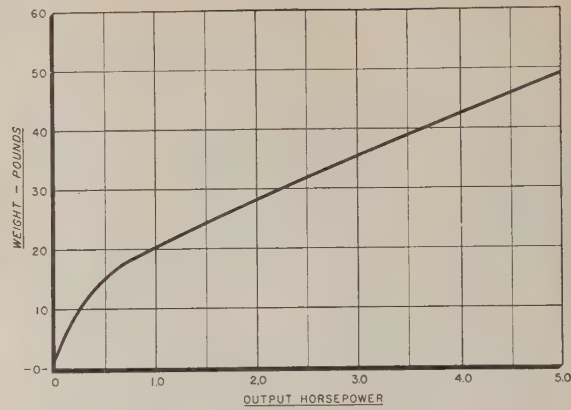

Fig. 8 Weight of Electrohydraulic Power Packages at Various Horsepowers

weights of the power packages are plotted against output hydraulic horsepower in the curve, Fig. 8. It is interesting to note the slope in the horsepower ratings below the 1-hp point.

Power Position Controls

The need for combination controls is definitely brought out in the field of the power position control. Utilizing hydraulic power as the desirable power source, three different types of follow-up systems which allow the utilization of efficient power position control will be discussed.

Mechanical Follow-Up. A simple mechanical follow-up system applied to hydraulic 4-way valves is shown in Fig. 9. We have developed several planetary-gear follow-up systems; but find that this walking-beam system, through sheer simplicity, has much to offer. The walking beam provides a simple follow-up mechanism on a sensitive rocker-arm-type selector valve. Movement of the master or control lever, opens the valve which in turn operates the power cylinder; displacement of the follow-up mechanism shuts off the valve when the cylinder reaches the désired position. Being of a simple and straightforward type, motoring and hunting and other difficulties usually present in power position controls, are absent.

Hydraulic Follow-Up. Another method, shown schematically in Fig. 10, utilizing the "Adel isodraulic" system, has a hydraulic follow-up. It offers many advantages in weight saving, installation, simplicity, and smooth operation. It consists basically of three cylinders in parallel; cylinder 1 forming the master unit, cylinder 2 the valve-operating unit, and cylinder 3 the follow-up unit.

Working on the principle that with three cylinders in parallel, one being held, the second being moved, the third will move an amount equivalent to the first. Thus the follow-up cylinder is held by the power cylinder. As the master cylinder is moved, the second cylinder operates the 4-way valve which in turn applies pressure to the proper side of the power cylinder. Movement of the power cylinder operates the follow-up cylinder which automatically shuts off the valve when the desired position is reached.

Electrical Follow-Up. Utilizing the electric control on hy-

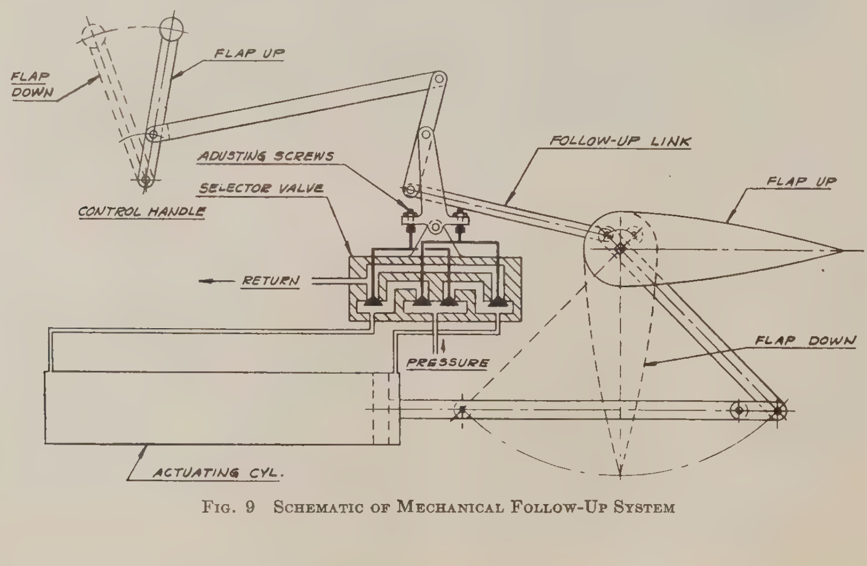

Fig. 9 Schematic of Mechanical Follow-Up System

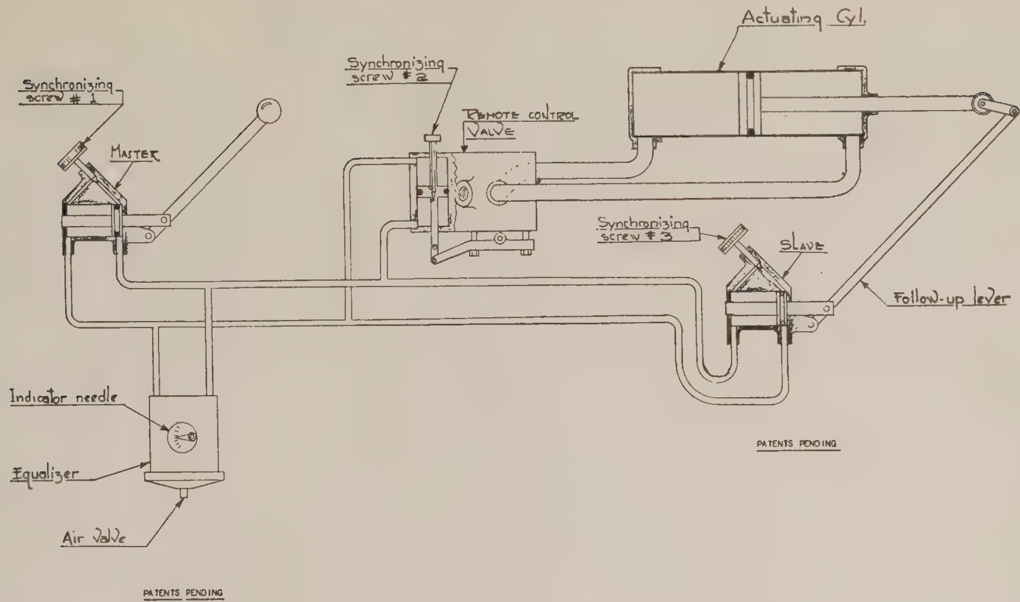

Fig. 10 Schematic of Hydraulic or Isodraulic Follow-Up System

draulic power, through the medium of a solenoid 4-way valve, another type of power-position-control combination is presented. Fig. 12 shows such a hydronic follow-up system. The master potentiometer upsets a bridge relationship with the follow-up potentiometer; thus the relay control box actuates the solenoid valve which in turn allows the cylinder to go to the desired position at which the follow-up potentiometer shuts the valve off.

"Hydronic" system, for long distances, is the lightest of the three, as shown in the graph, Fig. 13. Up to around 50 ft distance, the isodraulic position control enjoys weight advantages; and below 20 ft distance the mechanical system is the lightest.

Other factors, however, invariably enter into the follow-up choice. Actual position indication and system "feel" are naturally obtained in the mechanical and isodraulic; whereas, no feel is obtainable in the hydronic. Long distances are easiest to install with the hydronic, with the isodraulic coming second, and the mechanical third. The isodraulic and mechanical follow-up systems are self-contained and not dependent upon any outside source of power for actuation and followup; whereas, electrical power is required for hydronic operation. Multiple installations are easier to obtain electric-wise, requiring only slight revisions in circuits. For utmost simplicity and comprehension, when servicing in the field, the mechanical would rate first; with isodraulic second; and hydronic third.

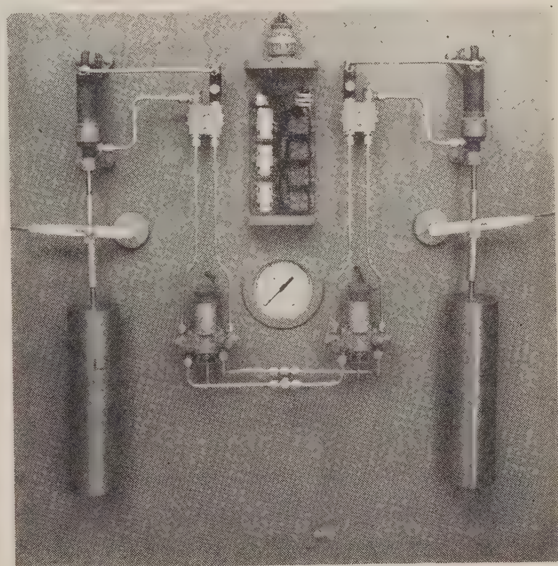

Fig. 11 Electrical or Hydronic Follow-Up System

Conclusion

Combination controls have presented many features which have and will be used to weight- and cost-saving advantages in the future as well as in the past. Actual manufacturing experience and operation of solenoid valves over the past several years have given much desirable and worth-while knowledge which allows us to design into these units more dependability and ruggedness than originally conceived.

Utilization of small electric motors, currently being supplied on other aircraft devices, has allowed us to design and build dependable motor-operated valves. Both solenoid- and motor-operated valves appear to be permanently in the picture and give the aircraft designer a wide choice of characteristics from which to choose.

The power package is beginning to come into its own; and as further improvements are made, weight-savings and operational advantages will become more and more apparent.

New methods and advancements of old methods on follow-up systems for hydraulic controls, utilizing basically mechanical, hydraulic, and electrical combinations, give to the industry versatility with smooth hydraulic power.

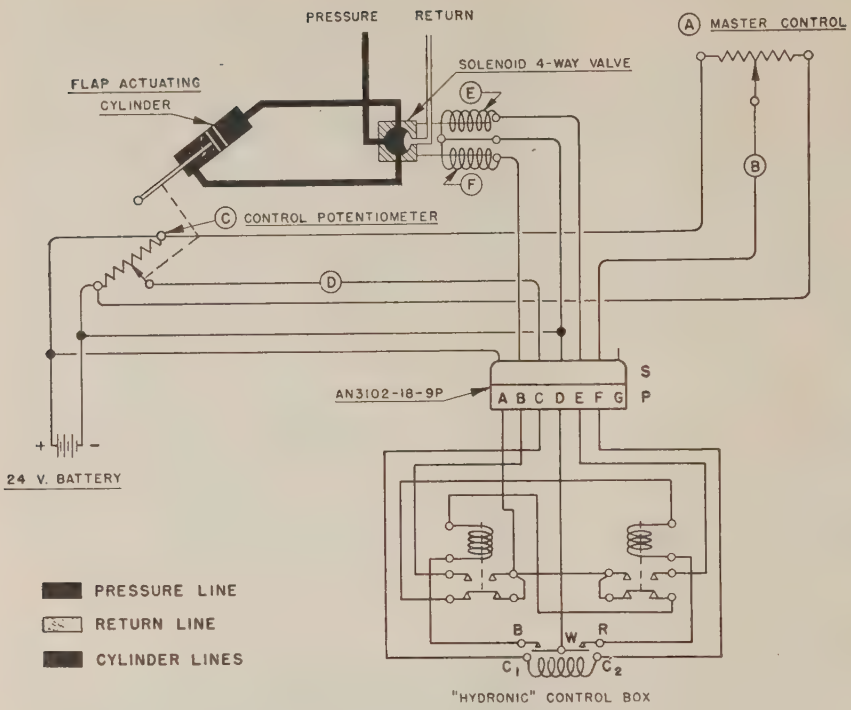

Fig. 12 Schematic Diagram of Follow-Up System Shown in Fig. 11

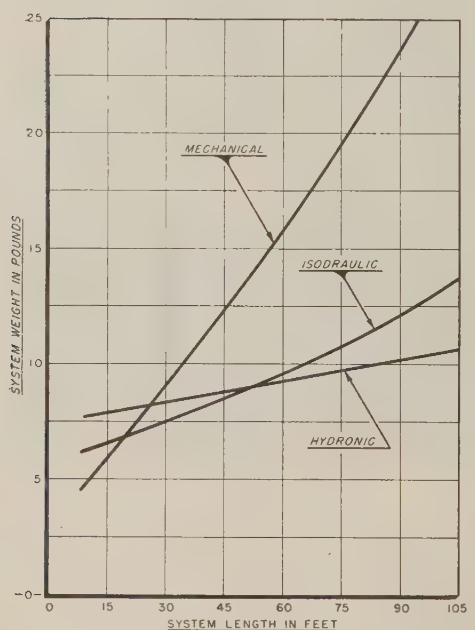

Fig. 13 Relative Weights of Power-Position-Control Systems

Discussion

R. A. Gund.[3] In discussing combination controls for aircraft, the author assumes that hydraulic power is the desirable power. It should be pointed out that for minimum weight the desirable power source may be either hydraulic or electric, depending on the characteristics of the load. Another factor which must be evaluated is the power supply on the aircraft—is it principally hydraulic or electric?

On large airplanes, where the power applications are generally located at some distance from one another, hydraulic lines are more difficult to run than electric wires. In military craft hydraulic lines are much more vulnerable.

The most versatile system would be one employing electric distribution with either electric or hydraulic power applied at the load, depending upon individual requirements. If, for instance, the load requires a starting torque of less than 6 times rated torque, an electric motor should be used to drive the load directly. This will result in minimum weight and maximum efficiency. On the other hand, if a very high torque load must be driven, hydraulic power can be advantageously applied by the use of the combination power package.

As an example, a split series-wound motor driving a screw jack could be directly substituted in place of the solenoid 4-way valve and hydraulic cylinder shown in Fig. 12.

Hydraulic power is most suitable for storing energy, as in power-boost systems and for hydraulic brakes.

Thus we may conclude that for minimum weight and maximum dependability all factors should be considered, and hydraulic power used only if it gives the most desirable results.

Author's Closure

In this paper the author has tried to avoid the controversial argument of hydraulic versus electrical modes of control. However, the author's assumption that hydraulic is the more desirable power source is further emphasized by Mr. Gund, and the arguments on the relative weight values of hydraulic and electric systems should be considered along with the many other factors involved before deciding which is the more desirable as a system.

[3] General Electric Company, Schenectady, N. Y.

The Effect of Measurement Dead Time in the Control of Certain Processes

By D. P. ECKMAN,[1] PHILADELPHIA, PA.

The purpose of this paper is to report on a series of automatic-control tests which demonstrate the importance of instantaneous controller-mechanism response in self-balancing potentiometer controllers. The automatic-control tests were made with both two-position and proportional-reset control. Other tests, dealing with the importance of measurement dead zone of the controller, are included, noting the relation between dead zone of measurement and dead time of controller response.

Introduction

One of the most important factors in automatic control of industrial processes is the manner in which the controller mechanism responds to changes in the measured variable. This measuring lag, as it is often called, forms an extensive part of the input-output controller-response relationship.

Generally, the factors of measurement dead time and dead zone are neglected in process-control analysis, particularly in mathematical analysis where their inclusion results in complications of a high order. Therefore it seems desirable to investigate the effect of these factors in automatic control in order to aid in the application of theoretical or empirical data to control problems.

Potentiometer controller mechanisms which incorporate continuous rather than periodic principles of rebalancing provide an opportunity for comparison test of these types of measuring means. Under simulated control conditions it is possible to analyze the dynamic action of the control system and the effectiveness of continuous measurement.

The measuring lag of a temperature controller is composed of two main parts: the lag of the primary element, and the lag of the controller detecting mechanism. It is this second part of the measuring lag, that due to the detecting mechanism, which we wish to study. For purposes of discussion we may classify the many methods of operation of self-balancing potentiometer controllers under two general types, i.e., periodic action and continuous action. A periodic-action potentiometer (recording type) is generally identified by a periodic mechanical system for detecting the position of a galvanometer pointer. When an unbalance of voltages is produced by a change in measured temperature, the detecting mechanism determines the amount of unbalance and drives a voltage divider or slide wire to balance against the new value of measured voltage. The periodic-action potentiometer is in wide general use for both measurement and control of process variables.

The continuous-action potentiometer (recording type) is generally identified (1)[2] by the use of continuous uninterrupted drive means, usually electronic, for obtaining the balance of measured against standard voltages. The use of continuous action in potentiometer balancing eliminates any lag due to a rebalancing mechanism, and avoids periodicity whereby a small dead time may elapse before changes in controlled temperature can be detected.

Tests With Two-Position Control

In order to determine the effect of dead time in the measuring system, it is desirable to conduct tests of two-position control. By this means all extraneous factors may be eliminated from the tests of automatic control. Processes were selected on the process-analog control board (2) by using various combinations of capacitance, resistance, and dead time.

The reaction curves for the processes to be tested are shown in Figs. 1 and 2. A reaction curve is determined by setting the

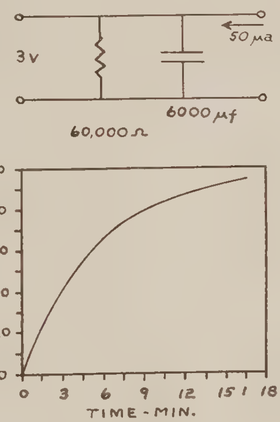

Fig. 1 Diagram and Reaction of Process No. 1

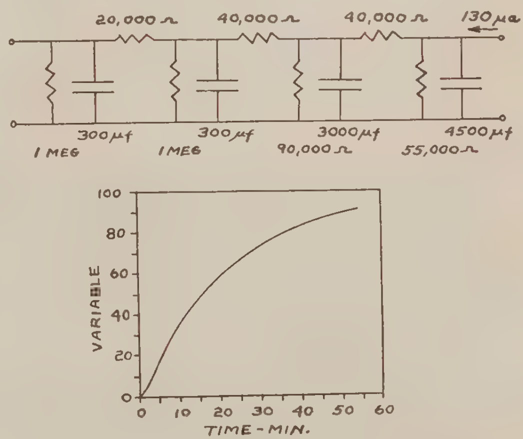

Fig. 2 Diagram and Reaction of Process No. 3

[1] Development Engineer, The Brown Instrument Company. Jun. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Industrial Instruments and Regulators Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

valve position and allowing the measured variable to balance. A small sudden movement of the valve to a new fixed position will result in gradual increase of the measured variable to a new balanced value. The resulting time rate of change of the measured variable is the reaction curve. All processes indicate that self-regulation is present. Although the reaction curves are shown for only one direction of change, it may be assumed that the negative change produces a similar curve.

Process No. 1, shown in Fig. 1, is essentially a single-capacity process with a time constant of about 6 min, which is the time required to achieve 63 per cent of final value. Process No. 2 is identical in characteristics except that a dead time of 0.1 min was introduced between controller action and resultant valve movement.

Process No. 3, shown in Fig. 2, is a two-capacity process and thus involves slight transfer lag as can be noted by the slower initial rate of change on the reaction curve. Process No. 4 is identical in characteristics except that a dead time of 0.25 min was introduced between controller action and resultant valve movement.

Each process analog was connected, in turn, to the potentiometer controller. Each potentiometer controller, a periodic-action type and a continuous-action type, was fitted with both pneumatic and electric two-position "on-off" control. Pneumatic control was accomplished through a pneumatic diaphragm mechanism. The proportional band (throttling range) of the pneumatic on-off controllers was 0.6 per cent of controller scale. The on-off pneumatic controller is not actually a two-position controller but is a proportional controller with a very narrow band. This controller, however, is commonly termed on-off. Electrical control was arranged so that the control contacts directly actuated the energy supply to the process. The differential gap of both electrical on-off controllers was 0.05 per cent of controller scale.

With all process and controller combinations, a cycle of controlled variable was recorded. The resulting control cycle is described in Table 1 which shows amplitude, and Table 2 which shows period of cycle of the controlled variable.

TABLE 1 AMPLITUDE OF CONTROL CYCLE; PER CENT SCALE

Potentiometer controller with	Type of control	Process no. 1	2	3	4
Continuous action	Pneumatic	0.12	4.2	2.2	5.4
Periodic action	Pneumatic	1.80	4.6	4.2	7.1
Continuous action	Electric	0.25	3.8	2.3	6.1
Periodic action	Electric	1.00	4.2	4.0	6.8

TABLE 2 PERIOD OF CONTROL CYCLE; MINUTES

Potentiometer controller with	Type of control	Process no. 1	2	3	4
Continuous action	Pneumatic	0.10[a]	0.28[a]	2.4	4.5
Periodic action	Pneumatic	0.50	0.80	3.7	5.0
Continuous action	Electric	0.03	0.25	2.4	4.7
Periodic action	Electric	0.27	0.67	3.9	5.0

[a] Actually throttling to the extent that valve never completely opens or closes.

The results shown in Tables 1 and 2 illustrate the improvement caused by eliminating the dead-time delay in the potentiometer measuring means. This effect is particularly noticeable in processes Nos. 1 and 3, where all dead time in the controlled system is concentrated in the measuring means. In processes similar to No. 4 where considerable dead time exists in the process itself, the elimination of measurement dead time results in proportionately less improvement.

Dead time existing in any portion of the controlled system results in nearly proportional increase in amplitude. It is obvious that if automatic control is to maintain an adequate process balance then dead time in the controlled system must be reduced to a minimum.

Applying the results of these tests to applications of automatic control, it is probable that in electrically heated furnaces and baths where the process lags are very small, the elimination of measurement dead time would result in much closer temperature control.

On the other hand, when controlling the temperature of such processes as heat exchangers, where the process lags may be of minutes duration, the elimination of measurement dead time may bring no noticeable improvement.

It is interesting to note that the measuring lag may be computed from the results of controlling process No. 1 with the electrical two-position controller. Here virtually all of the lag aside from the time constant of the process is associated with the potentiometer measuring means. Since in two-position control the period of cycling is about 4 times the total lag (3), then the measuring lag should be the period divided by 4.

Thus the measuring lag calculates to be about 0.0075 min for the continuous-action potentiometer measuring means and about 0.0645 min for the periodic type.

Tests With Proportional-Reset Control

In order to demonstrate the effect of dead zone in the measuring means, a controller may be applied to a process, and the deviation during a recovery from a load change may be expected to show differences depending upon the magnitude of dead zone (4). The dead zone is generally defined as the greatest range of scale values within which changes in value of the controlled variable are not detected.

With self-balancing potentiometers, the width of dead zone is generally related to the construction of its detecting and balancing mechanisms. As one contributing factor, the width of dead zone is inversely proportional to the number of active convolutions of the potentiometer rebalancing slide-wire since the controller pen and controller mechanism are generally positioned to the convolution of the slide-wire nearest the balance point.

The self-balancing continuous-action potentiometer used in the previous tests has 1600 active convolutions on the slide-wire (1). It was desired to test this same potentiometer with 800 and 400 convolutions on the slide-wire, thus multiplying the dead zone by 2 and 4, respectively.

These arrangements were made by altering the mechanical relation between full-scale pen motion and slide-wire contactor travel. An oversize drum on the slide-wire driving shaft reduced the amount of slide-wire contactor motion while maintaining the same pen motion. This change does not alter the responsiveness of the potentiometer balancing motor to a given unbalanced emf, but reduces the amount of slide-wire contactor travel for the same pen travel when a rebalancing action occurs. The electronic voltmeter on the process-analog control board was then readjusted to maintain the same range of 0 to 5 volts for full-scale pen travel.

The net result of these changes is to alter the number of slide-wire convolutions corresponding to a given pen travel while maintaining the same dynamic balancing action and the same scale calibration.

Three processes were selected and are described in Figs. 3, 4, and 5. Process No. 5, described in Fig. 3, is a relatively simple process having a moderate reaction rate and slight transfer lag Process No. 6, described in Fig. 4, possesses a moderate reaction rate and appreciable transfer lag. Process No. 7, described in Fig. 5, has a fast reaction rate and slight transfer lag. In all tests only the dead time inherent in the system was present and no dead time was intentionally added.

Each process analog was connected to the potentiometer controller and optimum adjustments of proportional band and reset rate selected (5). A pneumatic proportional-reset type con-

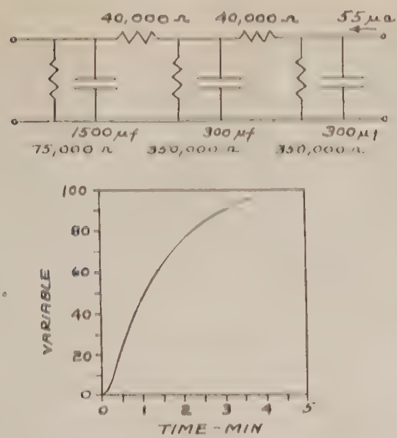

Fig. 3 Diagram and Reaction of Process No. 5

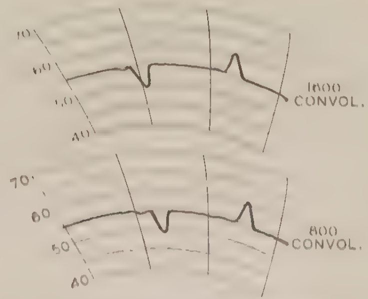

Fig. 6 Recovery Curves for Process No. 5
(45 per cent band, 0.8 per min reset rate. 1 time division equals 4 min.)

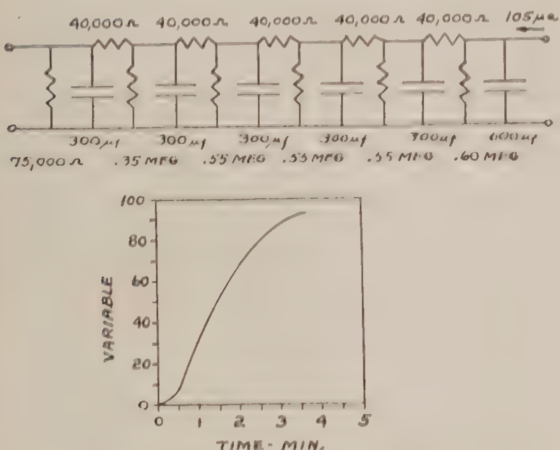

Fig. 4 Diagram and Reaction of Process No. 6

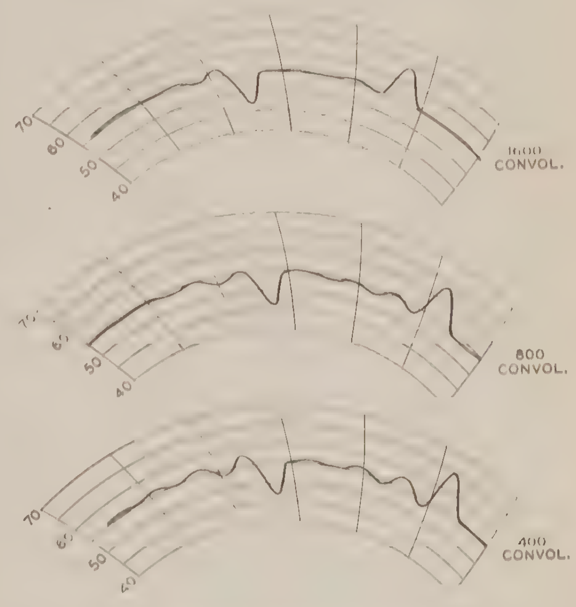

Fig. 7 Recovery Curves for Process No. 6
(60 per cent band, 0.4 per min reset rate. 1 time division equals 4 min.)

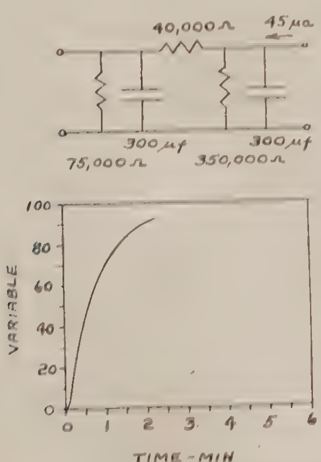

Fig. 5 Diagram and Reaction of Process No. 7

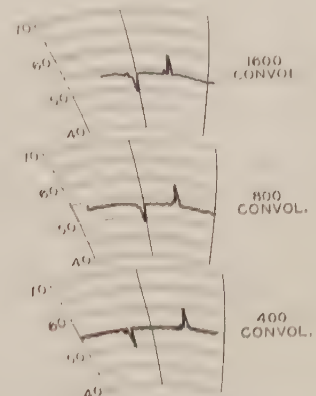

Fig. 8 Recovery Curves for Process No. 7
(25 per cent band, 3 per min reset rate. 1 time division equals 4 min.)

troller was employed for these tests. After the controlled variable had stabilized, a change in supply of energy to the process was made and the controller recovery curve obtained.

The supply change was introduced by adding a bias voltage to the existing voltage supplied to the current-input unit on the process-analog control board. This is, in effect, a change of flow through the control valve caused by a variation in pressure drop and requires that the controller correct for this change. The control valve has 200 effective positions.

The charts showing the recovery curves are reproduced in Figs. 6, 7, and 8. The results of these tests are given in Table 3.

TABLE 3 RESULTS OF TESTS

Process no.	Prop. band, per cent	Reset rate per min	Maximum deviation for ± supply changes — No. of effective convolutions: 1600	800	400
5	45	0.8	11.2	11.7	..
6	60	0.4	16.7	17.7	18.1
7	25	3.0	10.0	10.4	10.6

As shown by the recovery curves in Figs. 6, 7, and 8, both an increase and a decrease in supply were made. The increase in supply results in a rise of the controlled variable before it is corrected by the controller, and a decrease in supply results in a fall of the controlled variable before it is corrected. The maximum deviation was found by adding both the positive deviation and the negative deviation. In this manner various small nonlinearities in valve characteristics, process time-constant, and process lag may be averaged.

The difference between the various maximum deviations from a supply change may be attributed to the different widths of measurement dead zone in each test. With a larger dead zone, a short period must elapse between the time when an actual change in measured variable begins and the time when the controller senses the change. Thus a measurement dead zone creates dead time. This delays the controller corrective action and allows greater deviation.

The stability of control appears to be more cyclic when the dead zone is greater. With process No. 6 particularly, the recovery curve shows that the proportional band should be slightly increased in order to maintain the same ratio of succeeding amplitudes of cycling. There is an appreciably consistent lengthening of period when the dead zone is greater. The apparent decrease in stability of control and the lengthening of period of control points to the conclusion that an additional lag exists when the measuring means of the controller possesses a finite dead zone.

Conclusion

The brief test results given here illustrate the importance of measuring lag and measurement dead zone in automatic control. It is desirable to maintain measurement dead time as small as possible since it increases the quality of control, especially on such processes where all other lags are small. It is desirable to maintain measurement dead zone as small as possible since it increases the quality of control by accomplishing a reduction of dead time.

It is evident that more investigation is required to establish further the relationship between each component part of measuring lag and the measurement dead zone. In most control problems the effect of these factors cannot be ignored because of their influence on the quality of automatic control.

Acknowledgment

The author gratefully acknowledges the suggestions of Mr. T. R. Harrison in the investigation.

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Discussion

W. F. Hickes.[3] The author has shown us the detrimental effect of "dead time" in process control and has presented some evidence on the effects of "dead space" in terms of slide-wire turns. The results are interesting because they confirm the conclusions reached in our own laboratory investigations and field work.

It is difficult in practice to speak of dead time without bringing in dead space and vice versa because the two are so closely related. Let us assume we have a furnace at 1500 F. Let us further assume that with full heat on we have a heating rate of 5 deg F per min. Now if we have an instrument with a 0 to 1600-deg scale range and 1600 slide-wire turns, each turn will represent 1 deg, that is, the instrument will have 1 deg dead space. With a 5 deg per min change of temperature, the instrument should move every $^1/_5$ min and can hardly be much better than a periodic instrument with a dead time of $^1/_5$ min and could be expected to be inferior to a periodic instrument with the usual period of $^1/_6$ to $^1/_{10}$ min. It might be added that the figures quoted represent an actual furnace installation.

In comparing measuring instruments the author has described two types, the so-called continuous and the periodic. Actually there are at least four types. There is the periodic type described by the author, which has dead space due to slide-wire turns and dead time due to its periodic balancing action. There is another type which has dead time but not dead space, since the slide-wire serves only to set the control point, and the control mechanism operates directly from the galvanometer deviation. This type is nonrecording but has been very successful as a controller. There is the "continuous" type, cited by the author, which eliminates dead time by its continuous action but which still has the dead space due to slide-wire turns. There is finally a fourth type which has inherently neither dead space nor dead time, the dead time being eliminated by the same type of continuous balancing as in the previous case, dead space being eliminated by the substitution of stepless balancing for the conventional slide-wire.

The author has stated that the elimination of dead time is of greatest value where process lags are small and may bring no noticeable improvement where lags are large. This is undoubtedly correct although it should be realized that the typical thermal processes of inconsequential lags such as heat-treating furnaces have in general relatively slow maximum rates of temperature change so that dead time may be much less objectionable than dead space.

The really interesting cases would appear to lie in the extension of electronic controls of these general types to problems involving pressure and flow control. There are at present several makers offering units to convert pressure and differential pres-

[3] Foxboro Company, Foxboro, Mass.

sure into electrical values such as resistance which can then be measured on the type of instruments described. Since pressure and flow controls are normally characterized by very small lags and in many cases also by very rapid changes, the value of a truly continuous instrument free from inherent dead space as well a dead time should be apparent.

Author's Closure

The discussion of Mr. Hickes is greatly appreciated since it brings out many of the factors affecting the quality of automatic control when controller dead zone is present. As is stated, in control problems of inconsequential lag measurement, dead time becomes an appreciable factor.

Mr. Hickes's conclusion with regard to dead time due to dead zone is open to controversy because having a slidewire in which one convolution represents an interval of 1 F is not identical to having a dead zone of 1 F. Without going into the many details involved, a high amplifier "sensitivity" produces a condition wherein a rise in thermocouple emf will cause the slidewire contact to move entirely across one convolution to the edge of the adjacent convolution. Thus with gradually changing temperatures it is possible to obtain a relatively greater response with a slidewire balancing device having discreet positions than with a continuously balancing device.

With respect to measurement and controller dead zone, there are four general cases: (*a*) Control systems and process with a relatively fast reaction rate and little lag, (*b*) with relatively slow reaction rate and little lag, (*c*) with relatively fast reaction rate and large or appreciable lag, and (*d*) with relatively slow reaction rate and large or appreciable lag. The effect of dead zone is different in each case.

For example, in process control with fast reaction rate and little lag such as in flow or speed control, the author has observed cases of pseudo stability at narrow proportional bands which, with large, sudden upsets become cyclically unstable, sometimes violently so. This passage from the stable to the unstable region is apparently initiated by a fast change of the controlled variable through the dead zone and there may be little subsequent evidence of a square top wave.

With processes having a slow reaction rate and appreciable lag, the dead zone evidences itself as a consistent, additional lag and the quality of control is proportionately reduced.

Mechanical Oscillators and Their Electrical Synchronization

By S. W. HERWALD,[1] R. W. GEMMELL,[2] and B. J. LAZAN[3]

Mechanical oscillators provide an excellent means for obtaining structural serviceability tests. The adjustable-while-running type is particularly flexible. With it the answers to certain fundamentally difficult problems can be obtained quickly. The electrical system presented provides accurate remote control of frequency and phase angle between forces of two or more mechanical oscillators. In addition, means are incorporated for changing and indicating remotely the amount of unbalance of each oscillator unit. Rototrol units used with electronic and magnetic controls are the basic elements of this precise servomechanism.

HIGHER speeds in modern transportation and machinery, and present-day design trends toward greater efficiency and lighter weight have emphasized the importance of dynamic forces as a factor in design. The urgency of dynamic testing of materials and structures is indicated by a few recent studies (1),[4] which show that very few service failures in machine and structural members can be attributed to static forces alone; over 80 per cent involve dynamic forces.

Practical mechanical testing may be of three types, as follows:

(*a*) Testing carefully prepared specimens under simplified conditions of stress and environment to determine basic material properties.

(*b*) Testing actual shapes, assemblies, or structures under conditions which closely simulate actual service to secure direct design data.

(*c*) Testing in actual service.

Theoretically, if basic material properties and service conditions are known, one should be able to predict the behavior of a structure. Reasonable success has been achieved in such analysis of structures under static loads. However, predicting the behavior of a structure under dynamic loads in this manner is often unreliable because of such difficult variables as stress concentration, natural frequencies, damping, load distribution, etc. Thus past experiences indicate that at present the best procedure is to use specimen testing as a guide to the selection of materials and as an aid during the initial design stages; but to rely on structure testing for refining a design and checking actual serviceability.

Of course, no test of serviceability is as foolproof as actual service. However, to rely exclusively on actual service is usually impractical, time-consuming, and very expensive. Thus the simulated service test is a vital link in modern engineering analysis.

One of the most useful tools in simulated service tests is the mechanical oscillator, a device for producing sinusoidal alternating force or torque of controlled magnitude, direction, and frequency. Generally, it is a portable machine which may be carried to the structure and attached to it for field testing. Oscillators of this type are particularly useful in producing sinusoidal forces of large magnitude at relatively low frequencies. Furthermore, two or more of these oscillators can be connected electrically so that a definite phase relationship of the force outputs can be maintained. This provides the possibility of a more complete test, because a number of definitely related forces can be introduced to the structure simultaneously.

The Mechanical Oscillator

Probably the first mechanical oscillator capable of producing linear sinusoidal force was developed by W. Spath (2) in 1928, for testing railroad bridges. This oscillator utilizes the centrifugal force of eccentrically supported rotating masses as a source of alternating force (see Fig. 1). An eccentric E is attached to each of two shafts S which are motor-driven and geared so as to rotate in opposite directions. When both eccentrics E point vertically downward, position a in Fig. 1(a), the radial centrifugal forces add and the resultant force is vertically downward. After 90 deg of rotation to position b, the two centrifugal forces point in opposite directions and cancel each other. When the eccentrics reach position c they again add to produce a vertically upward force, and at position d they again cancel. In general, the net force is the vector sum of the two radial centrifugal forces, which is a sinusoidal alternating force such as shown in Fig. 1(b).

The same oscillator may be arranged to produce pure torsional vibration about an axis perpendicular to the plane of the paper at b in Fig. 1(c), by driving shafts S in the same direction and rearranging the eccentrics as shown.

The frequency of the alternating force equals the frequency of rotation of the oscillator, and the magnitude of the force equals the total inch-pound unbalance in eccentrics E multiplied by the angular velocity squared. The value of the unbalance can be adjusted by changing either the radial location or the size of eccentric E.

Variations of the Spath unit have been developed using four (3) or six (4) eccentrics, or using a different arrangement for the eccentrics so as to permit greater flexibility in governing the direction of types of possible forces. Oscillators have been built (4) capable of producing alternating forces as high as ± 44,000 lb; others have unbalanced masses with eccentricity as high as 6800 in-lb.

In many types of tests it is imperative that the weight of the oscillator be kept to a minimum. Mechanical oscillators developed to meet this requirement include the scotch yoke and the hypocyclic oscillator (3). However, none has been as successful as the centrifugal-force type from point of view of simplicity of operation, ease of maintenance, and purity of the sinusoidal force. The scotch-yoke oscillator, for example, generally induces rather severe harmonics which make analysis of results difficult.

During the dynamic testing of a structure it is generally necessary to cover a wide range of alternating forces. In the early os-

[1] Special Products Engineering, Westinghouse Electric Corporation, Pittsburgh, Pa. Mem. A.S.M.E.

[2] Manager, Aircraft Sales, Westinghouse Electric Corporation.

[3] Associate Professor and Director of Testing and Metallurgy Laboratories, Syracuse University, Syracuse, N. Y. Formerly Vice-President and Chief Engineer, Sonntag Scientific Corporation, Greenwich, Conn. Mem. A.S.M.E.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Industrial Instruments and Regulators Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

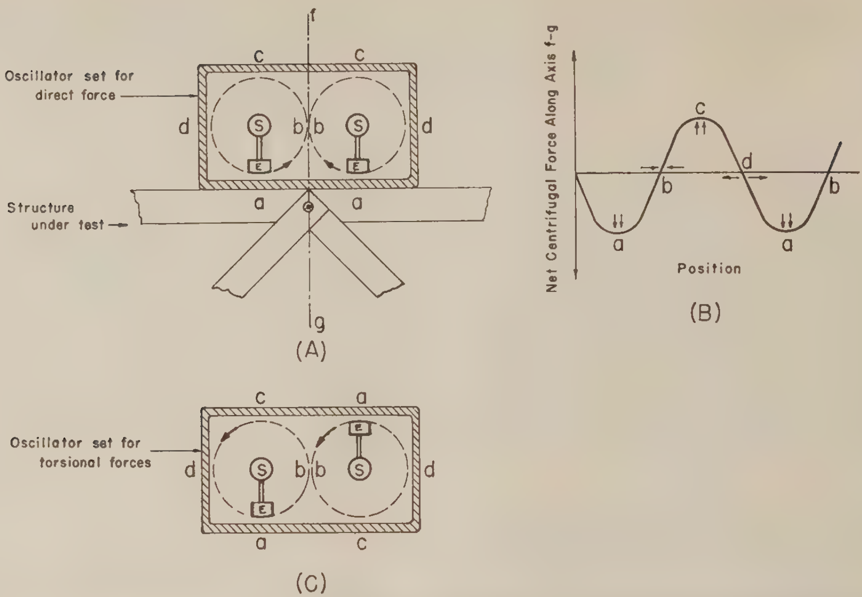

Fig. 1 Principle of Operation of Centrifugal-Force-Type of Oscillator With Two Eccentrics

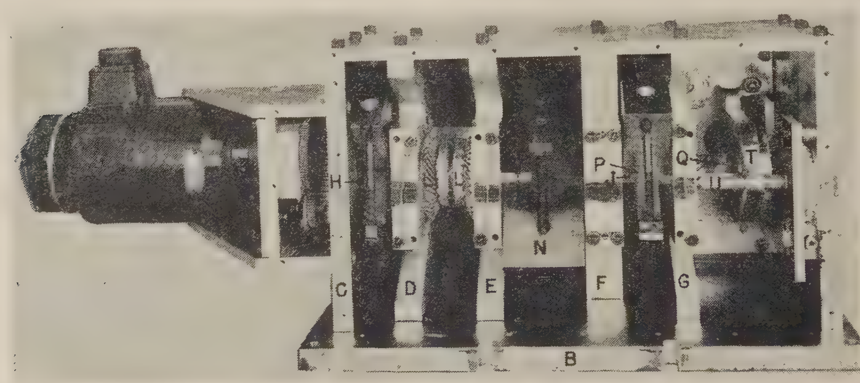

Fig. 2 Adjustable-While-Running Oscillator of the 45-Deg Rack Type With Drive Motor and Synchronization Control Equipment Attached

cillator every change in unbalance required stopping the unit and manually moving or changing the eccentric. This procedure not only consumed time, but also made certain types of tests very difficult. Recent improvements in centrifugal-force oscillators permit stepless adjustment of unbalance without stopping the unit. This may be accomplished either by adjusting a remotely controlled electric drive or by turning a knob attached to the oscillator.

Three different types of unbalance-changing mechanisms have been used: (*a*) differential gear changer (5); (*b*) helical key changer(6), which is now the most popular adjustable-while-running oscillator in use; and (*c*) 45-deg rack changer. Inasmuch as the special electric synchronization equipment to be described later was developed for the oscillator with the rack changer, this unit will now be described in detail.

The principle of operation may be understood by referring to Fig. 2. The oscillator consists of a base plate *B* to which are mounted pedestals *C*, *D*, *E*, *F*, and *G*. The ball bearings contained within the pedestals support rotating shaft *H* between *C* and *D*, rotating shaft *I* between *E* and *F*, and rotating shaft *J* between *F* and *G*. Shafts *H* and *J* are mechanically connected by a shaft running through the center of *I* so that they rotate together at the same speed and in the same direction. Spiral bevel gears *L* connect shafts *H* and *I* so that they rotate in opposite directions but at the same angular velocity. An eccentrically supported mass is attached to each of the three rotating shafts; eccentric *M* to shaft *H*, eccentric *N* to shaft *I*, and eccentric *O* to shaft *J*. At any given setting of the oscillator, eccentric *N* has an inch-pound unbalance equal to the sum of the unbalance in eccentric *M* and *O*. Furthermore, the inch-pound eccentricities in *M* and *O* are inversely proportional to the distances of *M* and *O* from eccentric *N*. This results in cancellation of moments.

Since eccentric *N* rotates in one direction as eccentrics *M* and *O* rotate at the same velocity in the opposite direction, pure sinusoidal force results (see previous explanation of Spath unit). It is possible to produce sinusoidal vibration in any direction perpendicular to the axis of rotation by unmeshing gears *L* (which enables shaft *I* to turn freely without a corresponding rotation of

shafts H and J), turning shaft I as desired, and remeshing the gears L.

The unbalance of all three eccentrics may be remotely and steplessly varied from zero to the maximum value while the oscillator is either stationary or running. This is accomplished as follows:

Each eccentric consists mainly of a large unbalanced mass and two racks P containing square teeth which make a 45-deg angle with the axis of the rack. These straight racks P slide in rectangular slots cut in the three shafts H, I, and J. Contained within these three tubular shafts are round members Q which have 45-deg square teeth cut on two sides to mesh with the 45-deg teeth on straight racks P. Thus as racks Q are moved axially, the meshing teeth will cause straight racks P to move radially, thereby changing the radial location of the center of gravity of the eccentrics. At zero unbalance, racks Q are so located that the center of gravity of the eccentric assembly consisting of P, etc., is at the center of rotation. To increase the eccentricity, the center of gravity of the eccentric is moved radially outward from the center of rotation.

Ball bearings contained within the shafts H, I, and J hold the racks in rigid axial location, but allow them to have angular freedom so that they can rotate with the parts they locate. This axial location is determined by the position of yoke T which is moved by motor-driven worms and screws U to adjust the radial location of the eccentrics.

This oscillator has a maximum unbalance of 150 in-lb and can produce alternating forces up to $\pm$ 1000 lb at speeds as low as approximately 500 rpm. Further refinements incorporated in this unit will be discussed in another section.

Fig. 3 Aircraft Rototrol of 3000 W

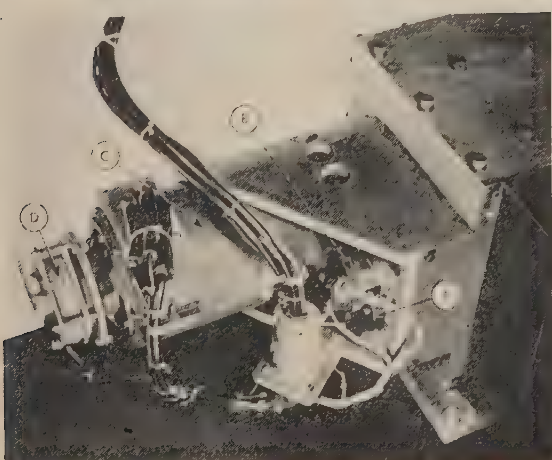

Fig. 4 Drive-Motor End of 45-Deg Rack-Type Oscillator

Electrical Control

The basic electrical equipment used to control accurately the frequency and force-phase relationship of two oscillators of the type shown in Fig. 2 is the Rototrol (7). This machine is illustrated in Fig. 3. When it is used with the proper electronic control, it provides the power required for precise regulation of the oscillator drive motors. The drive motors, shown as C in Fig. 4, rotate the unbalanced masses in the oscillators, producing sinusoidal forces in the manner described in Fig. 1. The motor shown is rated 3 hp at 3600 rpm.

Complete control of both oscillator units is obtained at the control box shown in Fig. 5. From here the speed of the oscillators,

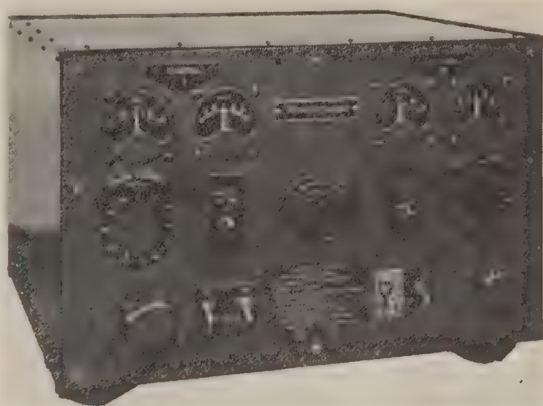

Fig. 5 Front View of Oscillator Control Box

Fig. 6 Front View of Oscillator Indicating Panel

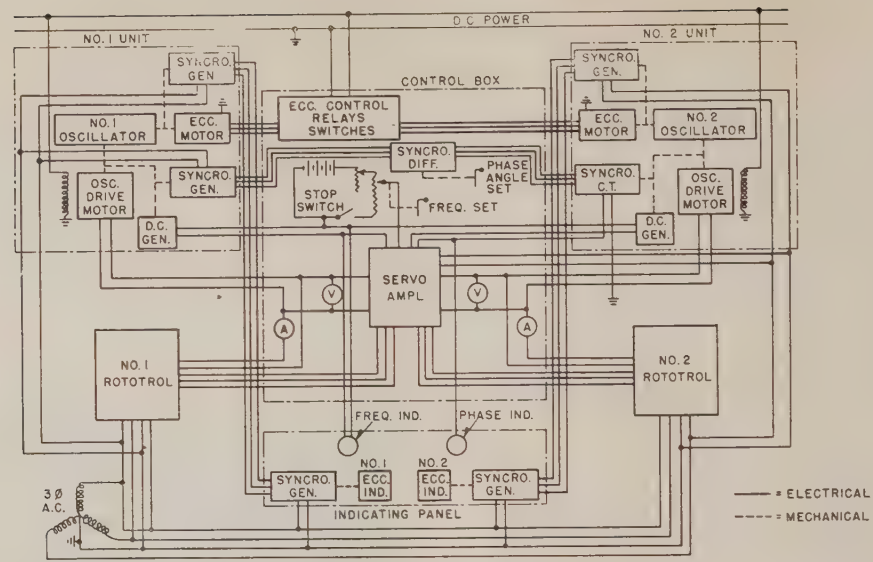

Fig. 7 Schematic of Frequency, Synchronization, and Eccentricity Control of Two Oscillator Units

the phase angle between their force outputs, and the amount of unbalance in each unit can be varied while the oscillators are running. Here also are the normal and emergency braking switches used to obtain decrement curves and emergency shutdown, respectively. Frequency can be fixed at anywhere from 100 to 1800 cycles per min; phase angle between forces can be fixed anywhere between zero and 360 deg; unbalance can be changed from zero to 150 in-lb on each oscillator independently.

Fig. 6 shows the indicating panel associated with this equipment. It contains the test record and is conveniently arranged for photographing. Thus the time of test, the frequency, the eccentricity in each of the units, and the phase error between the units can all be obtained on one photograph. The phase error mentioned should not be confused with the phase-angle setting on the control panel. Phase error is merely the inaccuracy of the phase angle as set on the control unit. The phase-error meter reads directly in degrees. In operation it showed substantially less than ± 15 deg, even with large eccentricities and with the units 180 deg out of phase.

The schematic diagram of the electrical system used to control the oscillators is given in Fig. 7.

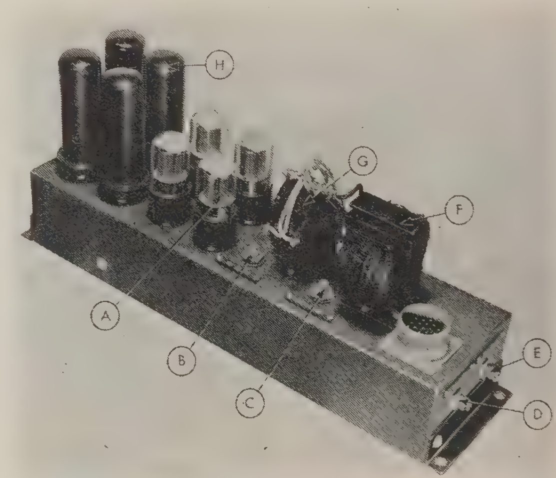

Fig. 8 Two-Channel Servoamplifier

Frequency and Phase Control

For the No. 1, or master unit, a desired frequency is maintained within close limits by matching the preset portion of the control-unit battery voltage with that of a d-c tachometer generator A, Fig. 4, driven by the No. 1 oscillator motor C, Fig. 4. The voltage difference which occurs when the generator voltage does not match the battery potentiometer voltage is fed into the amplifier, Fig. 8. The amplifier output controls the output voltage of the No. 1 Rototrol generator which in turn changes the speed of the No. 1 oscillator drive motor so as to reduce the voltage error between the tachometer generator and the preset portion of the battery voltage. Thus except for a very small voltage difference required to produce enough Rototrol output voltage to maintain the oscillator drive motor at the required speed, the system is self-compensating, tending always to reduce the voltage error. Since a d-c tachometer generator produces a voltage that varies linearly with speed, all one has to do to vary frequency is to adjust the battery potentiometer B, Fig. 9. This, of course, causes the drive motor to change speed so that the tachometer voltage again matches the portion of the battery voltage picked off by the potentiometer.

The No. 2 unit operates in a similar manner, except that there is one added feature. In order to maintain a given phase relationship between the Nos. 1 and 2 units, a synchrosystem is used to detect error from the desired phase relationship. This error is detected as an a-c voltage that varies as the sine of the angle of deviation from the desired phase relationship of the Nos. 1 and 2 units. An electrical schematic of this synchrosystem is shown in Fig. 11. The rotor of the synchrogenerator which has salient poles is excited single phase. This sets up the flux field A which cuts the stator windings. These stator windings are usually displaced mechanically 120 deg and connected to exactly similar windings of the synchrodifferential. The currents that flow in similar windings such as a and a' or b and b' of the generator stator and the differential rotor are identical. Therefore flux field B has the

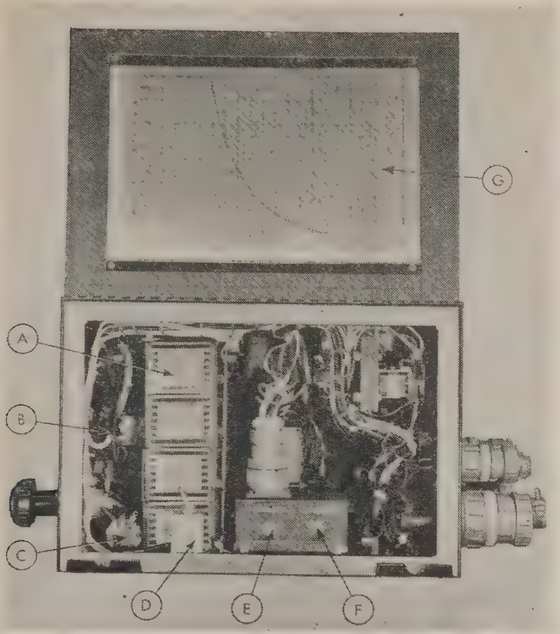

Fig. 9 Right-Side View of Oscillator Control Box

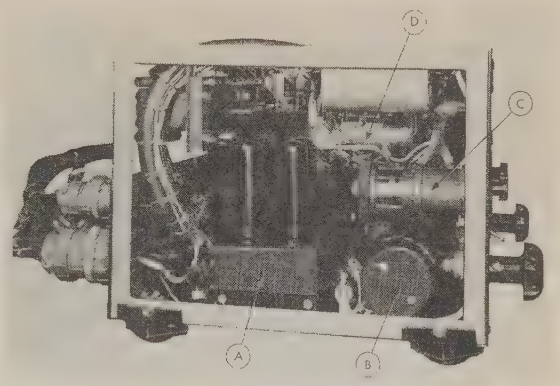

Fig. 10 Left-Side View of Oscillator Control Box

same position relative to the windings as flux field A and the windings have served as a means of indicating generator-rotor position in the differential. Similarly the combination of the synchrogenerator-rotor and the differential-rotor positions is indicated by the synchrocontrol-transformer flux field C. Thus the apparent position of the generator rotor as indicated to the control transformer by flux field C can be changed by rotating the synchro differential rotor.

The control transformer is similar to the synchrogenerator except that the rotor has a distributed winding to reduce reaction torque. With the synchrocontrol transformer rotor in the position shown in Fig. 11, no error voltage appears across terminals T_1 and T_2 as the rotor is at right angles to the flux field C. As the rotor is rotated the component of flux intercepted is proportional to the sine of the angle of rotation, consequently, the error voltage at T_1 and T_2 also varies as the sine of the angle of control-transformer-rotor rotation. The rotation of either the synchrogenerator or differential rotor produces a voltage at T_1 and T_2 that varies as the sine of the angle of rotation. This occurs because movement of either rotor shifts flux field C relative to the control-transformer rotor.

If the generator and control-transformer rotors are rotated at exactly the same speed the voltage at T_1 and T_2 does not vary as the relative position of flux field C and the control-transformer rotor does not change.

As used in this system the synchrogenerator B, Fig 4, is driven by the No. 1 motor and is electrically connected through the synchrodifferential C, Fig. 10, in the control unit to the synchrocontrol transformer which is physically located on the No. 2 unit exactly as in B, Fig. 4. The control transformer is driven by the No. 2 motor. The phase-relationship-error voltage obtained from the synchrosystem, as described, is superimposed on the frequency control in the amplifier channel for the No. 2 unit.

The voltage output of the No. 1 tachometer generator is used to give an indication of the frequency. A voltmeter, see Fig. 6, in the indicating panel, calibrated in cycles per min, is used for that purpose. The control-transformer error voltage, which is a measure of the phase-angle error between the Nos. 1 and 2 units, is indicated by an a-c voltmeter on the indicating panel. This phase-angle error is required so that enough control-transformer error voltage is produced at the No. 2 unit amplifier channel to hold a fixed phase relationship between the Nos. 1 and 2 units.

Eccentricity Control

The eccentricity of an oscillator unit is varied by operating a switch on the control-unit panel. It is a two-way momentary "on" switch, and by using relays and limit switches D and E, Fig. 12, the eccentricity motor B, Fig. 12, is operated in either direction so as either to increase or to decrease the eccentricity.

A synchrogenerator C, Fig. 12, is driven by the eccentricity motor. By electrically connecting this synchrogenerator to a similar one in the indicating-panel unit which drives a counter, an indication of the eccentricity in each of the units is obtained. This, of course, is the standard synchrotie connection. The

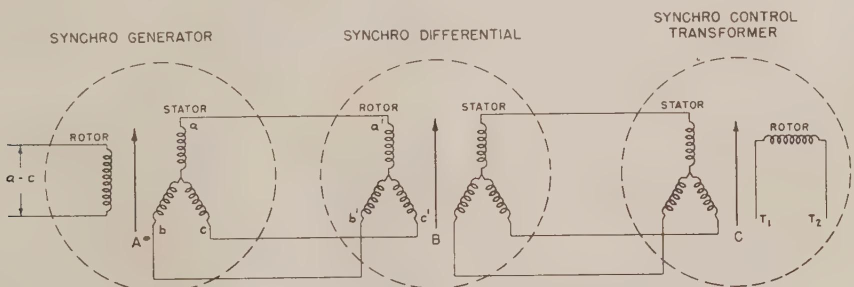

Fig. 11 Synchrosystem Schematic

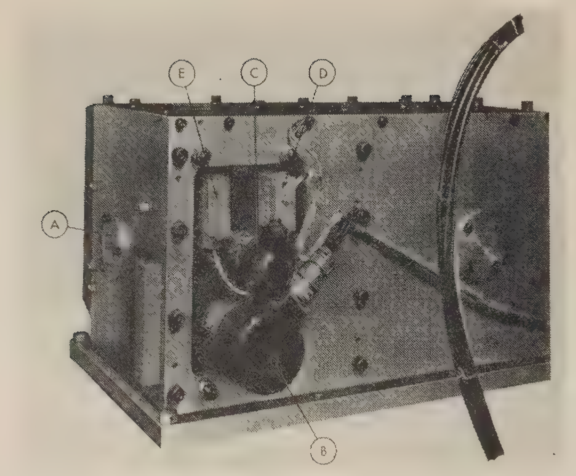

Fig. 12 Eccentricity Motor End of 45-Deg Rack-Type Oscillator

amount of unbalance in the unit can be determined mechanically by inserting a calibrated needle gage in bolt hole of *A*, Fig. 12.

Amplifier Operation

The amplifier shown in Fig. 8 is used as a means of detecting the low-power-level intelligences offered by the battery, d-c tachometer, and synchro machine circuits, and of converting them almost instantaneously into intelligence at a power level sufficiently high to control a Rototrol generator-control field. A Rototrol is in itself a fine power amplifier. Its power output, which is regulated by the control field, is great enough to run an oscillator drive motor, and yet it is a fast machine electrically, responding quickly to the intelligence the amplifier transmits to its control field.

The Nos. 1 and 2 channels of the amplifier are similar. Each has two stages of push-pull d-c voltage amplification for which a pair of 6SL7's are used, *A* Fig. 8, and one push-pull power-amplification stage in which two 6L6's, *H* Fig. 8, are used. A dynamotor *B*, Fig. 10, supplies direct current for the plates of the 6SL7's, whereas the plate supply for the 6L6's is alternating current. Each pair is supplied by a separate secondary winding of the same transformer. The difference between the two channels of the amplifier is that the synchrocontrol-transformer error voltage is introduced into the No. 2 channel only. This locks the No. 2 oscillator in a definite phase relationship to the No. 1 oscillator. As previously described, this phase relationship can be varied from zero to 360 deg by rotating the differential synchromachine rotor.

As in the case of all servomechanism (8) systems, proper damping means must be used to prevent "hunting." In this oscillator system damping is obtained by derivative feedback from the Rototrol armature circuit.

Conclusion

During the past 17 years the use of mechanical oscillators for simulated service testing has enjoyed steady growth. Originally developed for testing railroad bridges, oscillator testing has been found applicable to testing many other types of structures, such as aircraft, land vehicles of all types, buildings and foundations, and ships. In testing such structures mechanical oscillators have been used to determine such properties as fatigue strength, natural frequency of vibration, damping capacity, resonant stress, flutter, riding comfort, and other vibration-response characteristics. Oscillators have also been used for many types of soil investigation (such as the behavior of an airport landing strip under vibration), and as a laboratory tool to study fatigue, stress-relief possibilities, dynamic modulus of elasticity, dynamic creep, etc. A relatively small oscillator is a particularly powerful dynamic testing machine when used to produce resonant vibrations; under these conditions the force exerted on the part under test may be from 10 to more than 100 times the oscillator force (3).

Certain types of dynamic testing require two or more oscillators properly synchronized and controlled to induce a desired vibration. For example, one method of determining flutter characteristics in aircraft wings is to attach an oscillator to each wing to excite controlled and properly synchronized vibrations during flight. The equipment described in this paper will perform such a task. The combination Rototrol and electronic control will maintain precisely the frequency and phase relationship of two or more oscillators. Additional electrical control provides means of adjusting the frequency, phase, and eccentricity of the oscillators while the units are operating.

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Precision Tachometer for Use in Wind-Tunnel Testing

By R. K. FAIRLEY[1] and H. L. CLARK[1]

The equipment described is a high-speed high-accuracy tachometer developed primarily to measure rotative speed during the power-testing of airplane models in wind tunnels by measuring the frequency of a tachometer generator. The frequency being measured is automatically compared with that of a tuning fork and indicated on an instrument having a scale length of 86 in.

One of the largest laboratory devices we have today is the modern wind tunnel, built to analyze the performance of planes and their component parts. While all tunnels are custom-built, they have one common characteristic, that is, the need for many accurate instruments. In addition to fairly good accuracy, good precision is also required since precision or constancy of indication will reveal trends resulting from model changes.

To build instruments of the accuracy required is sufficiently difficult, but to add to the difficulty, such instruments should be self-reading or recording. As the design of airplanes increases in complexity, the number of instruments which must be read simultaneously is multiplied. To expedite research work, not to mention economics which is sometimes forgotten under the stress of war work, it is highly desirable to obtain such readings at the touch of a button. Automatic recording avoids the errors of an observer, particularly those caused by fatigue. Not only are such errors obviated by self-recording features, but such readings may also be recorded in a calculating machine and much laborious manual calculating work avoided.

Some phases of model testing involve so-called "power on tests." This means that the model plane will be provided with a propeller and a power plant, usually an electric motor or motors. When a test is in progress it is necessary to know accurately the speeds of these electric power plants and the developed torques. Torque is usually measured indirectly, first, by calibrating each motor in a dynamometer and measuring the electrical input at each given speed. The motor is then mounted in a model plane in the wind tunnel, and at the same speed, the same watts, volts, and amperes are applied and the calibrated torque obtained.

During a test run the watts, volts, amperes, and speed are continuously measured and numbers set up in a printing machine. When stable conditions are reached these quantities, together with their associated multipliers, etc., are simultaneously recorded in the printing machine.

It is the purpose of this paper to describe the equipment which measures the speed and actuates the printing machine.

Principle of Operation

Accurate measurements are usually facilitated by comparison with a known value, commonly referred to as a standard. For example, accurate measurements of weight can better be made with a balance, using tested weights, than with a spring scale. The balance uses the comparison principle while the spring scale uses the deflection principle.

A similar principle applied to the measurement of frequency is often used employing an oscilloscope and Lisajou figures. The ratio between the unknown and known frequencies, such as $^3/_4$, $^5/_6$, etc., is deduced from the shape of the pattern. The equipment to be described automatically obtains this ratio using a self-contained known frequency, and indicates the value of the unknown frequency on a special long-scale instrument.

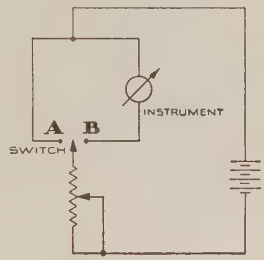

Fig. 1 Basic Circuit, Deflection Principle

A simplified circuit showing the principle of operation is illustrated in Fig. 1. The switch, normally in position A, snaps over to position B at the beginning of the cycle being measured. This is shown at point 1 in Fig. 2(a). After a definite length of time this switch automatically switches back to position A and stays there until the beginning of the next cycle of the frequency being measured (point 2). Current will flow through the instrument while the switch is in position B. The wave shape of this current is shown in Fig. 2(b), in which t_1 shows the definite length of time the switch stands in position B, and t_2 is the time for one cycle of the frequency being measured. The instrument will indicate the average value of the current, shown as i_{av}. When a higher frequency is being measured, as shown in Fig. 2(c), the switch snaps from position A to position B at the beginning of the cycle, stands in position B for the same definite length of time as before, then snaps to position A, and then waits in position A until the beginning of the next cycle. The current flows through the instrument as shown in Fig. 2(d). Since the pulses of current are spaced more closely together, the average current will be higher. This average current is given by

$$i_{av} = i_{max} \frac{t_1}{t_2} \qquad [1]$$

Since the frequency being measured is

$$f = \frac{1}{t_2} \qquad [2]$$

the average current is then

$$i_{av} = i_{max}\, t_1 f \qquad [3]$$

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[1] General Engineering and Consulting Laboratory, General Electric Company, Schenectady, N. Y.

Contributed by the Industrial Instruments and Regulators Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

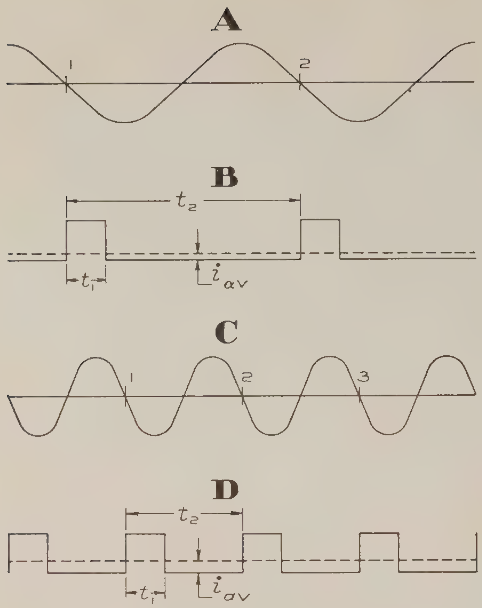

Fig. 2 Current Impulses, Showing Operation of Switch

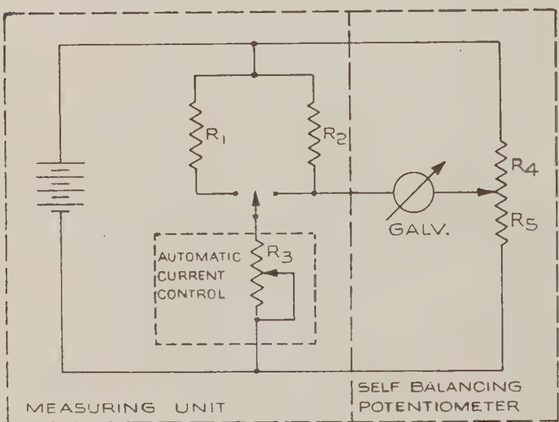

Fig. 3 Basic Circuit, Balance Principle

Equation [3] shows that the average current through the instrument is proportional to the frequency being measured.

However, the use of an ordinary indicating instrument to measure the average current makes use of the deflection principle instead of the more accurate comparison principle. Rearrangement of Equation [3] gives

$$\frac{i_{av}}{i_{max}} = t_1 f \qquad [4]$$

Here we see that the ratio between the average and maximum current is proportional to the frequency being measured since t_1 is a constant.

This ratio may conveniently be measured by using a self-balancing bridge, the basic circuit of which is shown in Fig. 3. The bridge consists of resistors R_2, R_3, and R_4 plus R_5 which is a self-balancing potentiometer. (The resistor R_3, denoted "Automatic Current Control," is required to hold the current through this arm of the bridge strictly proportional to the supply voltage. This is necessary because of the somewhat variable voltage drop across the switching element which is to be described later.) The voltage across R_2 is proportional to i_{av}, while the voltage across R_2 plus R_3 is proportional to i_{max}. When the slider on the potentiometer is adjusted so that no current flows through the galvanometer

$$\frac{i_{av}}{i_{max}} = \frac{R_4}{R_4 + R_5} = t_1 f \qquad [5]$$

Here we see that the value of the frequency being measured is determined by the ratio of two resistances and the constant t_1 which is the time for one cycle of the tuning fork.

Since changes in the value of the supply voltage produce like changes in i_{av} and i_{max}, the ratio of i_{av}/i_{max} remains unchanged. Variations in the power-supply voltage do not change the value of the frequency indicated.

As shown in Fig. 4, the equipment consists of two essential units, a measuring unit and a self-balancing potentiometer. The measuring unit performs the switching operations which determine the ratio of i_{av}/i_{max} while the self-balancing potentiometer measures this ratio.

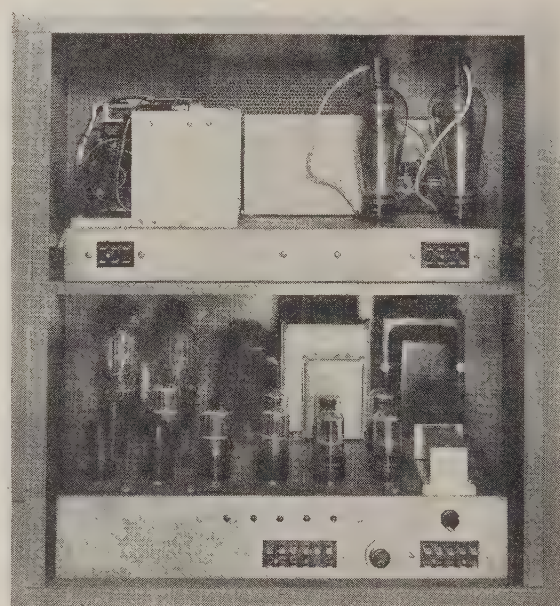

Fig. 4 Precision Tachometer

Measuring Unit

Since the required switching speed is very high, electronic instead of mechanical switches are used. To form an electronic switch two tubes are connected as shown in Fig. 5. Each tube acts as one pole of a single-pole double-throw switch, that is, only one tube conducts current at a given time. For instance, assume tube V_1, in Fig. 5, is carrying current. This current flows through R_1, causing a voltage drop. Through the voltage divider R_5 and R_6, the grid of V_2 is held so far negative with respect to its cathode that no current flows in this tube. There is no voltage drop across R_4 so that through the divider R_2 and R_3, the grid of V_1 is held positive, and V_1 continues to conduct current.

To change the switch from its original position a negative impulse may be applied to the grid of V_1 decreasing the current in V_1,

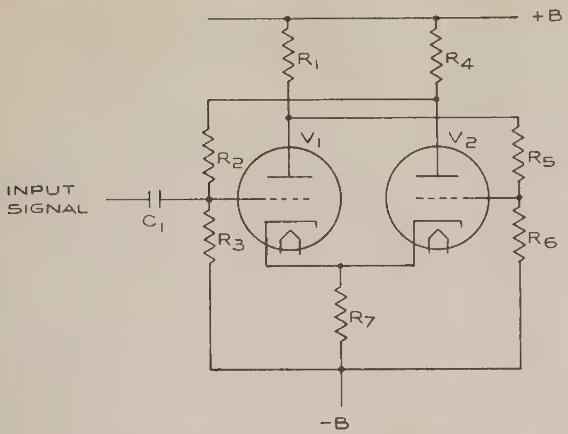

Fig. 5 Circuit of Electronic Switch

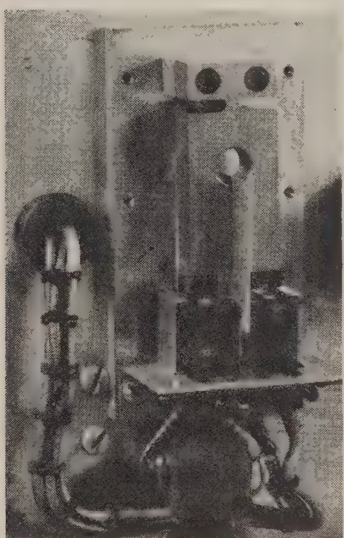

Fig. 6 Tuning Fork

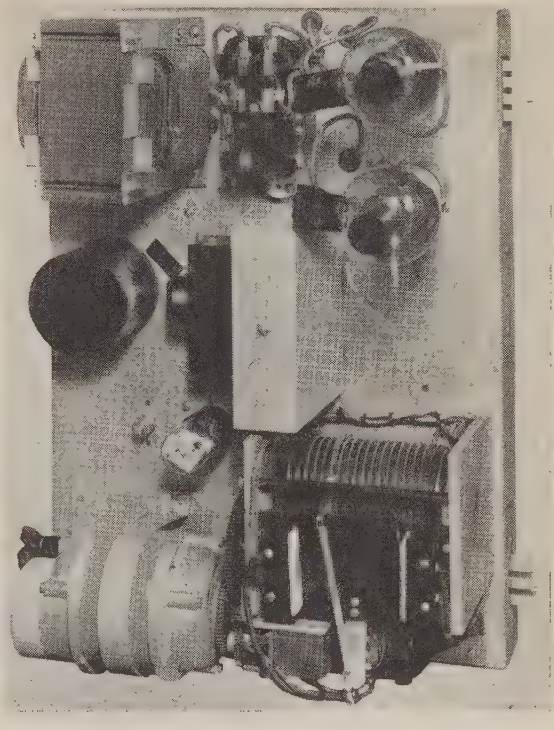

Fig. 7 Self-Balancing Potentiometer

causing its plate voltage to rise. Through the voltage divider the grid of V_2 is driven positive, causing this tube to conduct current. The resulting voltage drop across R_4 will, through the voltage divider, drive the grid of V_1 negative, thus holding off tube V_1 while tube V_2 conducts current. Thus a voltage impulse applied to one of the grids causes a transfer of current to take place similar to that of a single-pole double-throw switch.

Several of these electronic switches are used in combination to obtain the current pulses as shown in Fig. 2.

The timing element of the equipment is a tuning-fork oscillator. A close-up view of the tuning fork is shown in Fig. 6. A small Alnico magnet is mounted on the end of each tine of the tuning fork, and a coil opposite each magnet. Motion of the tuning fork generates a voltage in the coils, one of which is connected to the grid of an amplifier tube. This voltage, after being amplified, is applied to the other coil, thus sustaining the oscillations.

The frequency of the tuning fork, which is approximately 3000 cycles per sec, must in most cases be 3 times the highest frequency to be measured. Such a tuning fork can be made to provide a high degree of accuracy. The principal cause of inaccuracy is the temperature coefficient of the material. A material having a temperature coefficient of 5 parts per million per centigrade degree is used so that an accuracy of the order of 0.05 per cent is easily obtained.

Self-Balancing Potentiometer

The self-balancing potentiometer shown in Fig. 7 uses as a galvanometer an ordinary microammeter with a mirror instead of a pointer. This galvanometer reflects a beam of light from the light source to two phototubes, dividing it equally between the phototubes when the circuit is balanced. If one phototube receives more light than the other, one of the thyratrons passes current through the motor, which through a gear reduction drives the potentiometer in the direction which will bring the galvanometer to balance. The potentiometer will accurately follow variations of 2 per cent of full scale per second and has a maximum speed of 15 per cent per second. This speed is ample to follow the speed variations encountered in the application described.

The potentiometer consists of 10,000 or more turns of fine resistance wire wrapped around a core of Formex-insulated copper wire, and 14 turns of the whole wrapped around a grooved drum. A split nut carrying the slider rides in a groove so that as the drum rotates the slider follows the resistance wire. A pin on the split nut rides against a cam which is shaped so as to advance or retard the slider to compensate for nonlinearity of the resistance wire. The potentiometer unit is made accurate to 0.1 per cent or better.

Also connected to the gearing is a selsyn transmitter so that the position of the potentiometer can be transmitted to a remote location for operating the recording mechanism for indication or control. The indicating instrument shown in Fig. 8 has two hands, one of which makes 10 revolutions while the second hand indicates the number of revolutions made by the first hand. The

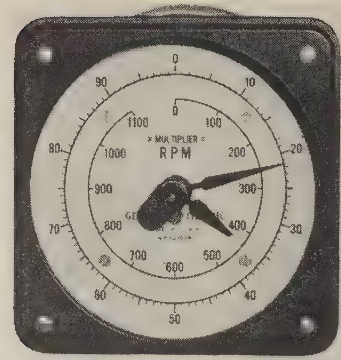

Fig. 8 Long-Scale Indicating Instrument

scale length for 10 revolutions is 86 in., making each division, which is 0.1 per cent, very nearly $^3/_{32}$ in. so that a reading to the accuracy of the equipment can be made without estimating fractions of a division. The transmitter and receiver selsyns each turn 150 revolutions for full scale so that an error of 5 deg in the selsyn, for example, would cause less than 0.01 per cent error in the indication.

Accuracy of Instrument

The method of calibration is largely responsible for the high degree of accuracy obtainable with this equipment. The known frequency of the tuning fork is connected to the input of the equipment with the circuits so arranged that one half the frequency of the tuning fork is measured. The indicating circuit is then adjusted so that the proper reading is obtained, thus correcting for any variations in the electronic circuit. Since the circuit is essentially independent of tube characteristics, these initial variations are small, and after being corrected as described, are negligible. The factors entering into the accuracy of the measuring unit then are the frequency of the tuning fork and the values of several precision resistors which are accurate to 0.05 per cent or better.

Adding up the several factors, 0.05 per cent for the tuning fork, 0.05 per cent for the resistors, and 0.1 per cent for the self-balancing potentiometer, the over-all accuracy of the device is 0.2 per cent or better.

Electromagnetic Torquemeter

By M. W. HIVELY[1] and D. F. LIVERMORE[1]

The electromagnetic torquemeter is a dynamometer of the transmission type. A calibrated shaft unit is directly connected in the rotating shaft where the torque is to be measured. This paper describes a slip-ring type of torquemeter in which the small torsional deflections in the shaft unit are measured by electromagnetic-gage elements and associated electrical components to give a continuous direct reading of torque with an accuracy of 1 per cent of full-scale torque. By proper mechanical and electrical design, effects of centrifugal forces and changes in brush resistance are held to negligible values. Several torquemeters of this type are now in use. They cover a range of 250 lb-ft at 8000 rpm to 10,000 lb-ft at 400 rpm. The same general torquemeter principle is readily applicable to much higher torque ranges.

Introduction

There is a growing demand for torquemeter equipments to measure the torque transmission through a rotating-shaft system. Some typical examples are the torque applied to an aircraft propeller or helicopter rotor, the torque transmission from a prime mover to a compressor load, and the torque transmission through drive shafts and axles in automobiles and trucks. The torquemeter will, of course, not replace the conventional cradled-dynamometer equipments which are often used for similar purposes, but nevertheless it has a very definite field of application of its own.

The torquemeter development has been going on for many years, and the various designs are almost as numerous as the development engineers who have worked in this field. Substantial progress has recently been made.

The electromagnetic torquemeter discussed herein is a transmission dynamometer of the torsion type in which a calibrated shaft unit is inserted as a section of the rotating shaft.

Most torquemeters for rotating shafting make use of the same fundamental physical principle: An elastic member, when subjected to an external torque within the elastic limit of the member, will deflect torsionally an amount proportional to the applied torque, and will return to its original state when the load is removed. The amount of angular twist in a fixed length of shaft is measured by some means. It is in this means of measuring the twist that the several types of torquemeters vary. The General Electric electromagnetic shaft unit, to which most of the discussion to follow is devoted, measures the torsional deflections, over a fixed shaft length, with electromagnetic gages. Changes in torque produce corresponding changes in the air gaps of the gages mounted in the shaft unit. These changes can be measured electrically by use of suitable electrical components which are a part of the torquemeter equipment.

Description of Electromagnetic-Torquemeter Shaft Unit

The electromagnetic torquemeter consists of the shaft unit and electrical components required to translate torsional deflections of the shaft unit into electrical-instrument readings.

The shaft unit is shown in Figs. 1 and 2. It consists of a hardened alloy-steel shaft 20 to 30 in. long which is coupled between the prime mover and the load. A section of this shaft is made with a reduced diameter so that approximately $^1/_3$ deg of torsional displacement is obtained in 2 to 4 in. of shaft length.

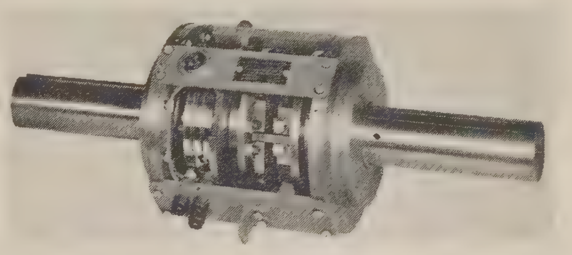

Fig. 1 Electromagnetic-Torquemeter Shaft Unit

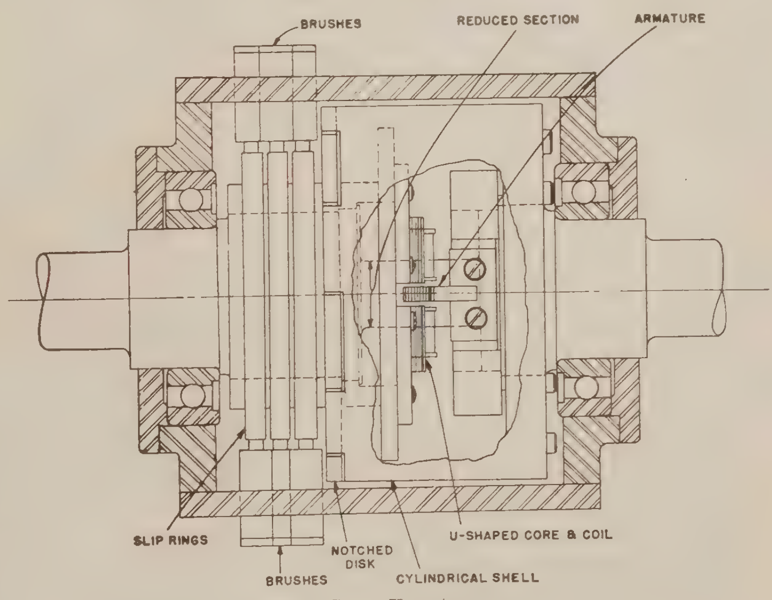

Fig. 2 Shaft-Unit Assembly

[1] General Engineering and Consulting Laboratory, General Electric Company, Schenectady, N. Y.

Contributed by the Instruments and Regulators Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

A bracket fastened rigidly on one end of this section transfers this torsional displacement to the other end of the reduced section where it is measured by securely attached electromagnetic-gage elements, consisting of two U-shaped laminated cores and associated coils with an armature common to both of them. Two such brackets and gaging elements are located diametrically opposite each other with the gaging elements electrically connected together. Pure torque affects each of these elements equally. Torque and shaft-bending affect one element such that it sees a displacement which is a function of the torque, plus a displacement which is a function of bending. The other element sees a displacement which is a function of torque minus the bending effect. The two gage elements electrically average these displacements, and the bending effect in one element is canceled by an opposite effect in the other element.

A cylindrical shell spanning this reduced section of shaft and covering both gage elements is securely fastened to the shaft just beyond one end of the reduced section. A disk is securely attached to the shaft just beyond the other end of this reduced section. Both cylindrical shell and disk are notched and fitted together in such a manner that they do not touch each other until the torque is approximately 250 per cent of the maximum rating of the shaft unit. From that point on the cylindrical shell and disk carry a large part of the overload torque. This feature protects both the reduced section of shaft and the gage elements if overloads are encountered. Such overloads might occur due to sudden loading or excessive torsional oscillations. In Fig. 1 the shaft unit is shown with cover and cylindrical shell removed; however, the notched disk can be seen at the right of the slip rings.

The electrical connections to the gaging elements are made through three slip rings and two sets of brushes. Careful consideration has been given in the design of the shaft unit to minimize the effects due to variation in contact resistance between brushes and slip rings. How this is accomplished is discussed in the section entitled "Design Considerations."

Each slip ring is engaged by two brushes located 180 deg apart. The brush rigging and cover assembly is supported by end plates which are held concentric with the shaft by standard ball bearings, serving to locate the brushes in the proper axial position with respect to the shaft. The cover assembly is held stationary by a suitable anchor.

Electrical Components

The electrical components shown in Figs. 3 and 4 translate the torsional deflections as detected by the shaft-unit gaging elements into electrical-instrument readings. The shaft unit has already been described. The oscillator supplies 2000-cycle power to the torquemeter power unit which contains impedance-comparing elements. The basic circuit of the power unit and shaft unit is shown in Fig. 5. Coil 1 is located on the shaft unit diametrically opposite to coil 3. Coil 2 is adjacent to coil 1 with coil 4 diametrically opposite. When a torsional displacement increases the impedance of coil 1, it also increases the impedance of coil 3 and decreases the impedance of coils 2 and 4. These coils are connected to two capacitors $C1$ and $C2$ and to the indicating instrument through a unique rectifier arrangement as shown.

When the average impedance of coils 1 and 3 is equal to the average impedance of coils 2 and 4, there is no current flowing in the torque indicating instrument. For average impedance of 1 and 3 greater than that of 2 and 4, the current will flow in the opposite direction from that obtained when the average impedance of 1 and 3 is less than that of 2 and 4.

An electrical zero-position adjustment of the torque indicating instrument is obtained by potentiometer R. The voltmeter V is used when setting the transformer-secondary voltage at a predetermined value to give correct sensitivity.

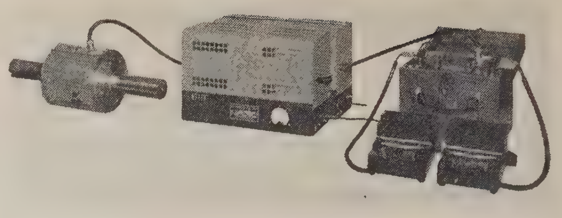

Fig. 3 Electromagnetic-Torquemeter Shaft Unit With Electrical Components

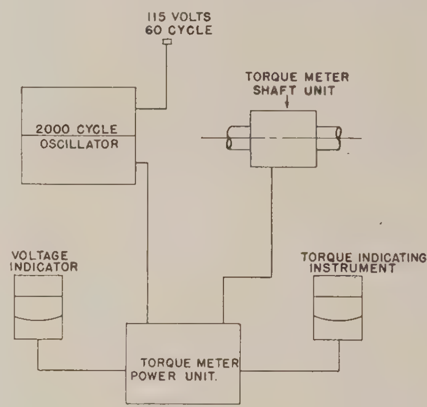

Fig. 4 Interconnection Diagram

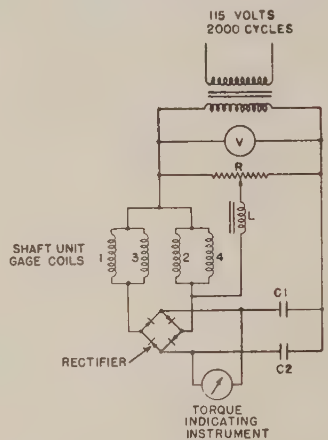

Fig. 5 Schematic Diagram of Torquemeter-Gage Circuit

The torque indicating instrument is a direct-current milliammeter with its scale marked in pound-feet of torque.

Design Considerations

Since the electrical connections to the gaging elements are made through brushes and slip rings, the contact resistance between brushes and slip rings constitutes a part of the impedance measured by the electrical components. By making the impedance of a gaging element large in comparison to the effective alternating-current resistance, the brush-contact resistance has a negligible effect on the impedance. For example, consider the vector diagram shown in Fig. 6; R, X, and Z are, respectively,

the effective alternating-current resistance, the reactance and the impedance of the gage element, including brush and lead resistances. Let ΔR be the change in R due to variations in brush-contact resistance. If ΔR is 1 per cent of the total R, Z is changed by 1 per cent for $\theta = 0$, by 0.65 per cent for $\theta = 30$ deg, by 0.25 per cent for $\theta = 60$ deg, and by approximately zero per cent for $\theta = 90$ deg.

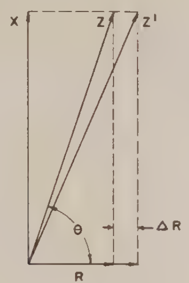

FIG. 6 VECTOR DIAGRAM OF GAGE ELEMENT

A value of θ close to 80 deg is obtained by use of laminated cores and armatures of very good magnetic properties, the use of coils with a large number of turns, and the use of 2000-cycle excitation. At 80 deg a 1 per cent change in total resistance produces only 0.04 per cent change in impedance. The electric circuit used compares impedances independent of phase angles. Since an impedance differential of 40 per cent is obtained at 100 per cent torque, this 0.04 per cent change in impedance due to 1 per cent resistance change will produce only 0.1 per cent change in torque reading. The variation in brush resistance observed under normal operating conditions is from 1 to 2 per cent of the total resistance. This small change in brush-contact resistance is obtained by using silver slip rings and graphite brushes.

Variation in brush resistance was found to increase somewhat with increased slip-ring surface speed. While a sharp line cannot be drawn as to the limiting speed to be tolerated, it has been found desirable to keep the slip-ring peripheral speeds at 80 fps or less.

The action of centrifugal forces on the gage elements is also an important consideration. Gage coils and laminations must be supported rigidly and strongly: (1) to hold the gages in place and (2) to prevent centrifugal forces from displacing the gage elements in a manner which would produce a drift in torquemeter readings as a function of speed. The particular unit shown in Fig. 1 has been run at speeds over 5000 rpm without encountering difficulties from centrifugal effects. A torquemeter of the same general type has successfully been run at 8000 rpm. This was accomplished by a more rigid gage-element supporting structure and by using small slip rings.

Over-All Performance

In general, an accuracy of 1 per cent and a repetitive accuracy of 0.5 per cent of full-scale torque can be expected with the electromagnetic torquemeter described in this paper. Tests made using a cradled dynamometer to calibrate the shaft unit at operating speeds show that static calibrations agree so closely with dynamic calibrations that only static calibrations are needed. A typical calibration curve is shown in Fig. 7. The calibration is seen to be very linear except for a slight deviation at full torque. When best accuracy is required, corrections obtained from calibration data should be applied to the torquemeter readings.

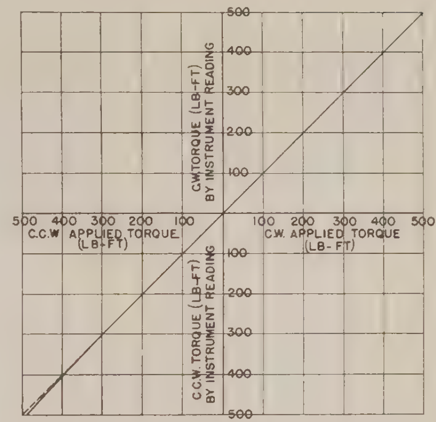

FIG. 7 CALIBRATION CURVE FOR SHAFT UNIT RATED AT 500 LB-FT MAXIMUM TORQUE

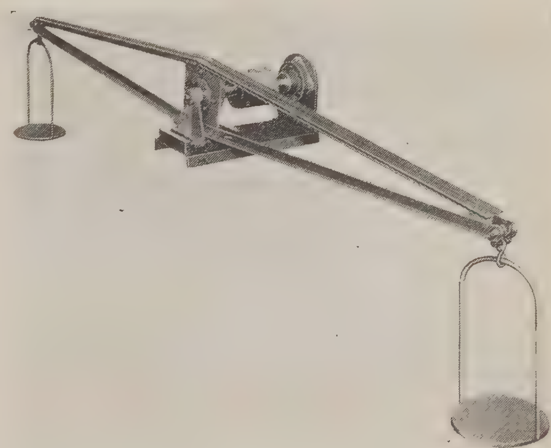

FIG. 8 STATIC CALIBRATING STAND

Calibration

Fig. 8 shows a static calibrating stand with a shaft unit bolted by the attached half-couplings to a fixed plate on one end of the calibrating stand and a pivoted torque arm on the other, through which a known load is applied by adding standard weights to the weight pans hanging from knife-edges at the ends of the arm. The knife-edges provide an accurate loading point, and the ball-bearing central pivot supports the applied weights and provides a nearly frictionless pivot, making it possible to apply an accurately known torque. Thus very accurate calibrations can be obtained.

Applications

The operation of electromagnetic torquemeters now in practical use has proved to be very satisfactory. The torque and speed ranges covered by these torquemeters extend from 250 lb-ft at 8000 rpm to 10,000 lb-ft at 400 rpm. These ranges are by no means considered as limits for electromagnetic torquemeters.

Modern trends in the design of rotating equipment are toward lightweight with accompanying high stresses. Therefore it becomes not only desirable but essential, in many cases, to know just what torque is being delivered through a section of shafting.

In addition to the torque instrument reading obtained, the torquemeter output may be used with electronic control equipment for regulating the torque delivered by a rotating shaft.

Someday torquemeters of one type or another will be standard equipment in large aircraft, on ships, and many other places where rotating shafts are used to transmit power. For the present, however, torquemeters will likely find their greatest use in production-test setups where comparative data are needed to standardize a product, or in research and development test installations where they can be used to obtain data useful in improving the design of a product.

Discussion

J. G. Fleming.[2] It is suggested that the following points discussed by the authors might be further clarified:

1 An accuracy of 1 per cent of full scale is cited in the paper. How long can this accuracy be maintained under normal operating conditions without rechecking for calibration and zero stability?

2 What is the speed of response of this instrument?

3 The application of the electromagnetic torquemeter for large-aircraft use is cited in the paper. How compact would the instrument be and what would be its approximate weight and cost for application with aircraft engines rated at approximately 1000 hp?

4 It is stated that the slip-ring contact resistance is very small compared to the circuit resistance and that therefore variations in slip-ring resistance do not affect the instrument operation. Fig. 5 of the paper indicates that the output is fed into a rectifier bridge which operates a galvanometer. The resistance of such rectifiers is usually quite large and undoubtedly much larger than the slip-ring contact resistance. The resistance of rectifiers increases on aging, and it appears that such a change would cause a gradual calibration drift. A bridge of this type could be operated with a null-balance galvanometer which would make the measuring circuit free from changes in resistance in the rectifiers.

N. S. Muir.[3] The writer wishes to confirm the authors' belief that the application of electromagnetic principles to the measurement of torque can provide a very satisfactory method capable of wide application in various mechanical systems. The authors may be interested in learning something about a similar application of the electromagnetic principle in a transmission dynamometer developed at the Royal Aircraft Establishment in England about 13 years ago which aimed at giving an accurate record of the mean torque transmitted by an aircraft engine to a propeller.

The research which preceded the development and construction of this particular torquemeter was initiated by the British Air Ministry early in 1930, aiming at the evolution of a satisfactory transmission dynamometer for the measurement of engine power in flight and in the then proposed 24-ft wind tunnel. From a critical discussion of the available methods as one would employ them in a practical transmission dynamometer, it was decided that the variable-air-gap method would provide a basic arrangement which would be favorable compared with the others, such as piezoelectric crystals, capacity units, resistance-pressure elements, etc.

One of the main features of the electromagnetic system in common with that of a capacity unit is that no direct connection need exist between the two moving members other than the flexible element providing the desired relationship between force and deflection. The line of attack chosen therefore for the research was the construction of a torquemeter in which the angular deflection of a spring element inserted in the drive between an engine and airscrew would be measured by electrical means, using variable air gaps. Thus we had two of the time-honored foundations for the transmission dynamometer design being remolded and combined in the light of modern knowledge of the application of scientific principles and the use of new materials. The basis of the electrical system chosen was attributable to Ford and may be described briefly as one in which the electromotive force (emf) due to change of inductance of a transformer secondary coil, consequent upon a change of air gap in an adjacent iron circuit forming the core, is balanced by an equal and opposite emf generated in a remotely situated receiver having a similar circuit operated by hand.

In the torquemeter being described, the deflection of the spring element was arranged to vary the air gaps in the system specially developed for the purpose, and an electrical balance was obtained by means of a micrometer-screw device on the receiver; the balanced position of the air gaps being read off a scale attached to the screw as in the Ford torsion meter. The null-reading condition of electrical balance is indicated by a moving-coil current-measuring instrument of adequate sensitivity. When forced torsional oscillations are present in the drive, the air gaps vary cyclically about the mean value, and the true mean deflection is still readable in terms of electrical balance of the moving-coil instrument, the mechanism of which is then suitably damped.

In the earlier air-gap instruments of Denny, Ford, and Moullin, the systems were energized from small alternators or by interrupted direct current of relatively low frequency. This latest unit to which reference is made is fed from a special 1500-cycle alternator designed to have a purely sinusoidal voltage output. The high frequency is essential to deal with the presence of torque oscillations in the airscrew drives in aero engines up to about 200 per sec, so that integration of the torque variation may be sufficiently accurate.

The basic design for our first torquemeter was completed in March, 1930, and the apparatus was ready for calibration in May, 1931. Preliminary experience was gained with this first unit, covering some 70 or 80 hr running on the test bench on a Jupiter VII supercharged direct-drive engine and using the Froude brake as the criterion of torque. At constant speeds, accuracy within 0.3 per cent was obtained, while over a range of speeds the greatest error was within 2 per cent. The apparatus was then transferred to an aircraft and successful use in flight was achieved. Special apparatus was provided to obtain an accurate measure of the rotational speed of the engine, since the computation of horsepower depends upon the product of speed and torque.

A second unit was provided primarily for use in the then proposed 24-ft wind tunnel at Farnborough to enable full-scale testing of propellers to be carried out. In this case the spring element took the form of a one-piece wheel in which the rim, spokes, and hub were carved out of a solid disk of about 18 to 20 in. diam and was capable, in conjunction with the electrical system, of giving readings of torque over a wide range from one-quarter to full torque of engines up to about 1000 hp.

It will be appreciated that the null-reading electrical system chosen is not the same as that described in the paper just presented, which uses an electrical system on the lines of that used by Moullin, where the output of an alternator or 2000-cycle oscillator is modulated by the torquemeter coils to give a direct rectified indication of torque. At an early stage of our investigation it was found essential to have an especially accurate apparatus for carrying out static-torque calibrations, and a tackle was

[2] The Bristol Company, Waterbury, Conn.

[3] Chief, Engine Development Section, British Supply and Air Commission, Washington, D. C.

specially designed for the purpose, which permitted the rolling load of a beam system to impose pure torques on the shaft carrying the torquemeter so that positive and negative torques could be smoothly applied according to the direction of motion of the load from the central position of no torque. The apparatus has ball bearings at every moving joint and is extremely delicate in its operation; torques of only 2 or 3 lb-in. being accurately obtained.

Limit stops were provided on the spring element which permitted a range of torque of from 60,000 lb-in. to approximately —12,000 lb-in. and some idea of the accuracy of the spring element in maintaining its zero at no torque is given by the fact that after taking the spring through several cycles of stress on the torque loading elements previously mentioned to remove any slight hysteresis in the material and then determine its zero setting, it returned after subsequent cycles of stress to this zero position within 1/40,000 in. at 9.625 in. radius. This radius refers to the position of a dial-gage reading to 1/10,000 in. which was attached in such a way as to record this circumferential deflection of the spring as it deflected. This dynamometer was used satisfactorily in large 24-ft wind tunnel at Farnborough for the measurement of torque in full-scale propellers.

Many applications of such a system can be made, and a dynamometer of this type lends itself to use either as a transmission coupling or as a static unit mounted in conjunction with aero-engine reduction gears or in any position where the torque to be measured reacts on a stationary member. From the writer's experience with this torquemeter, assurance can be given to the authors that their application is well worth development along the lines which they propose and for the applications which they mention. The writer's opinion, however, is that a null-reading system is much more likely to give enhanced accuracy than the use merely of the output from an unbalanced coil system. One important feature which must not be overlooked is, as previously stated, the accurate calibration of the spring element whether it be a piece of shaft or actual springs. This, in fact, is particularly important where in measurement on ship's propeller shafts a relatively long length of shaft is chosen. The writer agrees with the authors' attention to the slip rings and brushes used in the system. In the British torquemeter which the writer has just described, we have recently redesigned the recording circuit so as to eliminate completely any errors arising from faulty slip-ring contacts.

Author's Closure

Mr. Fleming brings up several points which will be discussed briefly in order.

(1) The electromagnetic torquemeter described should maintain its calibration. However, the zero adjustment may have to be made occasionally and should be checked whenever possible.

(2) The speed of response is limited by the indicating instrument and is approximately two seconds. A magnetic oscillograph may be used in place of the indicating instrument to record rapid torque variations up to several hundred cycles per second.

(3) The electromagnetic torquemeters described in the paper are not suitable for general flight applications on aircraft engines.

(4) The resistance of the rectifier section is approximately 20 ohms which is small in the light of the several hundred ohm gage circuit impedances. The advantages of a null system are recognized and are frequently put to good use. A continuous indicating system has the advantage of convenience over a null system where the balancing must be done manually. Accurate automatic null balancing systems are available.

Taking the Mystery Out of the Kadenacy System of Scavenging Diesel Engines

By P. H. SCHWEITZER,[1] C. W. VAN OVERBEKE,[2] and L. MANSON[3]

The "Kadenacy effect" of the Diesel-engine exhaust is utilized to create a vacuum in the cylinder for introducing the fresh charge. The result of the application of this system is exemplified in tests in 1939, of a converted Junkers opposed-piston engine in which the power was raised from 11 to 25 hp, or an increase of 130 per cent. This was accomplished solely by changing the characteristics of the inlet and exhaust ports and passages in accordance with the Kadenacy patents. However, the theory expounded by the inventor has been subject to question, as the authors explain, and while the results are entirely matters of record, the phenomenon can be accounted for by conventional thermodynamics based upon simple adiabatic expansion. This fact the authors have demonstrated by experiment. The formulas derived check closely with those of Kadenacy but they are based upon a more rational approach to the problem than Kadenacy's contention that supersonic velocities are involved in the exhaust process.

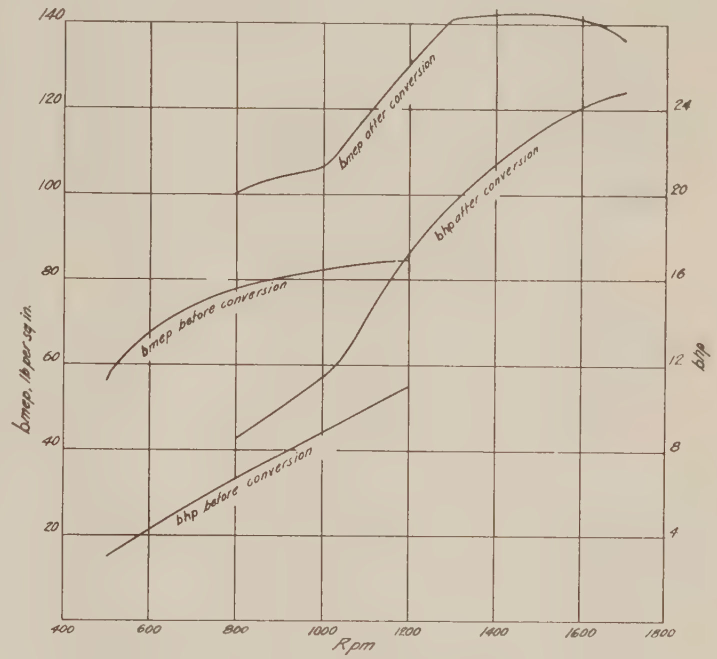

Fig. 1 Horsepower and Brake Mean Effective Pressure of Junkers Opposed-Piston Engine Before and After Conversion to Kadenacy System

During the last few years the Kadenacy system of scavenging Diesel engines has attracted some attention in this country and more abroad. That system is best known in the form of a blowerless two-stroke-cycle engine. The "Kadenacy effect" of the exhaust is utilized to create a vacuum in the cylinder for introducing the fresh charge. That by itself would not be remarkable if we were to attribute the effect to a tuned exhaust pipe. The Kadenacy system, however, embraces more than that. In numerous patents (1)[4] an ever-recurring sentence is that "at least a substantial portion of the burnt gases leaves the cylinder at a speed much higher than that obtaining when a flow resulting from an adiabatic expansion only is involved, and in such a short interval of time that it is discharged as a mass, leaving a depression behind it which is utilized in introducing a fresh charge into the cylinder, etc." This has been quoted from U. S. Patent 2,168,528. Kadenacy's other patents include various versions of the same statement and explain that mass means a "coherent mass" (Patent No. 2,123,569), "having properties similar to those of a resilient body" (Patent No. 2,102,559), and that the "ballistic" speeds involved in the evacuation are about 4 times higher than "the speed of adiabatic expansion," leaving behind them "a high depression which may reach a complete vacuum" (Patent No. 2,131,957).

The foregoing notions are so unorthodox that many engineers ignored and even ridiculed them. But it is not wise to ignore test results and Kadenacy has astonishing results to his credit.

The first commercial blowerless Kadenacy engine was built by Petter (2), and the same company is still building Kadenacy-type engines of an improved design (3); however, with a blower attached.

According to tests (4), described in 1939, the conversion of a Junkers opposed-piston engine resulted in a substantial increase of power. Some test results are reproduced in Fig. 1. The maximum power was raised from about 11 to 25 hp, an increase of 130 per cent. This is claimed to have been obtained without any alteration to the combustion chamber or fuel-injection equipment, solely by changing the characteristics of the inlet and exhaust ports and passages, in accordance with the Kadenacy patents. The scavenge pump was rendered inoperative and the inlet ports, which were still controlled by the upper piston, were arranged so that they communicated directly with the atmosphere. With the Kadenacy system fitted to the engine it was possible to run the engine to a much higher speed without any ill effects, it is claimed. The pistons remained in a cooler condition, the maxi-

[1] Professor of Engineering Research, Pennsylvania State College, State College, Pa. Mem. A.S.M.E.
[2] Ex-Cell-O Corporation, Detroit, Mich.
[3] DeLaval Steam Turbine Company, Trenton, N. J.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

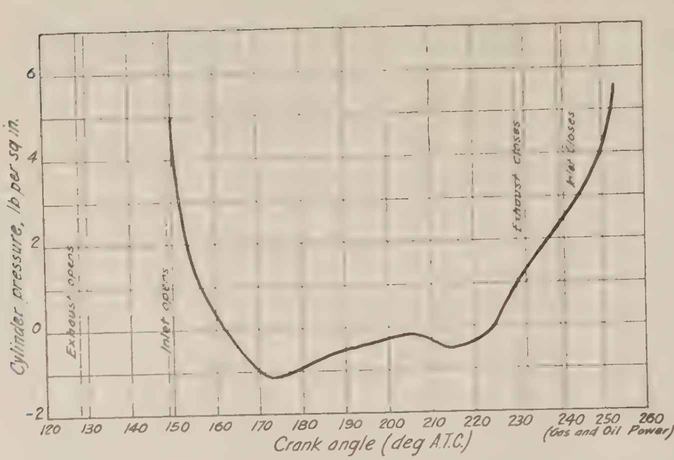

FIG. 2 CYLINDER PRESSURES IN THE JUNKERS-KADENACY OPPOSED-PISTON ENGINE

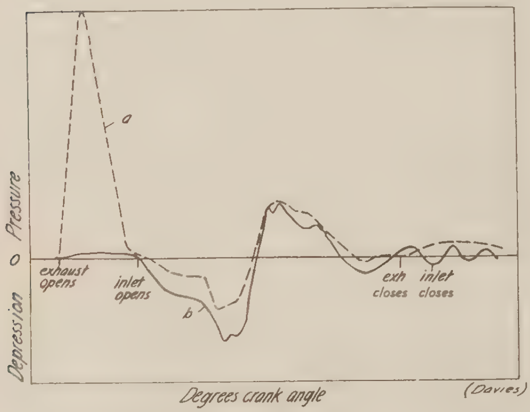

FIG. 3 PRESSURE FLUCTUATIONS IN EXHAUST DUCT—LINE A—AND INLET DUCT—LINE B—OF A JUNKERS-KADENACY ENGINE; 650 RPM

mum pressures were lower and the specific fuel consumption also was lower.

Fig. 2 shows the cylinder pressures in the Kadenacy-Junkers engine operating without scavenging air pump. It is notable that a depression over 1 psi below atmospheric occurred in the cylinder 50 deg after the exhaust ports had opened.

Fig. 3 shows the pressures in the intake and exhaust ducts of the converted Junkers engine from which it is seen that pressure changes are transmitted from the exhaust ducts to the intake ducts without appreciable delay. Tests at 800 rpm with various lengths of exhaust pipes gave Fig. 4 which shows that the Kadenacy effect does not depend upon a particular pipe length, only the time of maximum depression was retarded when the exhaust-pipe length was increased.

We have dwelt on this Kadenacy-Junkers engine because part of the tests reported were made by such an unquestioned authority as Prof. S. J. Davies of London, England.

Conversions of other two-stroke engines of either uniflow or the cross-scavenge type gave similar results, although not quite as spectacular as those shown in Fig. 1. Some engines fitted with the Kadenacy system had blowers and others had none. When a blower was used the scavenge-air pressure was considerably less, and significantly it decreased when the load increased. In the original engine the scavenging pressure increased with the load. The difference is explained by the fact that with the Kadenacy system as the load becomes greater the energy in the exhaust also becomes greater and with it the suction effect in the cylinder, thus reducing the resistance to the delivery of the air from the blower. Converted engines consistently showed appreciable power increase with lower specific fuel consumption and lower exhaust temperatures.

The mechanics of the Kadenacy principle were investigated by Davies (5) with a free-moving piston in a cylinder. He compressed a mixture of air and gasoline vapor with a hand crank and ignited it with a spark plug. The gas pressure sent the piston downward until it uncovered a slot through which the burnt gases discharged into the atmosphere. This sudden discharge caused so great a depression in the cylinder that the free piston rose in the cylinder and came to rest at about two thirds of its upward travel. With photoelectric apparatus, Davies recorded the piston travel and showed that the entire process was completed in a very short time. With a cylinder of 2.4 in. bore and 2.2 in. effective stroke the exhaust process lasted only 3.2 milliseconds and the residual depression was 17 in. Hg abs. Experiments with various lengths of exhaust pipes showed that the resulting depression and the time during which the exhaust slot remained uncovered were largely unaffected by the exhaust pipe. For instance, when the exhaust-pipe length was changed from 7.5 to 53 in., the duration of the exhaust period changed only about 6 per cent. The gas flow in the exhaust pipe was shown by high-speed photography of a lightweight "cursor" in a glass exhaust pipe. This revealed a very rapid back-and-forth movement of the gas column.

While the results obtained with the Kadenacy system are fully discussed in the literature, the constructional details are not disclosed. One or more of the expedients shown in the patent drawings such as tapered exhaust pipes and reflection-wave stoppers might have been employed to obtain the excellent results but apparently no particular importance is attached to them.

In view of these reports, the authors felt a necessity to de-

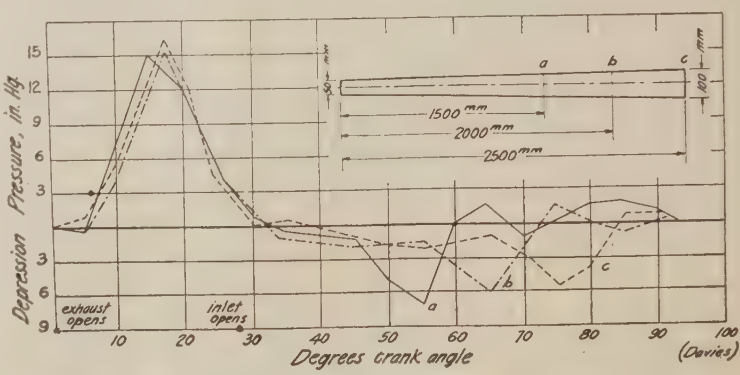

FIG. 4 PRESSURES IN EXHAUST PIPE OF JUNKERS-KADENACY ENGINE WITH VARIOUS LENGTHS OF EXHAUST PIPES; 800 RPM

termine in a conclusive manner whether the so-called Kadenacy effect really exists, aside from the pipe effect the nature of which is fairly well known (6).

Investigation of the Kadenacy Effect

Kadenacy claims that upon opening the exhaust port the gas leaving the cylinder with a very high (supersonic) velocity will evacuate it, leaving a partial or complete void behind, irrespective of whether a pipe is attached to the cylinder or not.

The depression thus created in a cylinder, no matter how short in duration, can be measured and no combustion needs to be involved in the test.

In order to measure the depression following a sudden gas outflow, a special setup was built at The Pennsylvania State College. It consists of a steel cylinder, Fig. 5, closed by a lid locked with a quick-acting latch. This lid is forced open by the initial cylinder pressure when the latch is released. Orifices of different diameters or nozzles were fixed at the lid end of the cylinder. A pickup for determining the lowest pressure reached in the cylinder during the exhaust was fixed at the closed end of the cylinder. This pickup consists of a brass diaphragm of $2^1/_4$ in. diam and 0.025 in. thick, exposed to the cylinder pressure on one side, and to an adjustable depression on the other. The diaphragm opens an electric contact when the pressure in the cylinder drops below the adjusted depression; the contact is inserted in a 6-volt circuit in series with a transformer coil. A neon lamp is placed in series with the secondary winding of the transformer. This gives a flash when the primary circuit opens.

The depressions have been measured for different orifice diameters and different initial tank pressures. The results are shown in Fig. 6 where the maximum depression after the gases have rushed out of the cylinder, are plotted against the cylinder gage pressure before the cylinder was opened. With the 6-in. orifice there was in effect no plate, as the cylinder was wide open. One curve corresponds to the use of a convergent-divergent nozzle with 2-in. throat diameter and $3^1/_8$-in. length. The opening on the pressure side was $4^1/_2$ in. and convergence took place for 2 in. Over the remaining $1^1/_8$ in. the nozzle diverged uniformly at an angle of 15 deg.

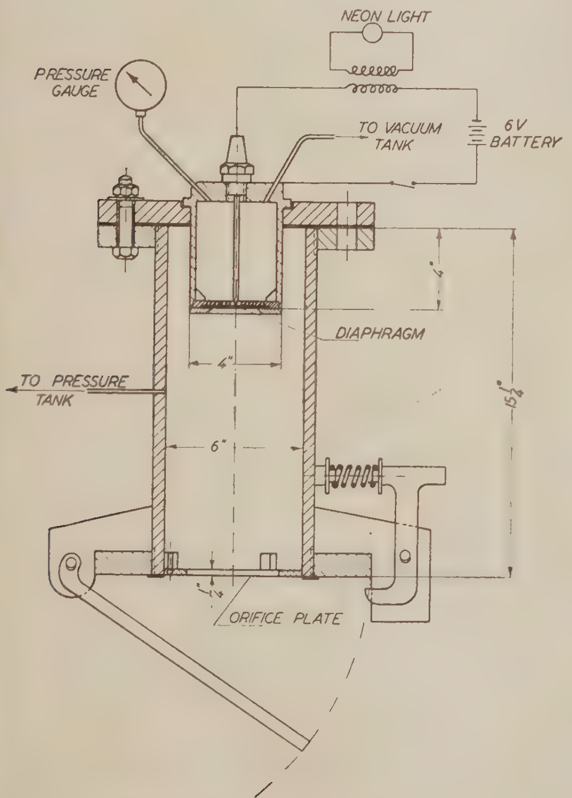

Fig. 5 Vessel With Quick-Opening Lid to Test Kadenacy Effect

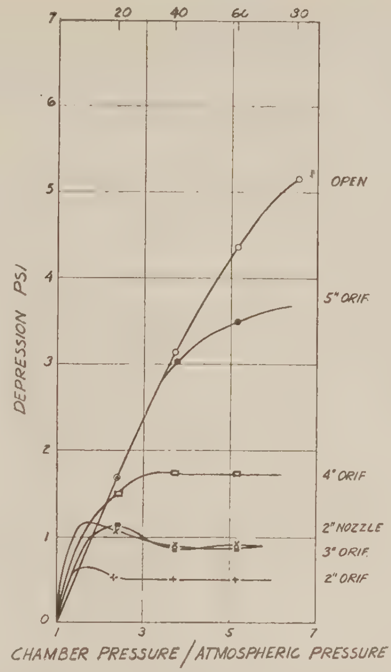

Fig. 6 Kadenacy Effect; Observed Depressions by Sudden Pressure Release, Penn State Experiments

The experiments show that the depression, although far from a complete void, is great enough to account for the results on two-stroke Diesel engines applying Kadenacy's patents.

Theory of the Kadenacy Effect

Various explanations of the Kadenacy effect have been published. Giffen (7) calculated the pressure waves generated by the sudden evacuation of a cylindrical vessel and obtained depressions of the order that we have observed. His calculations even showed that with an orifice appreciably smaller than the cylinder the depression does not increase continuously with the increase of the initial cylinder pressure but it reaches a maximum when the initial cylinder pressure is around 20 psia, and if it exceeds that, the depression begins to decrease. This agrees with our observations with orifices and nozzles of 3 in. diam or smaller.

Geyer (8) proposed a rather simple theory of the Kadenacy effect. He attributes the evacuation of the cylinder to the kinetic energy of the gases still in the cylinder, yet rushing out of the cylinder.

It is not improbable that Geyer got the idea for his theory from Kadenacy himself. For instance, in U. S. Patent No. 2,123,569, Kadenacy is using the analogy of a coil spring to explain what happens in the cylinder. He visualizes a helical spring on a table compressed, with a certain amount of energy

stored in it. If the spring is released gradually by allowing it to expand against a resistance, it will return to its free length. The work done by the spring will then become stored in the resistance. This, he states, corresponds to the release of compressed gases from a container through an orifice which is opened gradually.

On the other hand, if the spring, after having been compressed on the table, is released suddenly by removing the compressing means in a short interval of time, the spring while expanding will leave the table bodily. The energy stored in the spring imparts momentum to the spring. During its flight through the air after it has left the table, oscillations will occur in the spring but these oscillations will bear no direct relation with the motion of the spring body from the table. This case corresponds to the sudden release of the gas from the cylinder according to Kadenacy.

Geyer goes through a somewhat similar reasoning. The potential energy of the compressed gas is transformed into kinetic energy. The gas will leave the cylinder with a velocity that corresponds to this energy. At the time the pressure inside of the cylinder has dropped to atmospheric, the gas in the cylinder still has some kinetic energy, which will perform work against the atmospheric pressure. This work consists of displacing a certain volume of air against ambient pressure. The equivalent volume of air to replace the volume must come from the cylinder, and so a depression is created by the discharge of that amount of air.

Geyer's simple theory can have no pretension of describing accurately such a complex phenomenon as the sudden evacuation of a cylinder. Yet it not only gives a correct mental picture of the mechanism of the Kadenacy effect but surprisingly it even gives tolerable agreement with observed results. Geyer has applied his calculations to Davies' experiments (8), and we to the Penn State experiments (9), which are shown in Fig. 7.

In Fig. 8 is reproduced a comparison of the observed and calculated depressions when the sudden removal of the lid uncovered the full cylinder opening. With smaller orifices the observed depressions are higher than those calculated.

Ballistic Velocities

A recurrent statement in most Kadenacy patents is that the burnt gases discharge through the exhaust ports at a speed far in excess of the speed of adiabatic expansion. Patent No. 2,123,569 gives the "speed of adiabatic expansion" as 350 to 450 m per sec, and gives the "ballistic speed" of exhaust due to the "ballistic force" as 1400 to 1800 m per sec, that is about 4 times as high.

The velocity of sound in a diatomic gas of a temperature about equal to that of the exhaust gases is as a matter of fact 350 to 450 m per sec. In a straight or convergent orifice this is also the upper limit of discharge velocity of the gas.

The authors were unable to find out where figures of 1400 to 1800 m per sec had been obtained. But Kadancy uses these figures in porting design and obtains good results. This may be an indirect proof that his figures are correct. Let us examine this evidence.

Since the pressure in the cylinder when the exhaust port opens is about 5 atm, Kadenacy reasons, 4 volumes of exhaust gases have to be discharged to bring the pressure down to atmospheric during the blowdown period. It takes, he states, as much time to discharge 4 cylinder volumes of gas at 1800 m per sec velocity as one volume of gas at $^1/_4 \times 1800 = 450$ m per sec velocity. Therefore he envisages the "hypothetical velocity" of 450 m per sec and designs his exhaust ports in such a manner that the gas column which projects through the exhaust port during the exhaust-lead period with the hypothetical velocity of 450 m per sec, is just equal to 1 cylinder volume. If V denotes this volume in cubic meters, A_m the mean uncovered exhaust-port area during

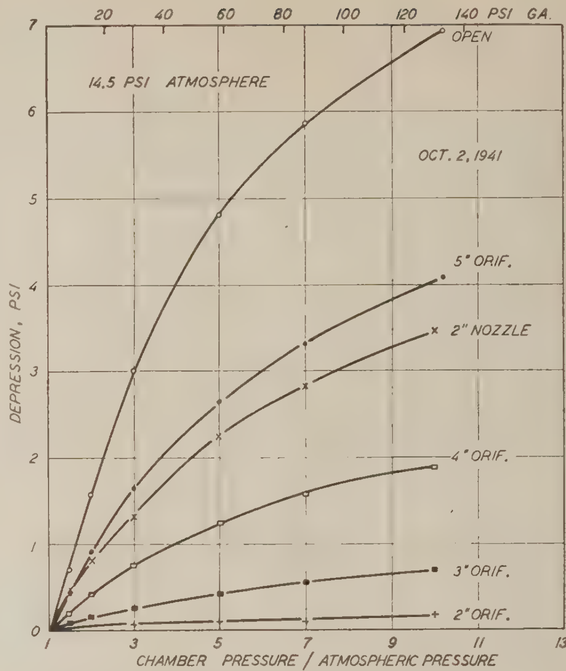

Fig. 7 Kadenacy Effect; Geyer's Theory Applied to Penn State Experiments

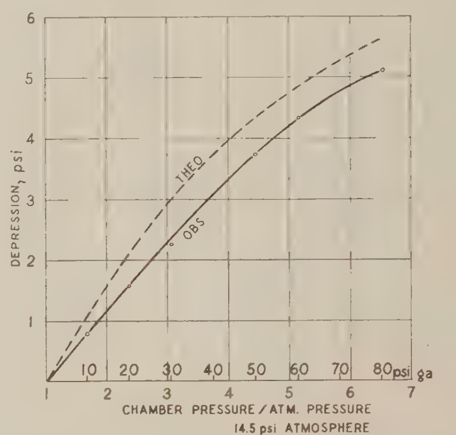

Fig. 8 Kadenacy Effect; Observed and Calculated Depressions; Open End

the exhaust lead (blowdown period) in square meters, and L the length of the hypothetical gas column in meters, then

$$V = A_m L \qquad [1]$$

the hypothetical velocity which is 450 m per sec, is given by

$$V_{hyp} = \frac{L}{t} \qquad [2]$$

where t is the time of the blowdown period in seconds, and

$$t = \frac{\alpha}{6n} \qquad [3]$$

Here α is the exhaust lead in degrees and n is the engine speed in revolutions per minute.

From Equations [1], [2], and [3]

$$V = A_m \, V_{\text{hyp}} \frac{\alpha}{6n} \ldots\ldots\ldots\ldots [4]$$

Putting $V_{\text{hyp}} = 450$ and rearranging, we get

$$A_m \alpha = V n \frac{6}{450} = \frac{V n}{75} \quad m^2 \text{ deg} \ldots\ldots\ldots [5]$$

This is Kadenacy's reasoning, and the resulting formula is identical with Kadenacy's formula Patent No. 2,144,065

$$\frac{W}{100\, KA} = \frac{a}{360\, N} = t$$

except for the different symbols. From this the proper exhaust lead can be calculated. By converting the metric units used in Kadenacy's Equation [5] to square inches we get

$$A_m \alpha = \frac{(39.4)^2}{(39.4)^3} \frac{Vn}{75} = 0.00034 \; Vn \ldots\ldots\ldots [6]$$

This is still Kadenacy's formula in a slightly changed form. On the other hand Schweitzer's formula (10) gives

$$A_m \alpha = 0.00033 \; V_{\text{disp}} n \ldots\ldots\ldots\ldots [7]$$

Ignoring the difference between V and V_{disp}, Equation [7] differs only 3 per cent from Equation [6].

Equation [6] is a consequence of the Kadenacy theory, but Equation [7] was derived independently of the Kadenacy theory, before we knew of the latter's existence. The Kadenacy theory supposes the existence of "ballistic velocities" which are about 4 times higher than those obtained from adiabatic expansion. Equation [7] was derived by conventional thermodynamics, and the discharge velocities used were substantially the same as sound velocities modified by discharge coefficients obtained by the Nusselt theory. The two theories give practically identical results.

The circumstance that Kadenacy got correct results from incorrect premises casts a doubt on Kadenacy's reasoning.

We believe Kadenacy's reasoning to be fallacious and in the following we try to show the error in it.

Fallacy in Kadenacy Theory

Kadenacy states[5] that instead of applying the true ballistic velocity, 1800 m per sec to 4 cylinder volumes of gases to bring down the pressure from 5 atm to 1 atm, one may use a hypothetical velocity of 1800/4 = 450 m per sec to 1 cylinder volume and get the same results. We believe this to be a mistake. One cylinder volume should be considered and not 4, even if the pressure in the cylinder is 5 atm.

A rough reasoning should suffice to make this clear. Let us consider two cases. In one case the density of the gas does not vary during the discharge. Before, in, and after the exhaust orifice, the density is the same. In this case (which is analogous to the discharge of water) if the cylinder volume V contains gas at 5 atm pressure and that is discharged through an orifice A with 450 m per sec discharge velocity, and $A = V/450$, then at the end of 1 sec all of the gas will be discharged and not $^1/_5$ of it only. It makes no difference whether the pressure in the cylinder was 5, 4, 3, 2, or 1 atm. In every case 450 m per sec discharge velocity is required to evacuate the cylinder in 1 sec. The assumption of a ballistic velocity of 1800 m per sec is unjustified.

[5] U. S. Patent No. 2,123,569, p. 4, lines 33–62.

Let us next consider the change in gas density. The discharge velocity refers to the throat of the orifice, and the specific weight to be considered is the one existing in the throat.

Calculation by conventional thermodynamics shows that the specific weight in the throat is approximately 0.78 kg per cu m in the beginning of the blowdown and drops gradually to approximately 0.4 kg per cu m at the end. Assuming a constant discharge velocity of 450 m per sec and a constant exhaust orifice of Am^2 during the t sec of the blowdown period, we shall discharge

$$450 \, \frac{0.78 + 0.4}{2} \, A \text{ kg of air}$$

If, conforming to Kadenacy, we make $A = V/450$, where V is the total cylinder volume, the amount discharged will be

$$D = 0.59 \; V \text{ kg}$$

while the amount that has to be discharged to drop the pressure from 5 atm to 1 atm is

$$D_1 = \frac{4}{5} V \gamma_{\text{int}} = 0.8 \; V \; 0.73 = 0.585 \; V \text{ kg}$$

It is seen that D approximately equals D_1, which means the exhaust can be disposed of in the required time period, assuming only sound velocity and no supersonic or ballistic velocity is required.

We conclude that while the value of Kadenacy's hypothetical velocity is sound and should give good results if applied to porting design, it in no way supports the existence of ballistic velocities in excess of sound velocities.

A further confirmation of this conclusion can be found in calculating from Geyer's theory the outflow velocities created by the sudden opening of an air-filled vessel. Fig. 9 shows these velocities, and for comparison, also the calculated sound velocities for various temperatures. For the detailed calculation the reader is

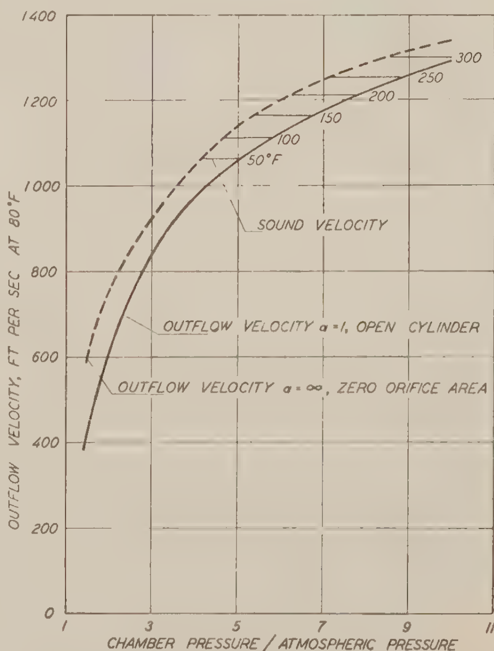

Fig. 9 Outflow Velocities Created by Sudden Opening of Vessel Filled With Air, From Geyer's Theory

referred to (9). An inspection of Fig. 9 reveals that with 5 to 7 atm chamber pressure the outflow velocities do not differ much from the sound velocities.

Conclusions

1 The astonishing performance claimed by Kadenacy engines should not be rejected as incredible as the effect is based upon a demonstrable physical phenomenon; the rarefaction that follows the sudden discharge of compressed gas from a closed vessel.

2 The Kadenacy effect is not due solely to the inertia of the gas column in the exhaust pipe, as 10 in. Hg depression was observed by the authors in a vessel without any exhaust pipe, and reported results show good performance in engines with varying lengths of exhaust pipes.

3 The Kadenacy effect vanishes if the discharge opening becomes relatively small. One inference is that an obstacle of the full exploitation of the Kadenacy effect is that exhaust ports or valves in engines cannot be opened rapidly enough.

4 Geyer's theory seems to give a fair picture of the mechanics of the Kadenacy effect and excellent numerical agreement in case of a large discharge opening.

5 Nothing in our observations supports Kadenacy's contention that velocities in excess of sound velocities are involved in the exhaust process. Conventional thermodynamics based upon simple adiabatic expansion gave formulas practically identical with those recommended by Kadenacy, and Geyer's theory applied to our experiments also successfully indicated discharge velocities of the order of sound velocities and no higher.

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Discussion

Paul Diserens.[6] The authors' discussion of the Kadenacy effect is particularly timely in view of the development of two-cycle engines, sponsored by Mr. Kadenacy, on the part of several American manufacturers at this time.

It would appear that the paper is, in effect, a continuation of the extended discussion of the Kadenacy engine which was reported in the British and the Continental technical press during the years 1938 to 1941, inclusive. While the authors make some reference to the numerous articles published at that time, it might be suggested that a bibliography comprising the more important references would constitute a valuable addition to the paper.

The writer finds it of particular interest to compare the apparatus used by the authors in their experiments, with that employed by Prof. S. J. Davies, of Kings College, University of London.[7]

It should be noted that in the Davies apparatus provision is made to insure the greatest possible speed in establishing communication between the pressure vessel and the atmosphere. In the apparatus used by the authors the speed of opening is limited because of the time required to accelerate the communicating valve.

It is evident that the results reported in the paper reflect the influence attributable to the speed of opening and therefore the results might conceivably be quite different if some other speed had been employed. Consequently, conclusion 3, i.e., "The Kadenacy effect vanishes if the discharge opening becomes relatively small." One inference that full advantage cannot be taken of the Kadenacy effect because "exhaust ports or valves cannot be opened rapidly enough," cannot be valid except when the exhaust-valve acceleration at the time of opening is low or at best comparable with that which obtained during the authors' experiments. For this reason it would appear that their results cannot properly be used as the basis for any general theory to explain the phenomena and should not be quoted as confirming such a theory based on pure speculation.

The authors assert that the hypothetical assumed average speed of 450 m per sec is derived from some preconceived theory of operation, and quote from various Kadenacy patent specifications in support of this assumption. This does not conform to my understanding of the Kadenacy patents. From a complete study of all of these it is obvious that the structures covered by the patents rest entirely upon experimental data from which empiric relationships are established. Obviously, variable coefficients must be determined in order to take care of such variable conditions as, for example, speed of opening of the exhaust valve.

An experimental engine, built by the company with which the writer is associated, develops mean effective pressures of more than 100 psi without a blower. Analysis of the exhaust gas, as well as direct measurements indicates that the volume of air passing through an engine exceeds the displacement by 60 per cent.

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[6] Director of Research and Development, Worthington Pump and Machinery Corporation, Harrison, N. J. Fellow A.S.M.E.

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An Analysis of Intercooled Supercharging

By RALPH H. MILLER,[1] MILWAUKEE, WIS.

In this and in an earlier paper,[2] the author analyzes the internal-combustion engine as a heat engine. He finds that with present-day materials, the output of the internal-combustion engine is limited by the temperatures reached in the internal surfaces and not by lack of air to sustain combustion. A method is developed by which the output capacity of four-cycle engines may be increased by more than 100 per cent without exceeding practical and previously established temperatures of internal surfaces.

In a previous paper,[2] the author summed up his analysis of intercooled supercharging in a rating graph which is reproduced in Fig. 1, herewith.

Curve *A* in the graph shows that with standard supercharging without air cooling 133 per cent of the mean indicated pressure (mip) is carried at 4 psi pressure with the same cycle mean temperature. This represents a gain of 36 per cent in brake mean effective pressure (bmep). Since that paper was presented a series of tests has been conducted to check the theories presented therein.

Efforts were made to correlate calculated cycle mean temperatures and the surface temperatures of the internal walls, which are obviously what affect the engine parts and not the temperature of the gases.

An engine operating at 600 rpm, and another engine tested at 327 and 360 rpm were used in these tests. Equipment was installed to measure the heat flow to the cooling-water jackets with great accuracy. Means were also installed for controlling the air-intake temperature to the blower or the engine manifold.

The cooling-water mean temperature was kept at 155 F and the rate of circulation maintained constant for all loads and speeds.

The formula for rate of heat transmission

$$dq = U \times dA\ (t_1 - t_0)$$

where

dq = Btu per unit of time
dA = area
$U \times dA$ = over-all conductance of wall per unit of time
t_1 = temperature of internal hot surface
t_0 = temperature of cooling water

permits us to conclude that the internal-wall temperature t_1 is unchanged when, in any given engine, changes are made in manifold pressure or temperature or combustion efficiency, but the load is adjusted so that dq remains constant.

The rate of heat flow to the cooling water (liners and cylinder heads) is plotted in Fig. 2 for the 600-rpm engine. Curve 1 shows the naturally aspirated engine from 58 to 85 bmep. Curve 2 is the engine supercharged with the Buchi system with two exhaust pipes to the turbine. Curve 3 is the same as curve 2 but with intercooling to 90 deg F in the air-inlet manifold.

The naturally aspirated engine is rated at 80 bmep. The heat flow to the cooling water (dq) is 12,000 Btu per min at this load. When the engine is operated supercharged with the Buchi system, a heat flow of 12,000 Btu is reached at a load of 102 bmep. Curve 1 in Fig. 3 shows a supercharging pressure of 6.7 in. Hg (3.3 psi) at 102 bmep.

The theoretical rating curve, Fig. 1, shows that on the basis of equal cycle mean temperature of the gases, a rating of 131 per cent of the naturally aspirated indicated rating should be carried at 3.3 lb supercharging pressure.

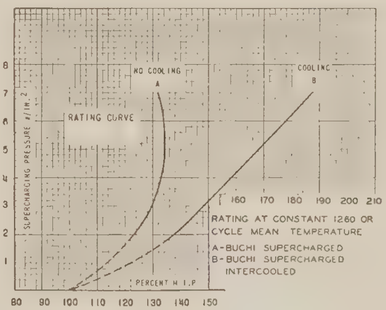

Fig. 1

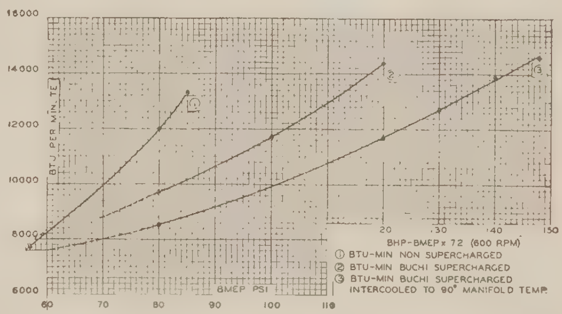

Fig. 2

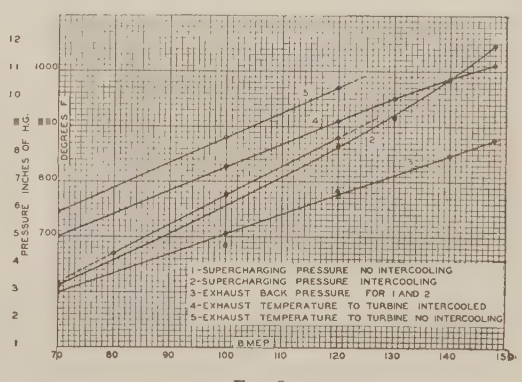

Fig. 3

[1] Chief Engineer, Four Cycle Diesel Division, Nordberg Manufacturing Company. Mem. A.S.M.E.

[2] "Rating Supercharged Engines on the Basis of the Mean Temperature of the Cycle," by Ralph Miller, Trans. A.S.M.E., vol. 65, 1943, pp. 685–696.

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Assuming a friction mean effective pressure (mep) of 20 psi at 80 bmep and that it increases with the square root of the bmep the mip at 12,000 Btu per min is as follows:

	Mip from Fig. 2	Mip from Fig. 1
Naturally aspirated, $80 + 20 =$	100	100
Buchi supercharged, $102 + \left(20 \times \sqrt{\frac{102}{80}}\right) =$	125	131
Buchi supercharged intercooled, $123 + \left(20 \times \sqrt{\frac{123}{80}}\right) =$	148	158

This shows a difference of $4^1/_2$ per cent in the noncooled and $6^1/_2$ per cent in the intercooled ratings between mip calculated from cycle mean temperatures and actual tests, recording heat flow to cooling water.

This discrepancy is probably caused by increased afterburning and diminishing expansion coefficient with the later cutoff when supercharging. The same cause brings about the nonparallelism of the curves 1 and 2, that is, the combustion efficiency falls off at a faster rate in the nonsupercharged than in the supercharged engine.

In Fig. 4 the rate of heat flow to the cooling water is plotted versus bmep. Curves 1 and 2 show the heat flow nonsupercharged and supercharged at 360 rpm. Curves 3 and 4 are plotted for 327 rpm, nonsupercharged and supercharged.

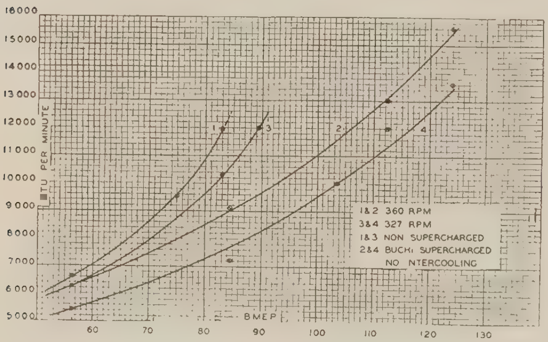

Fig. 4

If a heat-flow rate of 10,000 Btu per min is taken as a maximum for full load the bmep ratings will be as follows:

Engine rpm	327	360
Nonsupercharged oil Diesel, psi	82	77
Buchi supercharged oil Diesel, psi	103	93

The low bmep of this engine when supercharged would seem to be due to insufficient scavenging.

Intercooled Supercharging

At 4 lb supercharging pressure the bmep can be increased between 21 and 26 psi at the same heat flow to the cooling water when the supercharging air in the inlet manifold is cooled to the air temperature at the blower intake.

The graphs in Fig. 5 show manifold temperature versus bmep developed by the high-speed engine at two different rates of heat flow to the cooling water, namely, 10,000 and 12,000 Btu per min. Selecting for an example, the test point at 126 F manifold temperature and 105 bmep with a heat flow of 12,000 Btu per min, a cycle mean temperature of 1262 deg R is calculated as follows:

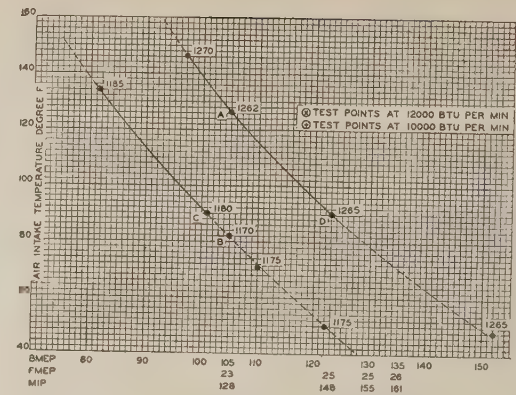

Fig. 5

$$\text{Mip} = 105 + \left(20 + \sqrt{\frac{105}{80}}\right) = 128$$

P_1 = supercharging pressure; nonintercooled line, Fig. 3 = 3.45 psig

Cutoff for 128 mip and 3.45 psi, Fig. 6 = 14.18

Ratio Tm/T_1 for cutoff 14.18, Fig. 7 = 1.92

$T_1 = 126 + 460 + t_{ch}$ = (Equation [9], reference 2) = 657 deg R

$Tm = 1.92 \times 657 = 1262$ deg R

The 10,000-Btu line shows a manifold temperature of 82 deg F at the same load. The initial temperature and the cycle mean temperature at this point are then

$T_1 = 657 - (126 - 82) = 613$ deg R

$Tm = 613 \times 1908 = 1170$ deg R

(The supercharging pressure P_1 has dropped from 3.45 to 3.3, Fig. 3, so that the Tm/T_1 ratio changes from 1.92 to 1.908.)

Reducing the manifold temperature from 126 to 82 deg F has reduced the cycle mean temperature from 1262 to 1170 deg R, and the heat flow to the cooling water from 12,000 to 10,000 Btu per min. We then have

$$\frac{12{,}000}{10{,}000} = \left(\frac{1262}{1170}\right)^n$$

where $n = 2.4$.

In other words, when the manifold temperature is reduced the heat flow to the cooling water, and therefore the internal wall temperature, decreases with the power of 2.4 of the cycle mean temperature.

Now, when the load is increased from C to D at 82 deg F manifold temperature to the point where the heat flow to the cooling water is again 12,000 Btu per min, it is seen that heat flow increases with approximately the same power of the cycle mean temperature, as follows:

At point C on the 10,000-Btu line, we have

$T_1 = 90 + 460 + 71.5 = 621.5$ deg R

P_1 = supercharging pressure (from Fig. 3) = 3.1 psig

Mip = 101 + 22.5 = 123.5 psi

Ratio Tm/T_1 (from Fig. 6) = 1.90

Then

$Tm = 621.5 \times 1.9 = 1180$

Point D on the 12,000-Btu line and 90 deg F air temperature reads 123 bmep; then

Mip = 123 + 25 = 148 psi

P_1 (from Fig. 3) = 4.2 psig

$T_1 = 90 + 460 + 70.5 = 620.5$ deg R
Ratio $Tm/T_1 = 2.04$
$Tm = 620.5 \times 2.04 = 1265$

The cycle mean temperature has increased from 1180 to 1265 deg

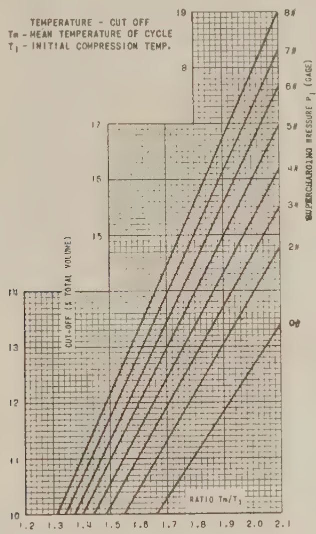

FIG. 6

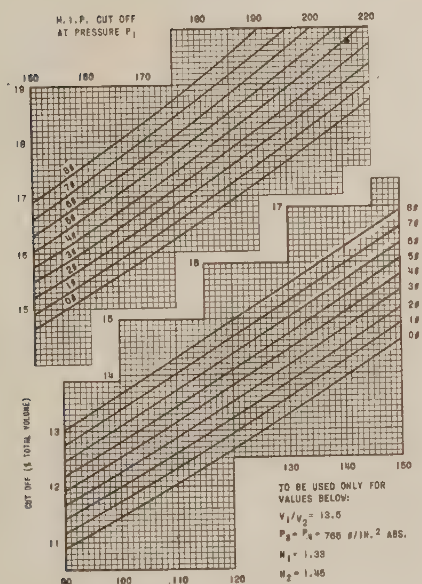

FIG. 7

R, and the heat flow from 10,000 to 12,000 Btu per min, and we have

$$\frac{12{,}000}{10{,}000} = \left(\frac{1265}{1180}\right)^n$$

where $n = 2.6$.

This proves that when the air-intake temperature is changed the load may be adjusted along the line of constant calculated

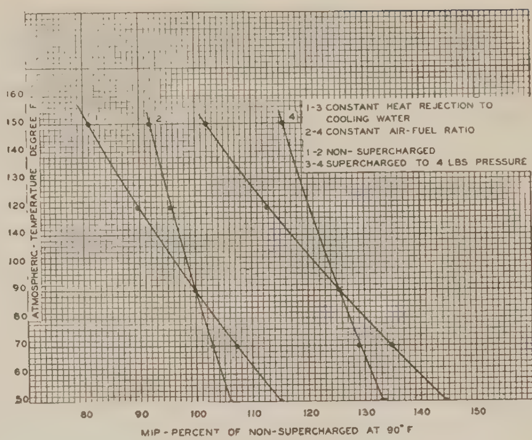

FIG. 8

cycle mean temperature. This loading will maintain the internal temperatures constant as indicated by the rate of heat dissipation to the cooling water.

The constant heat-flow curves in Fig. 5 will show an increase in initial compression pressure with increase in load which is characteristic of turbocharged engines. In Fig. 8 constant cycle temperature lines have been plotted against atmospheric intake temperature and mip for constant manifold pressures; 90 deg F is taken as standard and the point of rating. Curves 1 and 2 are for nonsupercharged and curves 3 and 4 for a supercharged engine. Curves 2 and 4 show the ratings which would be obtained by the conventional method which assumes that the mip is inversely proportional to the absolute temperature, or directly to the air density. This method does not derate sufficiently for temperature increases. Thus if an engine is operating with 115 deg F intake temperature, which is not unusual, it suffers a loss of 8½ per cent in load capacity. Conversely, when the temperature is decreased the load-carrying capacity increases at a rate faster than the air density.

FUEL CONSUMPTION

Table 1 summarizes data from a nonsupercharged, two noncooled, and three intercooled Buchi supercharged tests.

Although the theoretical cycle thermal efficiency is unaffected by the initial compression temperature,[3] these test results show an increase from 34.2 to 35 per cent brake thermal efficiency when cooling the intake air to 90 deg F, at 120 bmep. The fuel consumption is reduced from 0.377 to 0.371 lb per bhp per hr. The heat flow to cooling water drops from 14,500 to 11,700 Btu per min, and the calculated cycle mean temperature from 1365 to 1270 deg R.

When the bmep is increased from 105 to 126 along the 12,300-Btu per min heat-flow line in Fig. 2, the fuel consumption drops to 0.37 lb per bhp per hr, Fig. 9.

The lowest rate of heat flow to the cooling water is 11.15 per cent of the total heat at 147 bmep.

It will be noticed that the excess air is reduced with intercooling. Thus the uncooled engine at 105 bmep shows 27.1, and the cooled engine at 126 bmep 26.4 lb of air per lb of fuel, both at 12,300 Btu heat flow to cooling water.

Where a cooling medium of low temperature is available, such as in cold-storage or ice plants where brine may be used, or in

[3] "Internal Combustion Engine," by D. R. Pye, second edition, Oxford University Press, New York, N. Y., vol. 1, 1937.

TABLE 1 TEST DATA FOR 12-IN. × 14-IN. SIX-CYLINDER ENGINE AT 600 RPM

	Buchi supercharged — Noncooled			Buchi supercharged — Intercooled		
Brake mean effective pressure	80	105	120	120	126	147
Mean indicated pressure	100	128	144.5	144.5	151	174
Brake horsepower	576	756	864	864	907	1060
Indicated horsepower	720	921	1040	1040	1087	1253
Fuel consumption per bhp-hr	.386	.375	.377	.371	.3705	.376
Fuel consumption per ihp hr	.309	.313	.313	.308	.308	.317
Total Btu min hhv 19,600	72500	92700	106500	104500	110000	130000
Btu to cooling water	12000	12300	14500	11700	12300	14500
Btu to cooling water per cent of total	16.6	13.3	13.6	11.2	11.2	11.15
Brake thermal efficiency, per cent	33.7	34.7	34.4	35	35	34.5
Supercharging pressure	0	3.45	4.2	4.075	4.38	5.75
Air-intake temperature gage	90	140	147	90	90	90
Initial compression temperature T_1	675	671	677	621	619	617.5
Ratio T_m to T_1	1.895	1.91	2	2.06	2.07	2.19
mip from Figs. 6 and 7	100	127			151	
Total weight of air per minute	101	165	170	184.2	188	200
Weight of air per minute retained in cylinder	101	129.5	134	145	148	158.5
Weight of fuel per minute	3.71	4.76	5.46	5.33	5.6	6.65
Lb of air per lb of fuel	27.2	27.1	24.5	27.2	26.4	23.8
Volume of air, cfm total	1395	2210	2270	2460	2510	2675
Btu removed from air, min				2500	2750	3700
Btu removed per hp per min from air				2.90	3.03	3.5
Btu per cent of total cooling water plus air	16.6	13.3	13.6	13.6	13.6	14.2

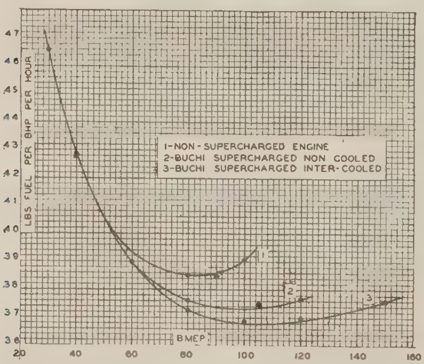

Fig. 9

places where cold well water is available, the air cooling may be carried below 90 deg F to advantage. Fig. 5 shows by extrapolation that with an air-intake temperature of 50 deg F, about 145 bmep is carried with the heat load of 105 bmep on the noncooled Buchi supercharged engine.

In Fig. 10 is plotted the fuel consumption in hhv per hour and per bhp per hour recorded on the intercooled supercharged engine at 327 rpm. The cooling-water heat-flow line for this run is shown on curve 4 in Fig. 4.

The author promoted and directed this development which was started in 1943. The first tests were made in January and February, 1945. The fuel consumption obtained at that time is plotted in curves 3 and 4 in Fig. 10. The engine was operated with air intercooling. At 130 bmep the fuel consumption was 6500 Btu hhv per bhp per hr. This may be the lowest fuel consumption ever recorded on a gas engine.

Without air cooling the consumption will increase about 3 per cent to 6700 Btu. This latter value has recently been confirmed by another enginebuilder who has since taken up the supercharged gas Diesel.

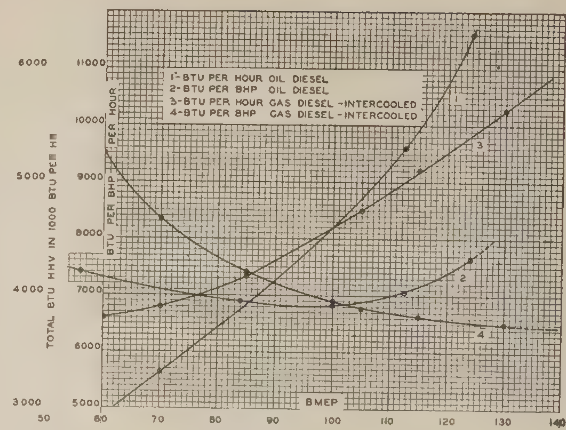

Fig. 10

On the basis of cycle mean temperature, the supercharged gas Diesel will carry a higher mep than the oil Diesel. This is due to better combustion efficiency at high loads.

High-Pressure Supercharging

Points A, B, and C, curves 1 and 2, Fig. 11, are transposed test points on the 12,000-Btu line in Fig. 2. By calculating T_1 for these points, the mean temperature Tm is found, thus permitting other points on the curves to be calculated from Figs. 6 and 7.

If the nonsupercharged engine could be scavenged, the mip would increase from 100 to 109. The curve starting at 109 and 0 supercharging pressure shows the gain obtained by increasing the pressure. The highest load occurs at 5 lb, where the mip is 125. Of this, 9 lb is gained by scavenging and 16 by increasing the supercharging pressure. Above 5 lb gage the load drops off.

By cooling the air in the manifold to 90 deg F or to the intake temperature of the blower, the mip at constant cycle mean temperature will follow curve 2. The 148-mip point is established from the actual heat-flow test curves in Fig. 2. The maximum cylinder pressures are equal at 765 psi for curves 1 and 2 in Fig. 11. The cutoff ratio and compression pressure versus scavenging pressure for rating line 2 is shown on lines 7 and 6. A maximum of 177 mip is reached at 9.5 lb supercharging pressure when the compression pressure equals the maximum cylinder pressure of 765 lb.

Therefore, when maintaining the maximum cylinder pressure of the nonsupercharged engine, the limit of supercharging pres-

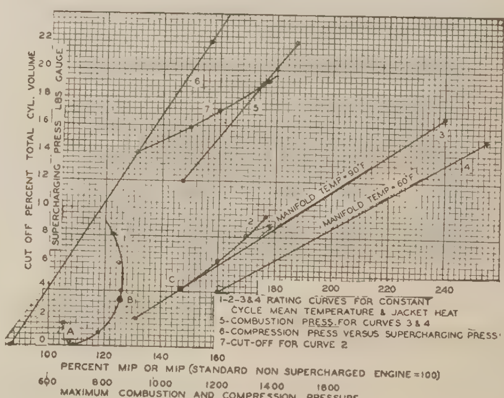

Fig. 11

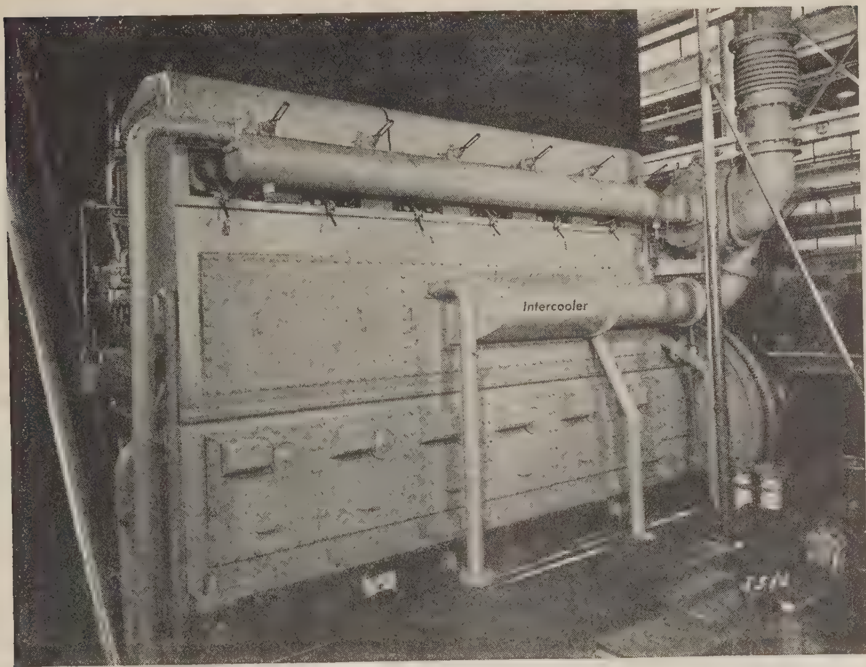

FIG. 12 SUPERCHARGED DIESEL ENGINE WITH INTERCOOLER

sure, with intercooling to blower intake temperature, is reached at about 9.5 lb gage.

Higher supercharging pressures can be used only by permitting the maximum combustion pressure to increase.

The 147-mip point on curve 2 in Fig. 11 has a combustion-to-compression pressure ratio of 1.266. If this ratio is maintained with increasing supercharging pressure and constant air-manifold temperature, we find that with constant cutoff (15.5 per cent) the mip for constant cycle temperature increases directly with the absolute supercharging pressure. The cycle mean temperatures are identical at all loads on curve 3, and the thermal efficiency is constant.[3]

Line 4 in Fig. 11 is plotted for 60 deg F air-intake temperature to indicate the as yet unexplored possibilities of high output of the four-cycle engine. For example, with a supercharging pressure of 22 psig a mip of 318 psi would be carried. The compression pressure would be 1170 psi, and the maximum combustion pressure 1480 psi. The heat flow to the cooling water would be about 5.6 per cent of the total, and the fuel consumption 0.32 lb per bhp per hr; 45.4 per cent of the total heat would go to work and 49 per cent to exhaust and radiation. While these figures will have to be adjusted downward to compensate for increased heat-transfer rate from gas to the walls, due to the increase in gas density, high-pressure supercharging nevertheless holds great promises.

CONCLUSIONS

On the basis of constant internal temperatures, the nonair-cooled Buchi supercharging system permits an increase of 28 to 30 per cent on the brake horsepower of the nonsupercharged engine, or an increase of between 20 and 28 bmep. The cost of equipping an engine with a turbocharger and the Buchi exhaust-pipe system is very nearly in the ratio of this increase so that the final cost per bhp of the supercharged engine is approximately the same as the cost of the standard nonsupercharged engine.

By the simple expedient of adding to the supercharging equipment an air cooler whose dimensions may be 15 in. square × 4 ft long for 2600 cfm, and supplying cooling water at 80 deg F to remove 3 Btu per min per bhp, the engine output is increased to 155 per cent of nonsupercharged rating.

The cost of this cooling equipment will be about $3 to $4 per hp, gained by cooling. The author believes this to be the lowest cost on record for a Diesel-engine horsepower. The extreme simplicity of equipment and ease of operation and maintenance should bring about universal acceptance of intercooling and render the uncooled turbocharged engine obsolete.

Compared with the nonintercooled turbocharged engine, intercooling to 90 deg F reduces the cost per horsepower by about $18^1/_2$ per cent. The weight and volume of the engine are reduced about 20 per cent for the same horsepower.

Beyond this lies high-pressure intercooled supercharging using multiple-stage turbochargers, which promises in the near future to revolutionize the design of internal-combustion engines. Those engineers who accept the challenge of this discovery and bend their efforts toward the solution of the problems involved in developing this new engine will help to guide the internal-combustion-engine industry toward greater expansion and increased importance in our civilization.

Discussion

G. J. HAISLMAIER.[4] The author analyzes the internal-combustion engine primarily as a heat engine. As such it is important that we maintain a balance of heat, adding it where necessary and removing it where it is undesirable. In the case of the supercharger intercooler, that is, the cooling unit itself, we are interested primarily in the removal of heat and generally

[4] Sales Manager, Contract Products Division, Young Radiator Company, Racine, Wis.

speaking, removing as much as possible. In terms of heat transfer we find the engine jacket water at a temperature level too high to afford a means of cooling the supercharged air. We look therefore to an outside source of water, normally the raw-water supply which is used for cooling the engine jacket water and the oil coolers, or in the case of a marine engine, the sea water, in which case we must be careful to design into the intercooler unit a construction with materials that are resistant to salt-water corrosion.

The intercooling of supercharged air is comparatively new; however, we did accomplish this and in production-lot quantities, starting back in 1939, on the smaller high-speed 4-cycle Diesel engines ranging in the neighborhood of 150 to 200 hp. Later on during the war period we did this for other engines also in the higher-speed Diesels, namely, in the 300-hp range. Fig. 13 of this discussion shows a small cooling unit used with a 300-hp marine supercharged Diesel engine. The core element of this unit is approximately 6 in. square and this will give a relative idea of the over-all size of the cooling unit. This unit is capable of handling anywhere from 300 to 600 cfm of supercharged air and of cooling this down to within about 25 deg of the temperature of the sea water. The unit was designed and manufactured entirely of salt-water-corrosion-resistant materials and, in being furnished for the requirements of the United States Navy, was built in compliance with the applicable specifications of Heat Exchanger Specification 66Cl of the Heat Transfer Section of the United States Navy, Bureau of Ships.

Fig. 14 shows the performance curves for this unit, which basic data have served well in approximating the size of the intercooler units being furnished for some of the larger slower-speed Diesel engines today. It will be noted that all of the characteristics are plotted against engine rpm. The sea-water temperature rise is negligible. The sea-water flow rate is such as dictated by the other parts of the cooling system. The air-pressure drop is within an acceptable range, and it will be noted also that the sea-water pressure drop is almost negligible.

In the meantime some of the engine designers had indicated a preference for an intercooler unit which would be round in shape, and from a design standpoint this seems to be the logical arrangement of heat-transfer surface, because of the over-all shape of the unit, the straight-through air flow, and the ease of installation to the engine. Naturally this round-type intercooler would be more desirable, fitting readily and neatly into the air duct from the supercharger blower to the engine intake manifold, and it is generally conceived by the engine designer as a unit which would be about the same diameter size as the air duct, or at least not much larger. An intercooler of this design has been developed, and is shown in Fig. 12 of the paper. This has been applied to one of the larger supercharged Diesel engines. The cooler itself, Fig. 15 of this discussion, is approximately 22 in. diam with approximately 1100 tubes, $^1/_4$ in. diam, making up the tube nest, the salt water from the engine cooling system flowing through the flanges as shown and the air flowing through the tubes. The unit handles approximately 2000 cfm with a pressure drop of about 8 in. water gage, and cools the air to within about 20 deg F of the water temperature.

As yet this type of unit has not been made available in production-lot quantities and there is some doubt that it will be, at least in its present shape and form. It must be remembered that the raw water, or the salt water in marine applications, passes around the smaller tubes in very narrow and restricted passages inside of this round unit. So far the design of this round-type unit has not been cleanable from the sea-water side. However, that factor is being given further consideration and the round-type intercooler may yet be available for the engine designer in the not too distant future.

As in all engineering considerations, design too is a matter of compromise, and in the design of the supercharger-intercooler unit itself the compromise is that of shape and size with performance in terms of cooling efficiency as well as of maintenance and serviceability in operation. There is also a compromise in the arrangement of the heat-transfer surface, that is, a compromise in over-all height and width against pressure drop in the air stream passing through the cooling unit.

The following is what might be considered an over-all compromise in terms of one of the latest intercooling units now being applied to one of the larger slow-speed Diesel engines. This engine, a six-cylinder model, 16-in. bore, 22-in. stroke, operating at 325 rpm when supercharged, has a rating of 1200 hp as against the 750-hp rating unsupercharged. With supercharger intercooling, it is expected to bring the rating of this engine up to 1500 hp at the same engine speed. The intercooler unit has over-

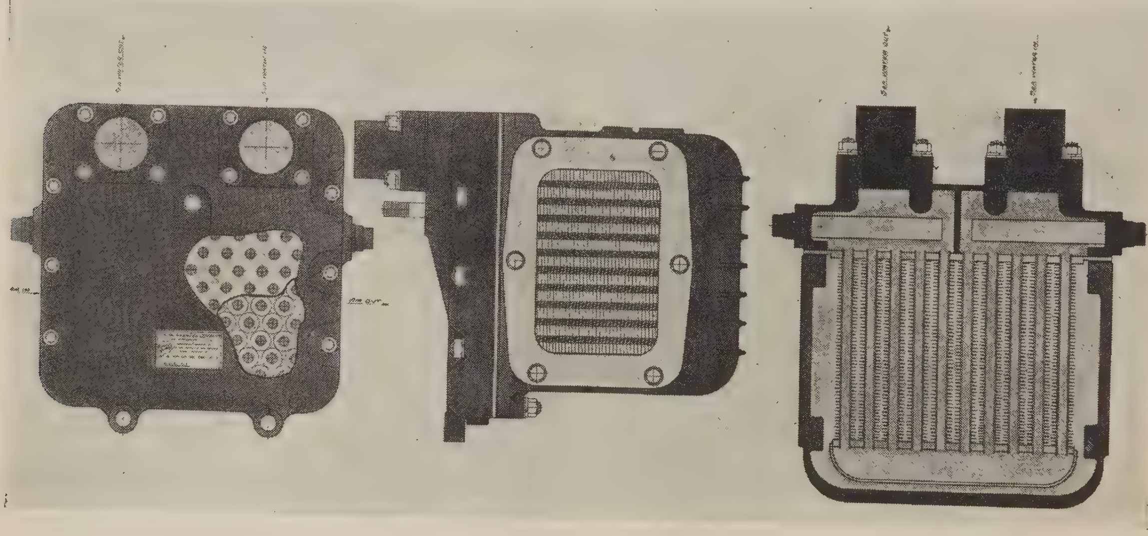

Fig. 13 Details of Supercharger Intercooler Heat Exchanger

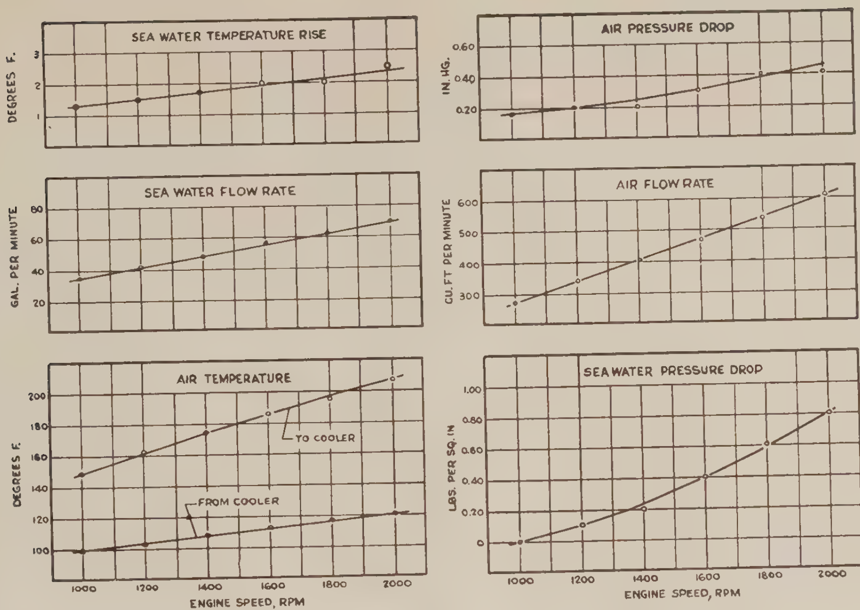

Fig. 14 Performance Curves of Intercooler Heat Exchanger

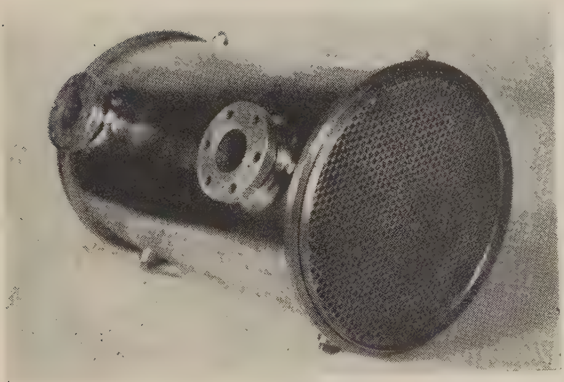

Fig. 15 Round-Type Intercooler Unit

all dimensions of 53 in. length, $18^1/_2$ in. height and depth in the direction of air flow of $7^1/_8$ in., although the core element itself to which the supercharged air stream is exposed, measures only 48 in. between the end headers, $16^1/_8$ in. height and $5^1/_8$ in. depth in the direction of air flow.

This unit in over-all volume is somewhat smaller than the intercooler unit to which the author referred in his paper, and it is offered to give cooling performance even a little better than that same unit. More specifically, it is designed to handle an air flow of 3900 cfm with an air-inlet temperature coming from the supercharger at 155 deg F, cooling the supercharged air down to 90 deg F. The cooling-water temperature maximum is 80 deg F, and with lower water temperatures the leaving temperature of the supercharged air coming off the cooling unit will of course be less, that is, approximately 10 to 15 deg above the water temperature.

In this particular case we have figured on a cooling-water flow of from 200 to 250 gpm, which is the same water flow available for the engine jacket-water heat exchangers and the lube-oil heat exchangers. The raw water would flow through the supercharger intercooler before passing through the other heat exchangers as there is only about a $2^1/_2$ to 3 deg F temperature rise in the cooling water passing through the supercharger intercooler. This unit, incidentally, has approximately 150 tubes, $^3/_8$ in. diam, and in its performance we estimate the pressure drop on the air side will run in the neighborhood of $2^1/_2$ to 3 in. water gage.

The weight of this intercooler unit is approximately 350 lb. As yet it is not being manufactured in production-lot quantities, but even on the smaller-quantity basis the cost figures today indicate that the \$2 per hp figure which the author quotes is not far out of our reach. In fact, depending upon the size of engine and the quantity manufactured at one time, we find current costs on intercooler units of this kind running from \$2.50 to about \$5 per hp, based upon the increase of horsepower which will be obtained from the intercooling alone. This does not take into account the cost of the application and the additional materials such as the duct work and manifolding which the engine manufacturer will have to consider in his design costs. It is a figure based on the cost of the intercooler itself as an accessory item.

This cost, however, will be less as time goes on, particularly as plans are being made eventually to work out a standard intercooler design which can be made adaptable to more than one engine installation and which conceivably may be manufactured in larger quantities, resulting in ultimately lower costs. It is not unreasonable therefore to look forward to approaching the figure of \$2 per hp or possibly even improving upon that figure.

Bavarian Motor Works Altitude-Test Facilities

By E. E. STOECKLY,[1] WEST LYNN, MASS.

Several large-capacity high-altitude low-temperature test chambers for aircraft-engine work were constructed during the war. The present paper describes in some detail the equipment and operation of the first of these at the Bavarian Motor Works.

Early in the year 1941 the German Air Ministry authorized the construction of the first large-capacity high-altitude low-temperature aircraft-engine testing laboratory. This decision was predicated on the conviction that the further rapid development of high engine powers at altitudes could most rapidly be accomplished by ground-level testing under actual high-altitude low-temperature high-speed flight conditions. Of paramount importance was the cooling problem of both air- and liquid-cooled conventional engines, and the satisfactory performance and operation under altitude conditions

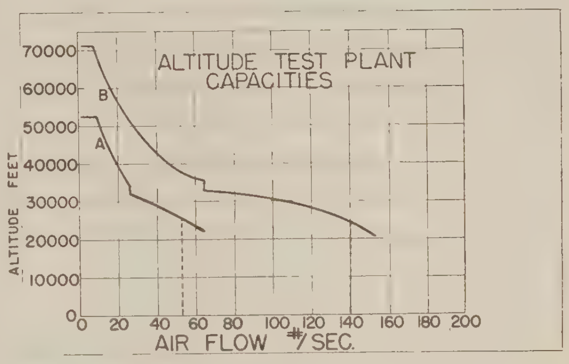

Fig. 1 Test-Plant Capacities

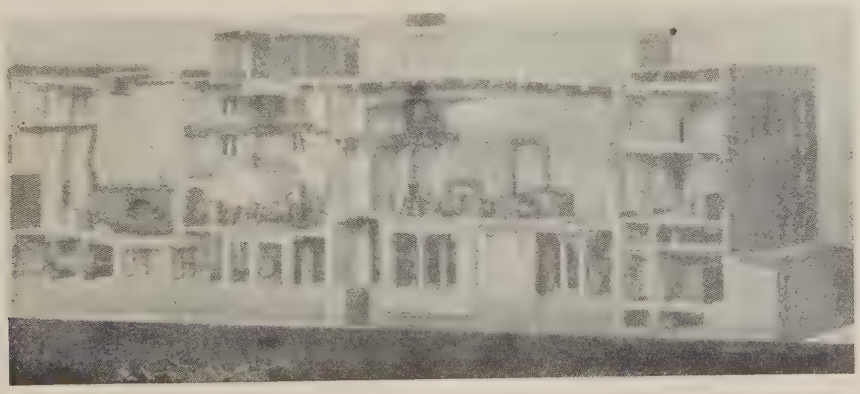

Fig. 2 Cross Section of Test Plant

of their then little-known jet-propulsion engine. Prior to 1941 all laboratory altitude testing consisted of holding only the engine exhaust and carburetor air inlet at altitude conditions. Because of the experience and success that the B.M.W. had with such testing, it was directed to build the first really large altitude-test plant. The plans called for a maximum refrigerated air supply of 53 lb per sec at —80 F, and an evacuation capacity of 53 lb per sec at 26,000 ft altitude, with a maximum operating altitude of 52,500 ft, as given by curve A, Fig. 1.

Shortly after authorizing the first plant, the German Air Ministry, often referred to as R.L.M., authorized the construction of four more such plants, some of which were to be over twice as large as the Bavarian Motor Works (B.M.W.) plant. The new plants were to be for the research establishment of the Luftwaffe, known as E.D.L; German Experimental Establishment for Aviation, know as D.B.L; Daimler Benz and Junkers Motor Works. This policy was to make altitude-testing facilities available to the three leading German reciprocating and gas-turbine aircraft-engine manufacturers, as well as the two leading government research institutions. The two government plants were to have exhausting and refrigerating capacities as shown by curve *B*, Fig. 1. The plants at the three engine manufacturers were to be as given by curve *A*, Fig. 1, with space and foundations provided for eventually increasing the plant capacity to curve *B*, Fig. 1.

Between 1941 and 1943, as the hopes for larger engines rose and fell, doubling the capacity of the B.M.W. plant was authorized and canceled several times. Some 3 years after construction was started, the B.M.W. plant, a cross-section model of which is shown in Fig. 2, went into operation in October, 1944, with a refrigeration and evacuation capacity of 53 lb per sec at 26,000 ft altitude and a maximum working altitude of 52,500 ft. The plant operated successfully from the very beginning and was in continuous service up to the time of occupation by the American forces, except for short outages to repair bomb damage. Be-

[1] Engineer, Aircraft Gas Turbine Division, General Electric Company. Mem. A.S.M.E.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

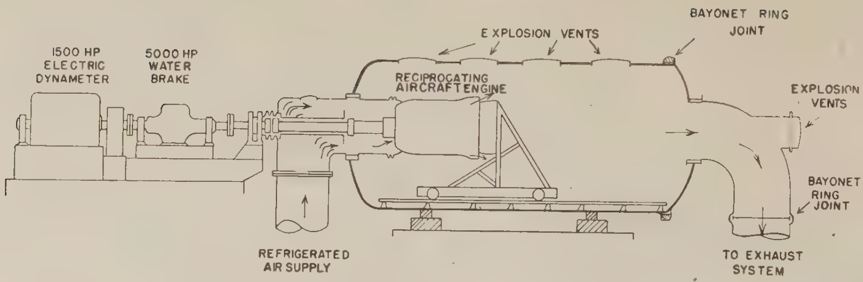

Fig. 3 Altitude Test Chamber for Reciprocating Engines

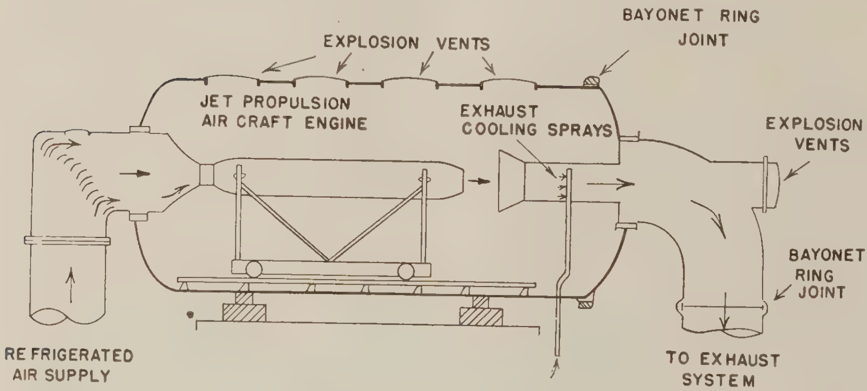

Fig. 4 Altitude Test Chamber for Jet Engines

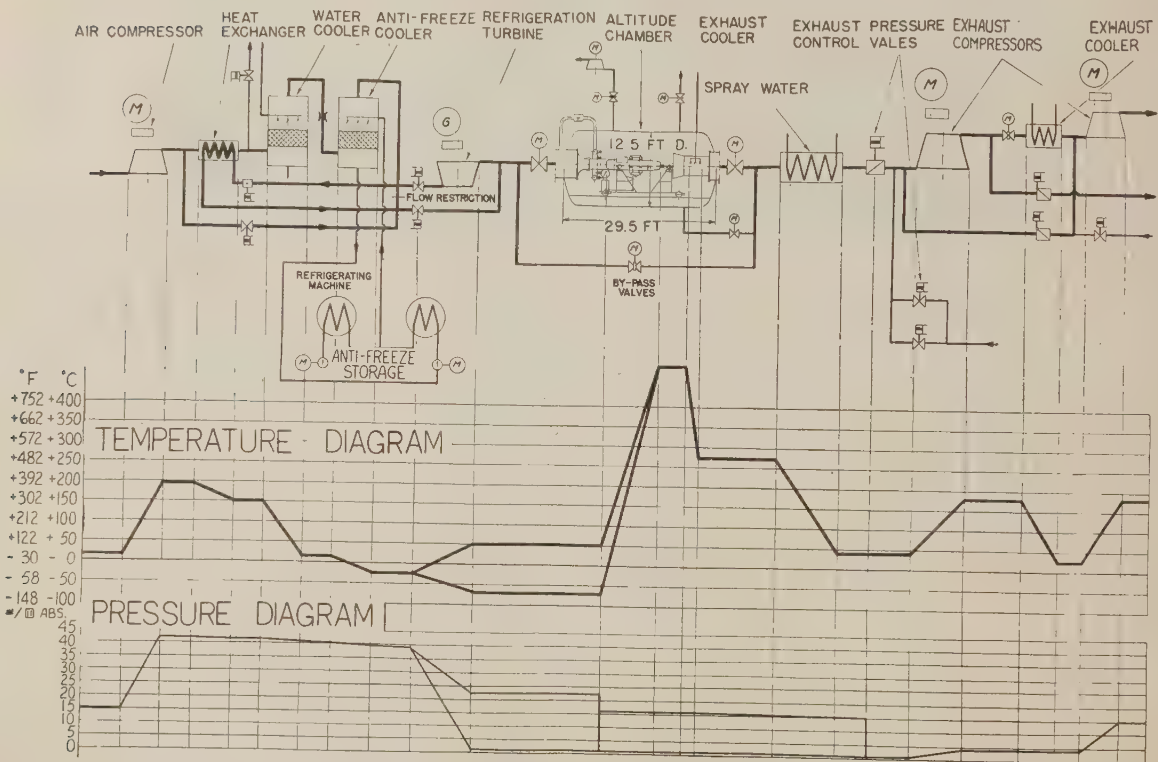

Fig. 5 Schematic Piping Arrangement, B.M.W. Plant

cause of the critical shortage of electric power in the Munich area, brought about by heavy Allied air bombardment, the plant could be operated only from late evening until early morning. As the construction of the other test plants was lagging behind, the German Air Ministry made the B.M.W. plant available to the other engine manufacturers on a priority basis.

From October, 1944, to April, 1945, several reciprocating engines and approximately 20 Junkers 004 and B.M.W. 003 jet-propulsion engines were completely altitude-tested.

B.M.W. Plant and Equipment

Test Chamber. The altitude test chamber, as shown in Fig. 3, consists essentially of a large cylinder in which the aircraft engine to be tested is mounted on a movable carriage. Refrigerated air under the desired conditions of temperature, altitude, pressure, and velocity is rammed into the front end of the engine. The exhaust system connected to the rear section of the cylinder removes the exhaust gases and cooling air. In the case of air- or liquid-cooled reciprocating engines and propeller-drive aircraft gas turbines, the power output is transmitted by a drive shaft through the inlet-air elbow to a combination water brake and electric dynamometer located on the left-hand side of the test chamber.

Fig. 4 shows the manner in which a jet-propulsion engine is mounted for testing. The air connection to the front end of the engine is through a free-sliding labyrinth joint transmitting no axial forces. With the unit mounted in a frame on rollers, the thrust output of the engine is transmitted by a system of links and bell cranks to a thrust-measuring scale located in the test chamber and read through a turnable periscope from the operating control room.

Fig. 5 is the schematic piping arrangement of the entire test plant with the principal pressures and temperatures at various points. The left-hand side of the figure shows the refrigerated-air-supply system, and the right-hand side shows the exhausting system. Starting on the left-hand side, normal atmospheric air is compressed to approximately 2.6 atm, cooled by direct-contact water sprays to 60 F, followed by cooling to — 10 F, by an antifreeze solution, and finally cooling to altitude temperature and pressure by expansion through an air-refrigeration turbine. In order to make available a wide latitude of air temperatures and pressures at the test-chamber entrance, the by-pass lines and heat exchanger shown in the line diagram were added. As will

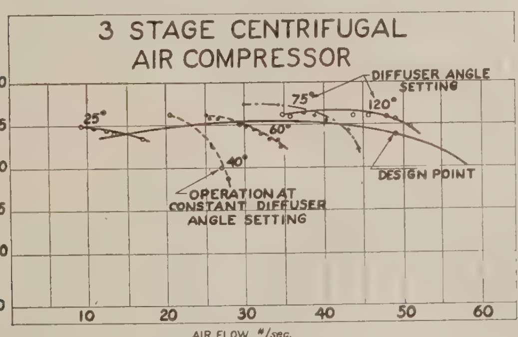

Fig. 6 Characteristics Air-Supply Compressor

be shown later, the purpose of the heat exchanger is to make possible rapid changes in temperature to the altitude test chamber without producing icing conditions.

Air Compressor. The air compressor is a three-stage radial-flow centrifugal unit with automatically controlled variable diffuser vanes and was built by the Brown Boveri Company. Fig. 6 shows a plot of pressure ratios for various air flows for this compressor. The design point and expected curve are given for various constant values of diffuser angle from 25 to 120 deg. These curves show that the compressor can operate stably between air flows of 10 lb per sec to approximately 55 lb per sec. An automatic control, actuated by air flow, varies the diffuser angle setting for operation along the designed curves. Fig. 7 shows the cable and linkage system used to control the angle of the diffuser vanes.

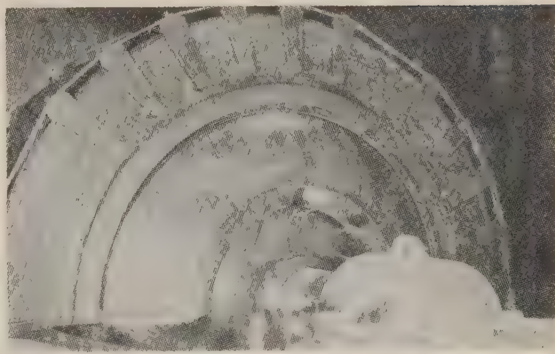

Fig. 7 Variable-Diffuser-Control Air-Supply Compressor

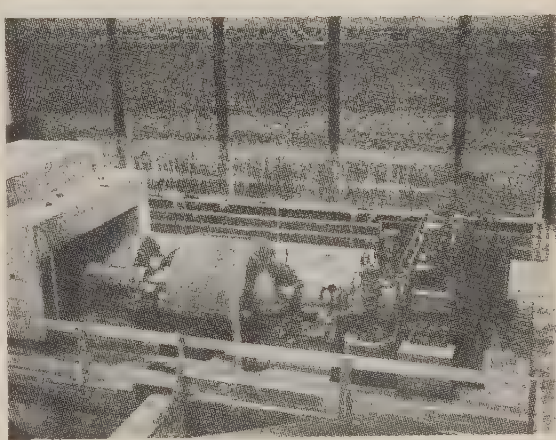

Fig. 8 Air-Supply Compressor and Refrigeration Turbine

Fig. 8 is a view of this compressor with the refrigeration turbine that drives it shown on the right-hand side. The difference in power between the turbine and compressor is made up by the 3300-kw motor on the left-hand side, which is connected to the compressor through a step-up gear.

The compressor driving motor was made large enough to carry full load on the compressor without the help of the refrigeration turbine. Operation of the compressor and refrigeration turbine is at a constant speed of 4370 rpm, while the driving motor runs at 990 rpm. The concrete blocks surrounding the motor were placed there during the last few months of the war as a protection against damage due to Allied bombing attacks. Similar concrete protection was placed around all other vulnerable apparatus as protection against bomb damage.

Air-Cooling System. The air leaving the air-supply compressor is cooled by a heat exchanger from 0 to 70 F, depending upon the method of operation. The next stage of cooling is accomplished

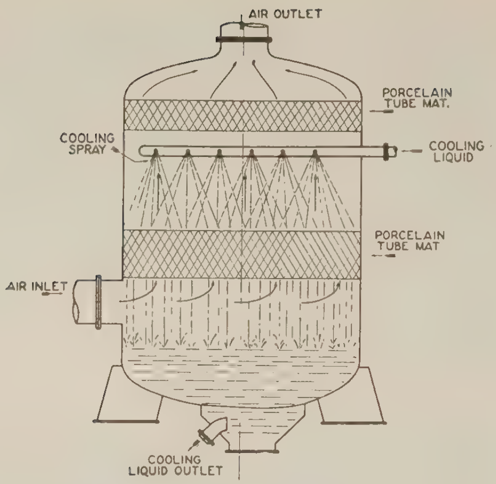

FIG. 9 AIR-COOLING CHAMBER

FIG. 10 SEVEN-STAGE FREON COMPRESSOR

in a water-cooling cylinder, a cross section of which is shown in Fig. 9. The air enters at the bottom left-hand side and passes up through a mat of small porcelain tubes each about $^5/_8$ in. long and $^3/_4$ in. OD $\times$ $^1/_2$ in. ID. These tubes are piled indiscriminately to form a mat with large surface area and many projecting corners. The total depth of this mat is approximately 36 in. Cooling water that is distributed evenly over this mat by means of sprays flows down through and around the tubes, finally collecting in the bottom of the cooler.

After the air has been cooled by contact with the water flowing over the surfaces of the short porcelain tubes, it goes up through another similar mat of tubes approximately 24 in. thick. In going through this second mat the excess moisture carried in suspension is effectively removed by the many sharp corners and narrow twisting paths up through which the air must go. Air leaving the top of this cooling cylinder is approximately at the compressor-discharge pressure of 2.6 atm and saturated with water vapor at a temperature of 60 F. At this temperature and pressure the air will hold in saturation approximately 50 grains of moisture per lb of air. In summer time this may be less moisture than in the incoming air but more than it is likely to hold in the winter months.

The next stage of cooling is similar in construction and operation except that an antifreeze solution is used instead of water. The outgoing air temperature from the second stage of cooling is about 10 F below zero. At this temperature and 2.6 atm, the saturation moisture content is reduced to 1.5 grains per lb of dry air. To accomplish this cooling the antifreeze solution enters the cooling chamber at —16 F and is removed at —6 F. Due to lowering the air temperature, approximately 1320 lb of moisture is condensed out of the air every hour. This is absorbed in the antifreeze solution from which it is removed by special multiple-effect evaporators. This is done on a continuous basis by sending a small fraction of the liquid leaving the air cooler through the concentrating process before going to the antifreeze cooling system. The antifreeze solution used was obtained under the trade name of Reinhartin and is believed to be of a calcium-chloride base with an inhibitor to reduce corrosion.

Cooling Antifreeze Solution. Cooling of the antifreeze solution is accomplished by an integrally designed refrigeration unit manufactured by Brown Boveri Company and sold under the trade name of "Frigibloc." This unit, consisting of a seven-stage centrifugal freon compressor, shown in Fig. 10, is built integrally with a condenser, evaporator, and all necessary accessories. Ther efrigerant gas used is freon F-11. It is extracted at the end of five stages and after passing through an intercooler is returned to the entrance of the sixth stage. Elaborate oil shaft seals are used to prevent air leakage in and freon leakage out. The compressor is driven by a 400-kw direct-connected motor running at 2950 rpm.

Fig. 11 shows the schematic arrangement of the component parts. The centrifugal compressor is shown at the center top and the evaporator, consisting of a tube heat exchanger with the antifreeze solution circulating inside the tubes, is shown directly below the compressor. The intercooler between the fifth and sixth stages is shown on the left-hand side, while the condenser with its liquid aftercooler is shown on the right-hand side. The schematic system for the shaft seal is shown in the upper left-hand corner, while the purging system is shown in the lower left-hand corner.

Refrigeration Turbine. The —10 F air coming from the antifreeze cooling chamber is expanded to altitude pressure in a refrigeration turbine designed and built by Brown Boveri Company. It is of conventional design having a single stage and eight equal admission-valve arcs. Good mixing between the air being by-passed around the turbine and that going through was obtained by building the by-pass directly into the turbine. The turbine buckets and wheel were made of a special nickel steel to withstand the low temperatures encountered. No insulation was applied to the outside of the turbine as it was felt it would interfere with accessibility and maintenance. This results in some heat absorption by the cold air but does not appear to be serious. The turbine is operated at a constant speed of 4370 rpm and is directly connected to the air-supply compressor as a means of loading. Fig. 12 gives the minimum air temperature that can be obtained at the test chamber after correcting for turbine and pipe-line heat loss. As can be seen from the curve, a minimum test-chamber air-inlet temperature of approximately 50 deg F below standard N.A.C.A. altitude temperatures can be maintained with flows of 20 lb per sec and more up to 40,000 ft altitude.

DANGER OF ICING

In the operation of a turbine-expansion cycle of this type for air-cooling there is always the danger of icing resulting from moisture condensation. As air is expanded to lower pressures, at constant temperature, it will hold greater amounts of moisture in saturation. As the temperature is reduced at constant pressure, the amount of moisture that can be held in saturation is rapidly reduced. Fig. 13 shows the plot of constant-saturation-moisture-content lines on a pressure-temperature field. From these curves it can be seen that with a turbine-inlet moisture content of 1.5 grains per lb of dry air, the moisture content can be kept below saturation along the N.A.C.A. altitude-temperature curve only up to 30,000 ft altitude and again above approximately 65,000 ft altitude. Between these two altitudes at N.A.C.A. temperatures and through broader limits at lower

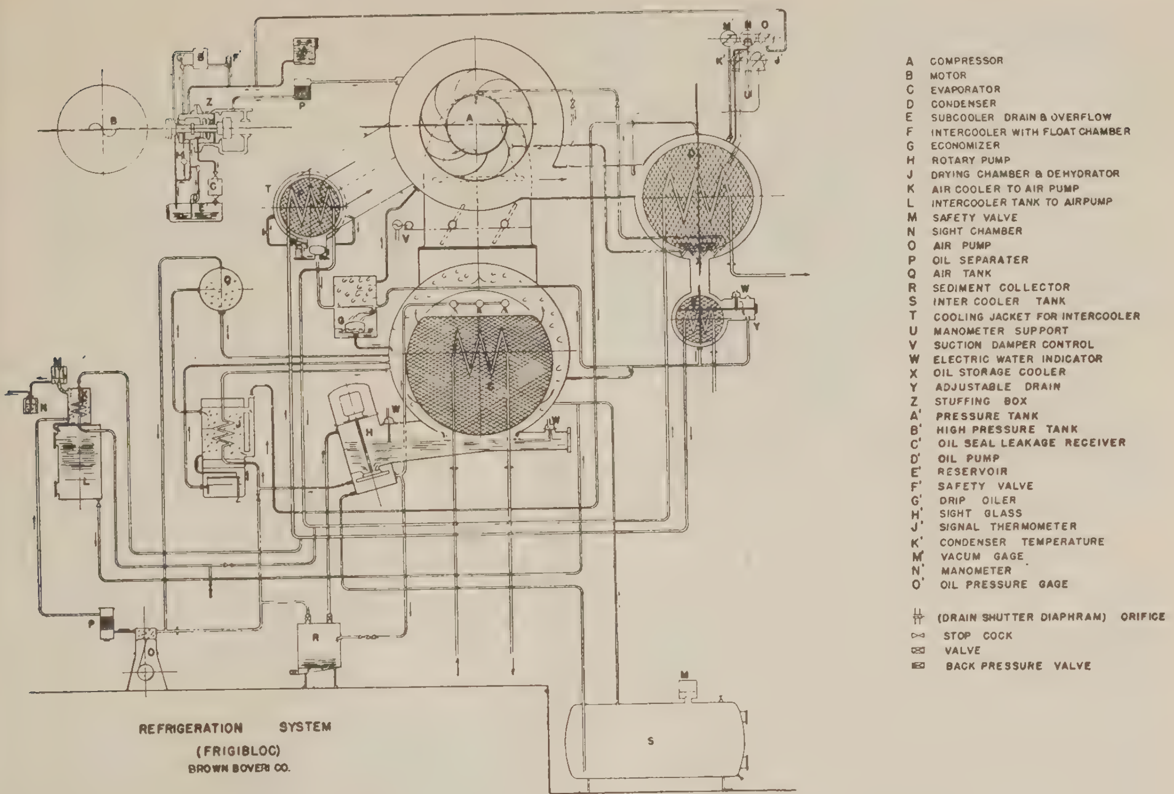

Fig. 11 Schematic Arrangement of Frigibloc Refrigeration Unit

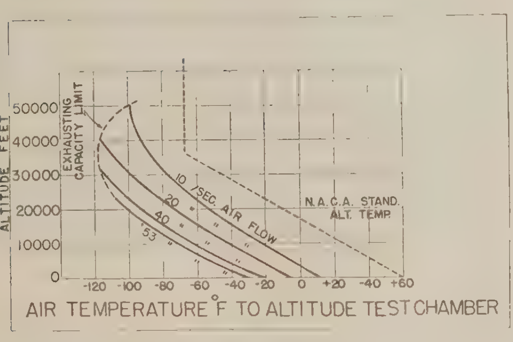

Fig. 12 Minimum Air Temperature to Altitude Chamber

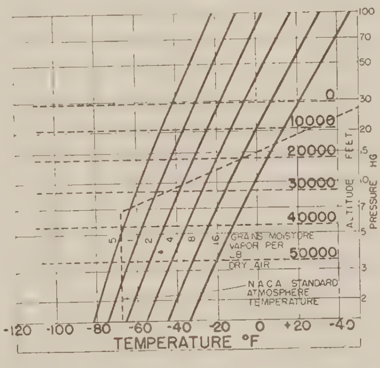

Fig. 13 Saturation Moisture Content at Various Temperatures and Altitudes

than N.A.C.A. temperatures, the air has more moisture than it can hold in saturation and the possibility of ice formation exists.

The designers of the B.M.W. test plant were aware that such icing might give trouble and considered in the initial design stages the installation of chemical driers for removing additional moisture from the air. Because of the size and expense of such driers, they decided to try the system without and find out by experience just how serious the icing problem would be. To date they have operated the plant several hundred hours, although not more than 6 hours continuously, in the range in which icing troubles should occur, as indicated in Fig. 13, and report that they have had no sign of trouble.

The author's personal experience with similar altitude-refrigeration equipment in the United States has shown that icing problems increase rapidly as the air-moisture content gets above 3 to 4 grains per lb and decreases rapidly below 3 grains. This evidence indicates that quite likely little icing trouble will be encountered if the moisture content can be held down to 1.5 grains.

If air temperatures higher than those obtained going through the refrigeration turbine are desired, air can be by-passed around the turbine and the air temperature to the test chamber raised to —10 F, with a moisture content of not over 1.5 grains. If still warmer air is desired, the piping is so arranged that the —10 F air from the antifreeze cooling chamber can be returned to

a heat exchanger between the supply air compressor discharge and the first water air cooler. The maximum temperature that can be obtained in this manner is 120 F. By this method air dried to 1.5 grains can be delivered to the test chamber at temperatures ranging from +120 F to —100 F. This is of particular value when obtaining compressor efficiencies by the temperature-rise method, especially with low inlet-air temperatures.

The heat exchanger between the compressor and the water cooler is used to rapidly heat the cold-air lines between the turbine discharge and the test chamber when shifting from a low temperature point to a higher temperature point. The by-pass valve from the air-compressor discharge to the turbine inlet is rarely used because of the moisture and icing problems encountered by moisture-laden warm air striking cold surfaces.

The inlet-air temperature to the altitude cylinder is controlled by changing the ratio of the air by-passed around the refrigeration turbine to the air that goes through the turbine, with the controls set so as to hold a constant total air flow. The inlet-ram conditions to the engine under test is obtained by varying the total air flow to the test chamber by increasing or decreasing both the air by-passed around the turbine and that going through the turbine. The controls are such as to maintain a constant ratio between the two.

Air-Exhaust System

The altitude cell is exhausted by means of two four-stage motor-driven centrifugal exhausters For low altitudes the exhausting units are operated in parallel while for higher altitudes the exhausters are operated in series. The change-over point is the break noted in curve A, Fig. 1. The larger of the two exhausters, shown in Fig. 14, operates at 3200 rpm and is driven by a 4500-kw 990-rpm motor through a step-up gear. The smaller unit operates at 4785 rpm and is driven by a 2700-kw 990-rpm motor through a step-up gear. These exhausters were built as multiple-stage centrifugal units rather than axial flow because of the belief they would have broader stability limits and would be less subject to performance deterioration due to accumulation of dirt.

Fig. 15 gives the pressure-ratio air-flow characteristics of the two exhausters. Fig. 16 is an outside view of the V1104 or smaller of the two exhausters. The two projecting flanges on the top half-casing hold membrane-type explosion vents that blow out in case of an explosion in the exhaust system. The motors driving these exhausters and the air-supply compressors are of the three-phase alternating-current asynchronous type with wound rotors. Starting is accomplished by inserting liquid resistors between the slip-ring connections. As the speed increases the liquid resistance is decreased until it becomes zero for normal operation. It was originally planned to control the exhaust pressure by speed control obtained by varying the rotor resistance. This did not prove to be practical and as a result the exhausters are operated at full speed, and chamber altitude pressure is maintained regardless of flow, within the limits of operation, by an automatic hydraulic system controlling the throttling valve just ahead of the first exhauster. When the system air flow gets below the stable range of the exhauster, air is bled in from atmosphere to the exhauster inlet.

The exhaust system is so valved that the change-over from series to parallel operation can be made by remote control while the units are operating and the test is going on. Smooth change-over is obtained on the three valves that it is necessary to change by making two self-acting, nonreturn check valves, and the third, a motor-operated gate valve. In changing over it is necessary to change only the motor-operated gate valve and the other two valves automatically follow in the proper sequence.

The hot exhaust gases leaving the test-chamber discharge are reduced to approximately 525 F by automatically operated water sprays. Additional cooling of the exhaust gases to approximately 100 F is accomplished by a surface gas cooler using water as a cooling medium. When the exhausters operate in parallel each discharges to the atmosphere through a brick stack. When the two operate in series an additional surface gas cooler is placed between the two exhausters to reduce the inlet temperature of the second one to approximately 100 F.

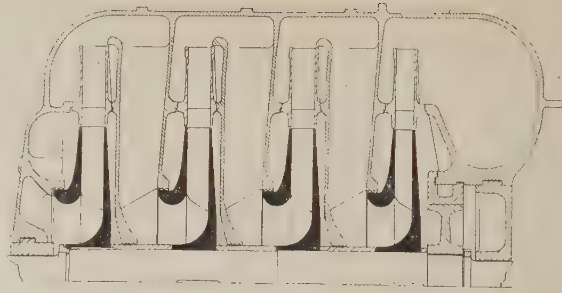

Fig. 14 Four-Stage Centrifugal Exhauster

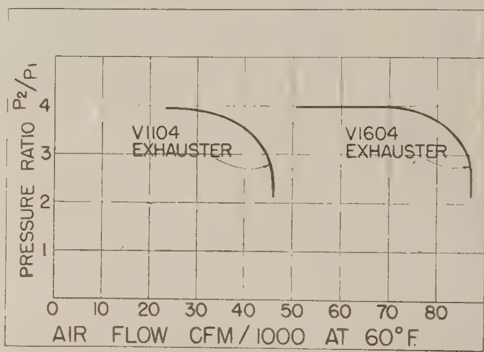

Fig. 15 Characteristics of Exhausters

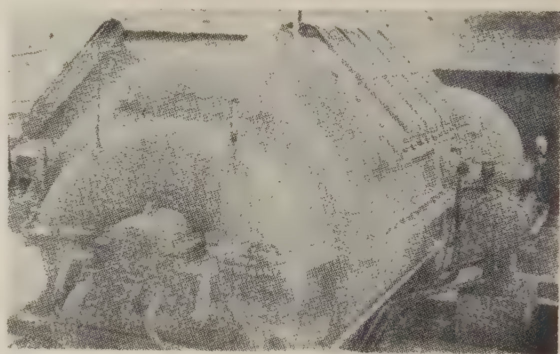

Fig. 16 Four-Stage Centrifugal Exhauster

Altitude Test Chamber

The altitude test chamber is a cylindrical body about 26 ft long and 12.5 ft diam. To withstand possible explosions, the chamber was tested with an internal pressure of 150 psi. The side walls are 0.8 in. thick and the end domes 1.38 in. thick. Since they had had no previous experience with such chambers, four large membrane explosion reliefs $34^1/_2$ in. diam were pro-

vided on the top side of the chamber. In order that the escaping gases from these vents would not damage the building, an explosion shaft was built above the chamber. To date no explosions have been experienced. This has been attributed to the large amount of excess air. Starting of the engine has always been on gasoline and some fires on starting have occurred. These have always been quickly extinguished by closing off the cooling-air supply to the chamber and injecting CO^2 gas. Experimental attempts were made to extinguish fires by quickly raising the altitude to 50,000 ft. This procedure failed to put out the gasoline fires. The author has seen similar oil fires in altitude chambers that were quickly put out by raising the altitude to 40,000 ft.

The entire exhaust-pipe system is protected against explosions and overpressure by a large number of explosion valves built especially for this particular job. These valves are spring-closing check valves of very light construction so that any sudden rise in pressure will quickly open them.

Access to the altitude chamber is obtained by removing the entire end dome and exhaust elbow. This operation is made entirely automatic, requiring only the pressing of several buttons, by the ingenious use of two power-operated bayonet-ring joints. Fig. 17, which is an end view of the altitude chamber, shows the large ring bayonet joint. After the two joints are opened a jib crane supporting the discharge elbow and end dome swings the assembly out of the way. In the center can be seen the exhaust nozzle of the jet engine under test together with the supporting test frame on which it is mounted.

All electrical, pressure, temperature, and fuel-supply lines to the test frame are made by quick connectors. This allows the

Fig. 17 Altitude Test Chamber

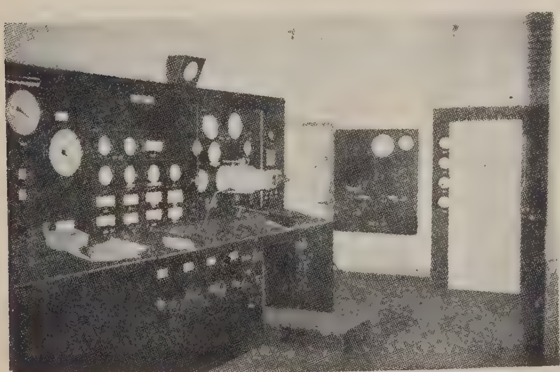

Fig. 18 Engine-Control Board, Altitude Test Chamber

Fig. 19 Refrigeration and Evacuation Control Room

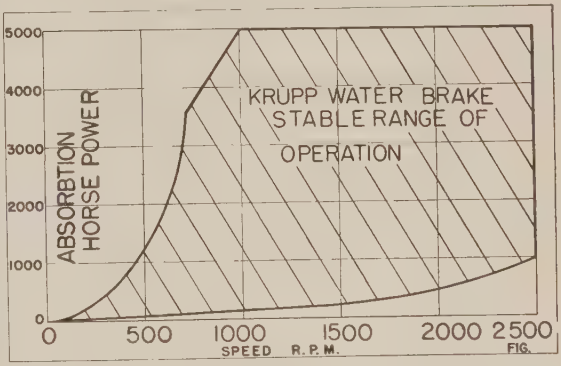

Fig. 20 Characteristics of Froude Water Brake

entire engine assembly to be quickly removed or installed. Such a complete change has been made in less than 2 hr. Fig. 18 is a view of the instrument board in the engine control room. In the center can be seen the turnable periscope used for viewing the engine in the protected altitude chamber. Important temperatures and pressures needed for the proper operation of the engine are given by direct-reading instruments directly in front of the operator. Special test temperatures are read on portable-type potentiometers on the left-hand side of the control board, while special pressures are read on mercury manometers on the right-hand side.

Fuel flow metering equipment is placed in a large tank that is vented to the altitude chamber. Two systems were originally set up to serve as a cross-check. One was a weight-tank system and the other was a volume measurement in which photoelectric cells operated an electric timer. Owing to trouble the photoelectric-cell volume method was discarded and the weigh-tank readings taken without a check method.

Speed measurements were made using a standard aircraft-type tachometer. On jet-propulsion engines, engine speed was held constant and of an exact value by making a matching stroboscopic disk appear stationary. The disk was driven by a small synchronous motor connected to a small synchronous generator on the engine. The flashing light source for the stroboscopic disk was from a special constant-frequency source and did not depend upon power-line frequency.

Fig. 19 is a view of the refrigeration and evacuation control room. From this point the operator controls the altitude pressure in the test chamber, the inlet-air temperature, and the inlet-air velocity or ram pressure. This information is trans-

mitted to the operator from the operator in the engine-control room over a public-address system.

Testing Propeller-Drive Engines

Propeller-drive engines were loaded onto a combination water brake and electric dynamometer. The water brake, whose characteristics are given in Fig. 20, was of the Froude design and had a capacity of 5000 hp over a speed range of 1000 to 2500 rpm. This brake was cradled with and directly connected to a Siemens Schuckert electric dynamometer that was capable of absorbing 1100 kw as a generator and developing 780 kw as a driving motor. The electric load of the dynamometer was loaded into a d-c — a-c motor-generator set that pumped back into the main a-c power lines. The total torque developed by the water brake and electric dynamometer was read on a single electric light beam weigh scale.

In spite of being designed and built under wartime conditions, no effort or expense was spared to make this plant the very best in all respects. That the Germans succeeded in this is testified to in the excellent manner in which the plant operated.

Peanut-Meal Plywood Glue

By R. S. BURNETT[1] and E. D. PARKER[1]

In this paper the results are given of work done at the Southern Regional Research Laboratory in which specifications were established for peanut meal for use in preparing plywood glue. A satisfactory formula has been developed. Joint tests show that the peanut-meal glue meets the requirements established for casein and casein-type glues. Comparisons of peanut-meal glues with other water-resistant glues are made.

THE FIRST large-scale application of an oilseed meal to the preparation of plywood glues began in 1927 with the use of soybean meal from Manchuria. This development took place in the Pacific Northwest to meet the needs of the Douglas-fir plywood industry for a cheap water-resistant glue. Although soybean-meal glue is more suitable for softwoods than for hardwoods, considerable amounts are used on hardwoods grown in the eastern and southern United States, especially in the manufacture of water-resistant box shooks (1).[2] While the preparation of plywood glue from other oilseed meals such as peanut (2, 3) cottonseed (4), castor-bean (5), and hempseed (6) meals has been investigated, to our knowledge only soybean-meal plywood glue has been used in significant amounts in this country. Of the latter 30,000 tons were consumed in 1942 (7). The present investigation is concerned with the preparation of a plywood glue from peanut meal.

The increase in peanut acreage in the South during the past ten years has made available a supply of protein-rich meal, but the preparation of plywood glue from this meal has been prevented by lack of the necessary technical information. The required information must include the development of a formula for preparing a suitable glue which must not only be capable of making a strong glued joint but must have flow properties that permit the glue mixture to fit effectively into the mechanical processes used to make plywood. Spreading characteristics, "working life," assembly time, etc., are all dependent on good flow properties. It is necessary to know also what influence the processing conditions employed to separate the seed into oil and meal have on the suitability of the meal for use in plywood glue.

As a result of work completed to date at the Southern Laboratory, specifications have been established for a peanut meal suitable for use in preparing plywood glue; a satisfactory glue formula has been developed; and information has been obtained with respect to the behavior of the glue under varying conditions of assembly time, pressure, etc. Comparisons of peanut-meal glues with other water-resistant glues have also been made.

Test Methods

The only way to evaluate a plywood glue is by gluing wood and measuring the strength of the joint obtained. Methods for evaluating plywood glues by the plywood and the block shear tests, developed by the U. S. Forest Products Laboratory, have served as a basis for Federal specifications for water-resistant glues (8, 9). The use and interpretation of data obtained by the application of these methods are discussed in several readily available publications (10, 11, 12). Such methods are subject to many variations which are difficult to control; they are not sufficiently accurate to measure small differences but they provide a means of making relative comparisons of one product with another. What actually is measured by such methods is the strength of the glued joint obtained rather than the strength of the glue. In other words, these methods of evaluation serve as a measure of the success of a gluing operation with a given glue.

The three principal tests required to evaluate a plywood glue are the dry and wet plywood shear test, the block shear test, and the measurement of the viscosity of the glue to determine its working life and setting properties.

The plywood shear test is made on specimens of birch plywood which have been prepared with the glue to be evaluated. The test pieces are subjected to tension in a standard plywood shear-testing machine until the joint fails. The load required to break the test pieces and the amount of wood failure is measured on dry joints and on those which have been soaked in water for 48 hours. This test is especially valuable for measuring the resistance of a glued joint to water. The specifications for casein and casein-type water-resistant glues (8, 9) require dry and wet plywood shear strengths of 340 and 140 pounds per square inch, respectively. There was no wood failure found in any of the wet joints tested except those made with one of the two lots of the casein glue mix containing blood and soybean meal reported in Table 9. In the plywood shear test the grain of the core of the 3-ply test pieces is at right angles to the faces.

In the block shear test the minimum requirement is 2800 lb per sq in. (8, 9). The grain of the two glued hard-maple blocks is parallel and a compression force is applied in the direction of the grain.

According to Federal specifications, "a glue shall be considered to have reached the end of its working life when it reaches a viscosity of 800 poises." For this test an orifice-type viscosimeter is recommended. However, for simplicity and speed of operation, the present investigators prefer to use a MacMichael rotating-cup viscosimeter employing a bob suspended from a torsion wire. When wire No. 26 is used with this instrument a reading of 290 deg is roughly equivalent to the maximum viscosity of 800 poises specified. Since the peanut-meal glues exhibit thixotropy, they must be stirred thoroughly to break down incipient gel formation before reliable viscosity readings can be made.

The gluing schedule followed in all of the plywood shear tests reported herein, unless otherwise indicated, was as follows:

Glue to water, proportion by weight	1 to 3
Temperature of wood and glue	75 to 77 F.
Glue spread	70–75 lb per 1000 sq ft
Closed assembly time	5 to 9 min
Pressure	150 lb per sq in. overnight at 75 to 77 F.
Age of glue mixture when used	1 hr
Moisture content of wood and of test pieces	6 to 7 per cent
Number of tests for each average value reported	15 (3 panels)

[1] Southern Regional Research Laboratory, New Orleans, Louisiana. One of the laboratories of the Bureau of Agricultural and Industrial Chemistry, Agricultural Research Administration, U. S. Department of Agriculture.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Wood Industries Division.

For the block shear test the gluing schedule was the same as for the plywood shear test with the following exceptions:

Pressure, 200 lb per sq in.
Number of tests for each average value reported, 5–15.

To control the moisture content in the wood and test samples used for the plywood and block shear tests an inexpensive, easily constructed, constant humidity room was built. This was described in a previous publication (13). Test samples held in this room for 72 hours at a relative humidity of 32 per cent reach an equilibrium moisture content of 6 to 7 per cent.

Formula Development

Finely ground oilseed meals are required for the preparation of plywood glues. Because of the 6 to 10 per cent of oil which remains in hydraulic-press meals a flour cannot be prepared by ordinary sifting methods. It is necessary therefore to employ an air separator to obtain a flour of the desired fineness. The peanut flours used in this investigation were ground and sized so that 80 to 90 per cent would pass through a 200-mesh sieve; the sieve tests being run after the oil was removed by means of solvent. The oil remaining in the meal serves to prevent the glue from foaming during the mixing and spreading operations.

A readily available peanut-meal flour, Table 4, meal No. 1, which had been produced under controlled conditions for use as a food was chosen for the preliminary work on the development of a plywood-glue formula. It had a high protein content and was processed to separate the oil and meal at relatively low temperatures. Subsequent work showed this to be a good choice and this flour, or one prepared under similar conditions, was used in the work reported here, unless otherwise indicated.

Casein and soybean-meal glues are usually made up with water and various combinations and amounts of sodium hydroxide, lime, and sodium silicate. As would be expected, preliminary experiments indicated that peanut meal so combined behaved in a manner generally similar to that of soybean meal and casein.

A joint which has a high dry strength can be prepared with peanut meal by the use of sodium hydroxide and water alone. The addition of lime to such a mixture provides an irreversible gel which is necessary for wet strength in the glue bond. The addition of sodium silicate in amounts up to 15 parts per 100 parts of meal greatly improves the spreading qualities of the glue. Best results are therefore obtained when all three alkalies are present in the glue mixture.

Taking into account viscosity characteristics, working life, water requirements, and the wet and dry strengths of the plywood joint obtained, the combination of alkalies which has given the best results with peanut meal is: hydrated lime 15 parts, sodium hydroxide 4 parts, and sodium silicate 15 parts per 100 parts of peanut meal. Some of the data obtained in arriving at this formula are given in Table 1. In the series of tests recorded in the table the amount of sodium silicate and tetrasulphide, discussed later, was held constant. The addition of hydrated lime up to 15 parts per 100 parts of meal was found to increase the wet shear value. The use of less than 10 parts of hydrated lime gave glue mixtures with unstable viscosities, whereas with the addition of 10 and 15 parts of hydrated lime the viscosities were stable at all concentrations of sodium hydroxide shown. Increasing the amount of sodium hydroxide from 2 to 6 parts increased the dry strength somewhat and lowered the wet strength. A good compromise therefore is the formula containing 15 parts of hydrated lime and 4 parts of sodium hydroxide per 100 parts of meal. However, any one of the last six formulas in Table 1 provides good joints and has good viscosity characteristics.

A plywood glue which will produce a joint having a wet strength up to about 80 lb per sq in. can be prepared by the addition of lime, sodium hydroxide, and sodium silicate to the peanut meal. However, to further increase the water resistance of the glue it is necessary to use, in addition to lime, another insolubilizing agent, and the most satisfactory one which has been tried is carbon disulphide (14, 15). Carbon disulphide is recommended for use in soybean glue. It is sold mixed with equal parts of carbon tetrachloride to eliminate hazard of fire and the mixture is called "tetrasulphide." The influence on the plywood shear test of carbon disulphide, in the form of tetrasulphide, is shown in Table 2.

TABLE 1 EFFECT OF SODIUM HYDROXIDE AND LIME ON THE VISCOSITY AND PLYWOOD SHEAR STRENGTH OF PEANUT-MEAL GLUE WITH A 1:3 RATIO OF MEAL TO WATER[a]

Hydrated lime, per 100 g of meal, g	Sodium hydroxide per 100 g of meal, g	Viscosity in MacMichael deg wire No. 26				Plywood shear tests	
		0.5 hr	1 hr	2 hr	4 hr	Dry strength psi; Wood failure %	Wet strength psi
3	2	44	52	48	65	308–10	0
3	4	85	220	300+	...	....	...
3	6	99	172	...	...	....	..
3[b]	6	48	96	234	300+	395–16	80
5	2	37	51	76	191	383–18	128
5	4	70	115	208	...	....	...
5	6	103	187	300+	...	....	...
5[b]	6	48	66	105	154	404–10	122
10	2	50	59	65	88	398–12	138
10	4	48	58	68	74	400–13	134
10	6	78	84	96	88	417–38	136
15	2	44	45	59	63	396–12	151
15	4	56	67	74	76	424–26	139
15	6	81	80	79	63	400–26	121

[a] The amounts of sodium silicate and tetrasulphide were held constant at 15 g and 2 ml, respectively, per 100 grams of meal.
[b] Ratio of meal to water increased to 1:3.25.

TABLE 2 INFLUENCE OF THE CARBON DISULPHIDE IN TETRASULPHIDE[a] ON THE VISCOSITY AND WATER RESISTANCE OF PEANUT-MEAL GLUE

Tetrasulphide per 100 g of meal, ml	Viscosity in MacMichael deg, wire No. 26				Plywood shear tests	
	1/2 hr	1 hr	2 hr	4 hr	Dry strength lb per sq in.; Wood failure, %	Wet strength lb per sq in.
0	53	45	45	37	432–18	48
1	52	54	57	56	433–13	118
2	51	69	71	73	444–15	140
3	61	97	94	97	412–19	150

[a] Equal parts of carbon disulphide and carbon tetrachloride.

From the standpoint of cost, viscosity stability, and water-resistance the use of 3 parts by weight of tetrasulphide per 100 parts of meal (2 ml per 100 gm) is satisfactory.

The complete formula for peanut-meal glue as compared with two commercial soybean-glue formulas is given in Table 3. This peanut-meal glue formula was used in all the work reported in this article, unless otherwise indicated.

The low water requirement of the peanut-meal glue is important because it permits the addition of less water to the veneer in the gluing operation, with the result that less water need be removed from the plywood in the drying tunnels than is necessary with glues which contain larger amounts of water.

TABLE 3 COMPARISON OF THE FORMULA DEVELOPED FOR PEANUT-MEAL GLUE WITH FORMULAS FOR OTHER WATER-RESISTANT PLYWOOD GLUES

(Quantities shown are parts per 100 parts of meal)

	SRRL peanut-meal glue	Commercial soybean-meal glue	Commercial soybean glue fortified with casein and blood albumin
Sodium hydroxide	4	7	5
Hydrated lime	15	3	12
Sodium silicate:			
Philadelphia quartz N brand or equivalent	15	15	25
Tetrasulphide	3[a]	3	2
Water	300	390	355

[a] Cc per 100 g.

TABLE 4 INFLUENCE OF PROCESSING CONDITIONS USED TO PREPARE THE MEAL ON ITS SUITABILITY FOR USE AS A PLYWOOD GLUE

Meal no.	Conditions used to prepare meal: Cooking time min.	Conditions used to prepare meal: Maximum temp F	Analysis of meal: Protein (N × 6.25) %	Analysis of meal: % of total meal N soluble in 1M NaCl	Analysis of meal: Lipids %	Ratio of meal to water in glue mixture	Viscosity, MacMichael deg Wire No. 26: ½ hr	1 hr	2 hr	4 hr	Plywood shear tests: Dry strength, psi; Wood failure %	Plywood shear tests: Wet strength psi	Block shear tests strength, psi; Wood failure %	Group
						Commercial hydraulic-press meals								
1	76	215	58.4	81.0	10.5	1:3.0	44	65	90	113	417–15	147	3234–16	I
2	108	240	43.8	44.3	7.0	1:3.46	42	47	51	49	298–12	27	1915–29	II
						1:3.25	64	68	71	66	322–12	55	.....	
3	90	235 to 238	50.2	52.0	7.6	1:3.46	37	38	38	39	366–12	113	2550–17	II
						1:3.25	56	59	64	60	337–16	123	.....	
4	...	....	53.3	38.6	7.2	1:3.46	54	58	58	59	334–10	119	2741–42	II
5	...	....	52.2	60.8	6.7	1:3.46	38	42	52	55	360–12	122	2780–32	II
						1:3.25	51	49	48	43	396–19	51	.....	
						1:3.00	98	103	94	75	365–83	83	.....	
6	80	210	51.0	70.9	8.2	1:3.25	40	50	62	72	342–17	151	.....	I
						1:3.00	53	71	81	...	358–17	194	2780–22	
7	...	....	44.8	63.2	8.8	1:3.46	72	84	97	104	298–13	153	2242–20	II
						1:3.46	63	77	80	...	335–17	134	.....	
8	...	....	52.5	45.5	7.0	1:3.46	58	60	70	...	397–17	136	.....	II
						1:3.46	60	65	75	78	361–15	139	2295–16	
						Commercial expeller meals								
	..	....	57.0	55.3	7.1	1:3.67	41	44	54	56	358–13	42	.....	II
						1:3.34	97	99	97	91	397–11	79	2823–15	
10	...	....	48.2	17.5	10.9	1:3.46	118	126	143	142	319–11	75	.....	II
						Pilot-plant hydraulic-press meals made from 2-year-old nuts[a]								
11	60	219	53.6	75.8	8.4	1:3.25	50	66	76	93	400–14	155	3378–11	I
12	60	240	54.6	69.2	8.5	1:3.25	55	71	80	91	374–23	152	3529–16	I
13	80	262	54.2	37.7	12.8	1:3.25	76	85	106	119	345–14	101	2909–3	II
						Pilot-plant hydraulic-press meals made from fresh nuts[a]								
14	60	220	52.5	72.3	7.0	1:3.25	38	42	49	62	325–14	110	.....	I
						1:3.00	65	81	97	111	398–25	143	2956–19	
						1:3.00	63	80	107	113	374–11	140	.....	
15	60	240	53.7	69.4	8.0	1:3.25	44	48	55	62	340–15	141	.....	I
						1:3.00	54	64	76	86	368–16	147	2829–6	
16	80	262	52.8	40.9	10.8	1:3.25	59	67	72	80	396–19	118	.....	II
						1:3.25	62	65	74	81	389–12	134	3045–22	
17	60	225	51.9	76.0	8.3	1:3.00	53	59	75	76	400–29	155	3308–39	I
						Solvent-extracted meal								
18	Not cooked		57.3	89.5	10.0[b]	1:2.9	45	54	74	85	397–30	148	.....	I

[a] Two lb water added to 50 lb peanut flakes before cooking.
[b] Ten parts peanut oil added to 90 parts solvent-extracted meal in order to prevent the glue from foaming in the spreader.

Choice of Meal

As pointed out, the processing conditions employed to prepare peanut meal have considerable influence on the suitability of the meal for use as a plywood glue. This is shown by the data presented in Table 4.

The solubility of the protein in peanut meal varies with variations in the processing conditions employed to cook and press the rolled meats. This variation referred to as degree of denaturization, can be estimated from the percentage of the total meal nitrogen which is soluble in 1 molar sodium-chloride solution (16). Most commercial hydraulic-press peanut meals appear by this test to be appreciably denatured (low nitrogen solubility) as indicated in Table 4 and in an earlier publication (17). If the meals in Table 4 are divided into those which have a nitrogen solubility in 1M sodium chloride of 69 per cent or better (group I) and those which have a lower solubility (group II) it can be seen that the meals in group I make more satisfactory glues than those in group II. In most cases both dry and wet plywood strengths as well as the block shear strength is low when glues prepared from meals in the low-nitrogen-solubility group are used, the low wet strength being especially marked. The lower test values in group II are probably due, in part, to the necessity for using more water in preparing the glue mixture when the protein in the meal has been appreciably denatured. It is also possible that denaturization inactivates some of the groups in the protein molecule which are capable of reacting with lime and carbon disulphide to increase water resistance.

An examination of the conditions of time and temperature used to prepare commercial meals 1 and 6 (group I) shows that maximum cooking temperatures of 210 to 215 F for 76 to 80 minutes provide conditions for satisfactory oil recovery and at the same time produce a meal which is suitable for use as a plywood glue.

Nevertheless, application of these conditions may not produce a satisfactory meal in all oil mills, and specific directions for preparing meal cannot be given. Each mill will need to determine the best procedure for preparing a meal in which 70 to 80 per cent of the total nitrogen is soluble in 1M sodium chloride. As a further check a plywood-glue test should be made with meals which meet the solubility specification. The complete specifications for meal which has uniformly given good results when used as a plywood glue follow:

Protein (N × 6.25)	50 per cent minimum
Percentage of total meal N soluble in 1M sodium chloride	70 per cent minimum
Sieve test	80 per cent or more through 200 mesh
Oil	6 per cent minimum

The meals listed in Table 4, Nos. 11–17, were prepared in a pilot-plant special-model hydraulic press. Our experience and that of others have shown that meals prepared in a small mill may differ considerably from meals prepared under the same cooking and pressing conditions in a mill of greater capacity. Although a satisfactory meal can be prepared in the pilot plant by heating rolled meats to temperatures as high as 240 F for 60 min (meals Nos. 12 and 15) it would be unsafe to conclude that these conditions would yield the same results in a larger mill.

It is probable that the amount of water present or added to the flaked peanuts and the rate at which the water is driven from the flakes during the cooking operation accounts for the difference in the degree of denaturization of the meal obtained in various mills even though the cooking time and temperature remain the same. Fontaine, Samuels, and Irving (16) have shown the influence of water vapor on the degree of denaturization of the protein in cooked peanuts.

Peanut shells are incompletely removed prior to processing or

are later added to peanut meal which is intended for use as feed, the minimum protein (N × 6.25) requirement being 48 per cent. The presence of shells in meal intended for use as plywood glue, however, acts as a filler and should be avoided. The soybean-plywood-glue manufacturer pays a premium in order to obtain a meal which has a high protein content. The peanut-meal manufacturer should have no difficulty in preparing a meal containing 50 per cent or more protein (N × 6.25) for use in preparing plywood glue.

The first series of pilot-plant meals was prepared from peanuts which were two years old. It was advisable therefore to repeat the tests when a fresh stock of peanuts became available. Results indicate that the age of the seed used has no influence on the gluing characteristics of meal.

Solvent-extracted meal (meal No. 18) can also be used for preparing plywood glue. It is necessary, however, to add oil to the glue made from this type of meal in order to prevent excessive foaming in the spreader.

Influence of Various Factors on the Gluing Characteristics of Peanut-Meal Glue

A satisfactory plywood glue must produce a good joint with the equipment and under the conditions ordinarily encountered in the plywood factory.

The temperature of the glue room and of the water used to prepare plywood glue varies considerably. It was therefore of interest to determine the influence of temperature on the viscosity of peanut-meal glue. The results shown in Table 5 indicate that within the range of 65 to 95 F temperature has little influence on the flow properties of peanut-meal glue.

TABLE 5 EFFECT OF TEMPERATURE ON THE VISCOSITY OF PEANUT-MEAL GLUE

Temperature at which glue was prepared and held, F	Viscosity, MacMichael degrees Wire No. 26			
	1/2 hr	1 hr	2 hr	4 hr
65	50	62	66	76
77	53	59	75	76
85	62	72	86	77
95	59	72	82	71

TABLE 6 EFFECT OF AGE OF THE PEANUT-MEAL GLUE MIXTURE ON PLYWOOD SHEER STRENGTH

Age of glue, hr	Plywood shear tests: Dry strength, psi Wood failure, %	Wet strength, psi
1	405–25	132
2	407–14	128
3	392–10	115
4	396–14	104

The influence of the age of the glue mixture on the results obtained with a peanut-meal glue is shown in Table 6. Test panels were prepared with the glue at the end of 1, 2, 3, and 4 hours. The wet strength obtained was somewhat low to begin with and decreased to some extent with age.

In the manufacture of plywood the glued veneer is stacked until a press load is accumulated, an operation usually requiring 10 to 20 minutes. If a glue is too fluid, an excessive amount may be absorbed by the wood during this period or an excessive amount may be squeezed out when the glued veneer is placed under pressure. If the glue is too viscous, absorption of water by the wood may lead to the production of a dry joint, with the result that a thick film of glue and a weak bond are obtained. The influence of the closed assembly time on the strength of the bond obtained with a typical peanut glue is shown in Table 7. The results indicate that an assembly time up to 15 minutes gives a strong bond. This is an acceptable assembly period.

The yellow birch used for plywood tests varies considerably in color, from white through yellow to red, depending on whether the veneer is cut from sapwood or from heartwood. Since the color of the wood to be used in tests is not specified in the literature, it was of interest to compare woods selected for color, and at the same time to determine the influence of pressure on joint strength. The results given in Table 8 indicate that joint strength is not appreciably influenced by variation in veneer color or by variation of pressure within the range ordinarily applied.

TABLE 7 INFLUENCE OF CLOSED ASSEMBLY TIME ON PLYWOOD SHEAR TESTS

Closed assembly time min	Plywood shear tests: Dry strength psi; Wood failure, %	Wet strength psi
5	414–24	159
10	397–38	147
15	356–21	142
20	296–22	95

TABLE 8 EFFECT OF PRESSURE ON JOINTS MADE WITH A SINGLE LOT OF PEANUT-MEAL GLUE APPLIED TO RED, YELLOW, AND WHITE "YELLOW BIRCH" VENEER

(In terms of plywood shear strength, psi; Wood failure, %)

Pressure	100 psi		150 psi		200 psi	
	Shear strength psi; Wood failure %					
	Dry psi–%	Wet psi	Dry psi–%	Wet psi	Dry psi %	Wet psi
Red	415–35	131	454–34	139	447–46	142
Yellow	412–22	148	405–44	150	414–32	128
White	402–27	138	391–34	153	398–30	151

Comparison of Peanut-Meal Glue With Other Water-Resistant Glues

A comparison was made of the results obtained with peanut-meal glue and those obtained with other water-resistant glues as shown in Table 9.

It was also of interest to prepare test panels with tupelo and red-gum woods in addition to the usual birch panels since these two southern hardwoods and Douglas fir are used in the greatest amounts for the manufacture of veneers in this country. The relative amounts (thousands of feet, log scale) of these woods consumed in 1937 for the manufacture of veneers was: Douglas fir 306,299, red gum 213,654 and tupelo 170,438 (18).

An examination of Table 9 reveals that peanut-meal glue is equal to soybean-meal glue when tested according to the official birch-plywood test and also when tupelo and red gum are used. Although the results reported in this investigation appear to indicate that the peanut-meal glues tested are slightly superior in most cases to the soybean-meal glues tested, the number of soybean-meal glues examined is too small to justify concluding that there is any significant difference in these two oilseed-meal glues. It is also probable that optimum conditions were not always employed in gluing the three woods with the two glues. The casein glue examined as well as the mixture of casein, blood, and soybean meal are in most instances superior to the oilseed-meal glues.

The satisfactory results obtained by gluing tupelo and red-gum veneers with peanut-meal glue and the proximity of these southern hardwoods to the raw material for preparing peanut-meal glue presents an opportunity of mutual advantage to southern agriculture and the southern plywood industry.

Influence of Added Blood Albumin and of Hot-Pressing on Peanut-Meal Glue

Blood albumin is sometimes added to soybean-meal glues to increase their water resistance. Glues with or without added blood may be heated in hot presses after the glue joint has been prepared in the usual manner by holding the glued product under pressure for several hours. Therefore preliminary tests which show the effect of hot-pressing at two temperatures and the effect of adding blood to peanut meal, as shown in Table 10, may be of interest.

Hot-pressing of plywood prepared with peanut-meal glue at 180 F has little effect, but raising the temperature to 240 F in-

TABLE 9 COMPARISON OF PEANUT-MEAL GLUE WITH OTHER WATER-RESISTANT GLUES WHEN APPLIED TO VARIOUS WOODS

Description of meal or glue used	Lot no.	Viscosity[a] — Change with time in hours: 1/2	1	2	4	Birch — Dry strength, psi; Wood failure %	Birch — Wet strength, psi; Wood failure %	Tupelo — Dry strength, psi; Wood failure %	Tupelo — Wet strength, psi; Wood failure %	Red gum — Dry strength, psi; Wood failure %	Red gum — Wet strength, psi; Wood failure %	Block shear tests strength, psi; Wood failure %
Peanut-meal glue	1	53	80	...	77	364-12	157	327-23	91	382-28	150	.....
	2	..	..	...	...	367-14	147	374-41	138	373-50	110	2857-13
Commercial soybean-meal prepared for use as plywood glue.[b]	1	..	..	...	...	393-26	151	401-7	0	328-9	105	.....
	2	71	72	86	102	396-15	73	268-44	112	352-16	33	2806-6
	3	58	63	71	88	396-30	142	274-43	143	370-10	138	.....
Casein-glue mix ready for use upon addition of water.[b]	1	..	..	...	...	495-30	195	408-25	138	407-77	151	.....
	2	30	42	67	136	523-43	214	357-70	185	403-51	171	3185-32
Casein-glue mix containing blood and soybean meal.[b]	1	48	67	76	...	489-36	259 15	340-21	166 10	356 12	184 13	.....
	2	15	47	148	Gel	409-22	167	251-28	138	341-20	112	3282-40

[a] MacMichael degrees; wire no. 26.
[b] Used as recommended by manufacturer.

TABLE 10 EFFECT ON PLYWOOD SHEAR STRENGTH OF ADDING DRIED BLOOD ALBUMIN TO PEANUT MEAL AND OF HOT-PRESSING

	Peanut-meal glue[b]	Peanut meal, 90 parts;[c] blood albumin, 10 parts	Peanut meal, 80 parts;[c] blood albumin, 20 parts
Ratio of meal to water in glue mix	1:3	1:3.25	1:3.25
Carbon tetrasulphide, ml per 100 g meal	2	1.5	None
Viscosity at 1/2, 1, 2 and 4 hours (MacMichael degrees, wire no. 26)	59, 68, 77, 77	42, 103, 155,—	180, 180, 188, 170
Plywood shear tests:			
Dry strength psi; wood failure, per cent	396-9	411-10	417-23
Wet strength psi	135	170	166
Shear tests on plywood hot-pressed at 180 F:[a]			
Dry strength psi; wood failure, per cent	395-25	430-10	454-16
Wet strength psi	156	202	224
Shear tests on plywood hot-pressed at 240 F:[a]			
Dry strength psi; wood failure, per cent	464-39	504-25	553-41
Wet strength psi	177	232	260

[a] Plywood dried overnight at 150 lb per sq in. pressure and subsequently hot-pressed at 150 lb per sq in. for 10 min at the temperature shown.
[b] Formula given in Table 3.
[c] Formula given in Table 3 with exceptions noted.

creases both the dry and wet plywood shear strengths. The addition of 10 and 20 per cent of blood albumin to the peanut meal increases the shear strength both before and after hot-pressing for 10 min under 150 lb per sq in. pressure at 180 and 240 F. In each case the panels were held at 150 lb per sq in. pressure overnight before they were hot-pressed.

The presence of blood albumin increased the viscosity of the glue considerably thus making it necessary to increase the amount of water; at the same time it decreased or eliminated the need of tetrasulphide.

Plant-Scale Formula and Details for Preparing the Glue Mixture

The formula recommended for preparing plant-scale batches of peanut-meal glue is as follows: The ingredients are added in the order listed:

Water, 200 lb
Peanut-meal flour, 100 lb
Add flour to water and stir to a smooth paste. Then add
Water, 55 lb
Hydrated lime, 15 lb mixed with 35 lb of water
Caustic soda, 4 lb dissolved in 11.5 lb of water
Tetrasulphide (equal parts carbon disulphide and carbon tetrachloride) (1 qt) 3 lb
Stir for 2 minutes and add silicate of soda (Philadelphia "N" brand or equal), 15 lb. Stir for a few minutes and use.

A spread of about 150 pounds of wet glue per thousand sq ft, 3-ply basis, and a pressure of 150 lb per sq in. is satisfactory for gluing plywood.

The foregoing formula is for application to dry hardwood veneers. It may be necessary to alter the water content of the glue when gluing softwood veneers or wet veneers.

Peanut-meal glue was tried on a plant scale in a commercial plywood factory. The glue performed well throughout the process and gave as good or better joints with poplar and black-gum veneers as did soybean-meal glue which was used under the same conditions on the same lots of veneer.

Summary

Specifications have been developed for peanut meal which is suitable for use in preparing plywood glue.

A glue mixture has been developed which has good flow properties and which is accommodated to ordinary gluing schedules for this type of glue. Joint tests show that the peanut-meal glue meets the requirements established for casein and casein-type glues.

Acknowledgment

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Locomotive Fuel From the Coal-Man's Viewpoint

By C. F. Hardy[1]

The sizes of coal for locomotives and the preparation of coal, screening, and segregation are discussed in this paper. It is suggested that from an engineering standpoint coal specifications for locomotives should be standardized because of the changes in equipment and construction, and that extensive tests should be made.

When the credit for winning World War II is apportioned a large share will be given to the coal-burning steam locomotive. With a tremendous task to do, using to a large extent different sizes and types of coal from those that were formerly furnished, the railroads were able to keep the nation's freight and passengers moving. Probably no other major coal user had so much dislocation of normal coal supply as did the railroads. Of necessity a great part of the coal which could be used for by-product, special purposes, or domestic use, was taken out of railroad fuel and replaced in the main by smaller sizes or lower quality. It is doubtful if any other type of fuel-burning equipment than the steam locomotive could have been operated under such adverse circumstances.

Now that the war is over, the possibility of improving sizes, type, and preparation of the coal for locomotive use is again pertinent. There have been tremendous advances in the practices of coal preparation and coal cleaning in recent years. From the simple bar screens of a few decades ago, coal-screening plants can now produce any practical size of coal. Along with this trend of sizing coal has come the study of its impurities and the best methods of removing these. Some coal-producing districts have installed more cleaning plants than others. This trend has been reflected in the type of coal furnished for locomotive use from these producing districts. These questions naturally arise: Why cannot all railroad coal be standardized as to size and preparation? Is it feasible for either the railroad or the coal industry to standardize locomotive coal? Will it help the performance of present or future locomotives?

The economics of the problem are bound up with the well-known interdependence of the railroads and the coal industry. A few details will suffice to show this interdependence. The railroads are the coal industry's best and most consistent customers and have bought during the last decade approximately 21 per cent of the total annual coal production. The railroads, by purchasing sizes which are temporarily not in demand, have permitted many mines to keep in regular production, whereas otherwise they would have to operate sporadically due to the lack of markets for one or more sizes. This effect is also shown by the fact that the monthly stocks of coal which the railroads carry have varied from month to month as much as 40 per cent, plus the fact that they buy widely varying sizes of coal. The other aspect of the interdependence is that the coal industry is responsible for supplying the railroads with 20 per cent of their total gross freight revenue in normal years and more than 80 per cent of the railroads' total annual fuel requirements.

[1] Chief Engineer, Appalachian Coals, Inc., Transportation Building, Cincinnati, Ohio. Mem. A.S.M.E.

Contributed by the Fuels and Railroad Divisions and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

Sizes of Coal Now Used and Recommended for Locomotive Firing

A complete analysis of sizes of coal used by locomotives entitled "1937 Railway Locomotive Fuel" (1)[2] was prepared by the former Bituminous Coal Division. Two hundred and fifty-two different size designations were shown in this report. A less elaborate compilation was made for 1944 and apparently it agrees fairly well with the 1937 figures. In 1937, 445 railroads used 89,935,580 tons of bituminous coal for locomotive fuel. This tonnage represented 82.6 per cent of all locomotive fuel, including fuel oil, anthracite, and wood. The coal came from all but one of the 23 coal-producing districts set up by the Bituminous Coal Act of 1937. (See map.)

Table 1 shows the amount of on-line, off-line, lake, tidewater, truck, and river, total railroad fuel, and the percentage of the districts' production going to railroad fuel. Of the major producing districts, District No. 7, Southern low-volatile, showed the smallest percentage (1.8 per cent) used for locomotive fuel.

Table 2 gives a breakdown by size groups of the total of all districts and of the six largest railroad-fuel-producing districts. From this table it is seen that run of mine and resultant run of mine of more than 4 in. top size are used to a larger extent than any other size coal both off-line and on-line. Off-line locomotive coal is bought on the open market and in this case the railroad has no specific interest in the coal producer. The railroad is free to specifiy the size it regards best for locomotive performance. In spite of this, there is little difference in the sizes bought off-line and those bought on-line. This total, 52,618,116 tons, is almost $6^1/_2$ times as much tonnage as is in the next-larger-size group. Also, more of it is produced for locomotive fuel than any other size in each district, except District No. 11, Indiana. (See first line of Table 2.)

Table 3 is included to show the comparison of the amount and sizes of locomotive fuel produced by each of the foregoing six districts and also District No. 9, Western Kentucky. The size groups of coals were fairly consistent with the run-of-mine sizes showing the largest increase.

In view of the recommendations in the literature for modified screenings or screenings with part of the fines removed, it is surprising that only about 492,000 tons out of the 6,700,000 tons of screenings were modified. (See second line of Table 2.) Even if the size listed as "nut" (see tenth line of Table 2) which might better be included as modified screenings, were included, it would bring the total up to only about 7,500,000 tons. The widespread use of mine run is substantiated by a series of tests run at the University of Illinois in 1917 which showed that mine-run coal (actually 5-in. resultant) is best suited for locomotive use. This series of tests also showed that 2 × 3-in. nut gave good evaporation at high rates and the lowest cinder loss (2).

Zern (3) and Robinson (4) show agreement fairly well on 5-in. or 6-in. resultant mine run with a limit of 23 per cent of minus 1 in. or $^3/_4$ in. Woodrich (5) advocates that the coal supplied for hand-fired locomotives should be of plus $1^1/_2$ in. or 2 in. when it leaves the mine. He states that the use of these coals would stimulate the coal producers to greater efforts in moving the

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

TABLE 1 LOCOMOTIVE FUEL BY DISTRICTS, 1937

	District	On line	Off line	Lake, tidewater, truck, river	Total R. R. fuel	Total all coal prod. by district	Per cent to locomotive fuel
1	Central Penn.	5,294,963	4,026,843	28,578	9,350,384	41,106,000	22.7
2	Western Penn.	7,633,309	1,757,774	345,217	9,736,300	72,253,000	13.5
3	Fairmont	3,113,145	4,710,128	495,977	8,319,250	24,010,000	34.6
4	Ohio	4,632,401	2,353,601	1,321,148	8,307,150	25,178,000	33.0
5	Michigan	4,569	35,151	....	39,720	562,000	7.1
6	Panhandle	59,055	16,670	141,815	1,017,540	4,203,000	24.2
7	Southern low-volatile	894,419	31,669	42,654	968,742	54,275,000	1.8
8	Southern high-volatile	7,552,729	5,019,310	633,058	13,205,097	91,874,000	14.4
9	West Kentucky	2,140,591	413,681	....	2,554,272	8,563,000	29.8
10	Illinois	12,027,277	2,899,641	....	14,926,918	51,602,000	28.9
11	Indiana	3,879,186	1,277,341	....	5,156,527	17,765,000	29.0
12	Iowa	638,558	84,320	....	722,878	3,637,000	19.9
13	Alabama	2,716,951	984,341	....	3,701,294	13,459,000	27.5
14	Arkansas, Oklahoma	146,493	3,665	421	150,579	1,967,000	7.7
15	Missouri, Kansas	1,576,170	308,488	200	1,884,858	9,038,000	20.9
16	Colorado	82,818	2,312	166	85,296	2,510,000	3.4
17	Colorado	1,451,611	476,108	....	1,927,719	5,515,000	35.0
18	Arizona, New Mexico	261,436	31,792	....	293,228	884,000	33.2
19	Wyoming	4,144,510	23,446	....	4,167,956	5,918,000	70.4
20	Utah	423,300	344,335	....	767,535	3,810,000	20.1
21	N. and S. Dakota	.....	.....	....	.....	2,298,000	..
22	Montana	1,946,721	171,502	....	2,118,223	2,965,000	71.4
23	Oregon, Washington, and Alaska	509,751	24,365	....	534,116	2,139,000	25.0
	TOTAL	61,929,963	24,996,383	3,009,234	89,935,580	445,531,000	20.2

TABLE 2 SIZES FURNISHED IN 1937, TOTAL ALL DISTRICTS AND SELECTED DISTRICTS, FOR LOCOMOTIVE FUEL

Description of sizes:		All districts	Selected districts 1	2	3	4	8	10	11
Straight ROM and resultant more than 4″ top size	On line	36,931,188	4,562,109	5,702,762	2,495,088	2,980,489	2,524,687	6,096,612	743,479
	Off line	15,686,928	2,302,055	1,114,164	3,060,740	1,992,274	3,879,129	1,353,285	457,206
Screenings less than 3″ top size, including 492,500 T modified	On line	6,220,594	21,288	253,816	10,542	95,447	3,316,455	1,638,073	85,026
	Off line	510,559	51,190	18,936	16,610	32,079	81,582	56,265	180,655
Chunks, top size 7″ to 6″ bottom size more than $^7/_8$″	On line	5,751,358	133,323	170,960	398,125	310,883	152,091	1,585,563	1,532,355
	Off line	2,582,213	215,812	273,993	314,257	1,828	174,841	883,593	383,034
Egg, top size 5″ to 3″ bot. size more than $^7/_8$″	On line	4,658,202	14,927	252,295	12,179	998,392	466,008	2,034,718	605,041
	Off line	2,784,823	320,690	116,757	611,040	218,581	489,214	363,212	163,956
Resultant ROM 3 in. and 4 in.	On line	3,080,414	67,567	334,882	4,613	51,960	627,411	5,242	102,238
	Off line	1,514,664	759,855	33,395	33,620	642	207,013	191	....
Lump, bot. size 3″ to $^1/_4$″ top less than 8″	On line	2,559,498	291,301	297,319	78,627	167,113	338,048	207,214	494,258
	Off line	1,052,794	246,373	191,622	331,575	72,801	47,197	17,795	83,094
Block, bot. size more than 6 in.	On line	856,078	....	31,150	106,095	8,664	6,644	157,698	196,856
	Off line	79,285	....	....	9,733	4,491	4,481	4,413	1,251
ROM, top size 8 to 10 in.; including crushed and mod.	On line	756,847	....	435,834	4,261	1,810	51,245	45,169	82,294
	Off line	303,531	8,218	....	38,801	30,587	129,289	414	103
Screenings, top size 1 in. or less	On line	603,319	163,702	44,675	3,458	119	21,155	37,033	36,875
	Off line	114,154	17,193	8,029	74,600	....	159	....	....
Nut, top size $2^3/_4$ in. to 1 in., bot. size more than 28 mesh	On line	512,459	40,746	109,616	157	17,524	48,985	219,955	764
	Off line	367,432	105,457	878	219,152	318	6,405	20,473	8,042
Total		89,935,580	9,350,384	9,736,300	8,319,250	8,307,150	13,205,097	14,926,918	5,156,527

Source: Bituminous Coal Division, "1937 Railway Locomotive and Powerhouse Fuel," Preliminary Analysis, Jan. 16, 1939.

TABLE 3 COMPARISON OF LOCOMOTIVE FUEL BY SIZE GROUPS, YEARS OF 1937[a] AND 1944[b] FOR SELECTED DISTRICTS

Description of sizes	Year	All districts	Selected districts 1	2	3	4	8	9	10	11
Lump coal and double-screened top size over 2 in.	1937	19,374,808	877,083	1,163,598	1,624,531	1,952,807	1,647,493	692,365	5,025,141	3,410,057
	1944	34,623,236	499,046	1,886,401	4,636,788	1,932,403	1,335,446	1,169,257	12,483,640	8,018,754
Double-screened top size not exceeding 2 in.	1937	1,492,155	147,172	33,170	223,317	82,378	117,153	19,300	705,104	57,612
	1944	2,336,088	58,296	176,550	380,497	31,869	479,163	71,201	872,228	154,816
Mine run and minus resultant top size over 2 in.	1937	63,903,181	7,615,848	8,293,585	6,074,051	6,160,233	9,568,849	1,784,920	8,118,499	1,380,243
	1944	87,474,978	8,204,731	9,265,025	9,770,851	9,811,046	13,697,020	5,632,733	10,565,071	1,513,507
Minus resultant and dedusted screenings, top size over $^3/_4$ in. and not exceeding 2 in.	1937	4,632,099	74,165	194,280	140,603	111,613	1,787,688	51,466	1,068,038	265,681
	1944	7,808,968	61,072	759,326	108,235	273,352	2,014,524	151,909	2,380,565	419,387
Minus resultant and dedusted screenings top size not exceeding $^3/_4$ in.	1937	506,453	178,880	51,667	42	119	26,068	6,221	9,065	36,875
	1944	986,355	243,786	62,581	130,076	34,008	105,746	77,331	102,392	1,590
Size not reported	1937	26,884	457,236	.....	256,706	.....	57,846	....	1,071	6,059
	1944	752,965	19,261	63,402	43,213	7,439	79,016	3,600	20,888	77,013
Total	1937	89,935,580	9,350,384	9,736,300	8,319,250	8,307,150	13,205,097	2,554,272	14,926,918	5,156,527
	1944	133,982,590	9,086,192	12,213,285	15,069,660	12,090,117	17,710,915	7,106,031	26,424,784	10,185,067

[a] 1937 Data rearranged in same size groups as 1944 data from "1937 Railway Locomotive Fuel."
[b] 1944 Data from Mineral Market Report, M.M.S. No. 1289.

minus $1^1/_2$-in. or 2-in. sizes. For stoker-fired locomotives, Woodrich recommends an average of 3-in. top size, with no lumps more than 8 in. maximum, and a bottom size of $^3/_4$ in.

The International Railway Fuel Association, and likewise the Railway Fuel and Traveling Engineers Association, have devoted much time to the subject of coal sizes (6). In 1942 the R.F.T.E.A. committee on coal sizing recommended run of mine with lumps reduced to 4 in. top size as the most economical size for use on hand-fired locomotives. For stoker-fired locomotives not adapted to handle smaller sizes, "properly cleaned coal not to exceed $2^1/_2$ in. × 0 in." was recommended as the most economical size. For stoker-fired locomotives properly fitted to handle small sizes, 2 in. × 0 in., or coal with a smaller top size, was recommended. It was pointed out that minus $^1/_4$-in. slack was objectionable, particularly in the softer bituminous coals. One-quarter-inch screenings should not exceed 35 to 40 per cent of the total when small sizes are used, according to this committee.

In 1943 the same committee recommended the use of mechanically cleaned coal because it gave the most efficient and economical performance. The principal basis of this recommendation was the experience of a railroad using coal largely from Districts Nos. 10 and 11, Illinois and Indiana.

TABLE 4 STEAM LOCOMOTIVES IN SERVICE OF CLASS 1 RAILROADS
December 31, 1943[c]

	Per cent	Total	Freight	Passenger	Freight or pass.	Switching
Total in service (A) (B)	100	39,498	24,353	6,399	1,738	7,008
No. built before Jan. 1, 1910	24.9	9,814	6,183	1,373	275	1,983
No. built bet. Jan. 1, 1910, and Dec. 31, 1914	23.6	9,330	5,742	1,834	212	1,542
No. built bet. Jan. 1, 1915, and Dec. 31, 1919	18.6	7,341	4,995	792	143	1,411
No. built bet. Jan. 1, 1920, and Dec. 31, 1924	17.2	6,797	4,306	1,225	235	1,031
No. built bet. Jan. 1, 1925, and Dec. 31, 1929	10.1	4,008	1,996	790	404	818
No. built bet. Jan. 1, 1930, and Dec. 31, 1934	2.4	960	485	176	171	128
No. built bet. Jan. 1, 1935, and Dec. 31, 1937	1.2	464	213	94	77	80
No. built bet. Jan. 1, 1938, and Dec. 31, 1942	2.0	777	430	115	221	11
No. installed bet. Jan. 1, 1943, and Dec. 31, 1943		429[d]				
No. installed bet. Jan. 1, 1944, and Dec. 31, 1944		329[d]				

NOTE (A) 15,734 of these locomotives are equipped with stokers.
NOTE (B) As of March 31, 1945, 39,658 locomotives were in service of Class I switching and terminal railroads. 21,710 were listed as freight, 6318 as passenger, and 11,630 as switching.
[c] Compiled by Traffic Department, National Coal Association, from American Association of Railroads, and Interstate Commerce Commission figures.
[d] No breakdown available.

The coal specifications set up by the various railroads are not standardized. Each road has its own specifications for locomotive use and these vary widely as to the sizes considered most suitable. Some roads may have facilities to further prepare the coal before using although the author has no data on this subject.

From the engineering standpoint, standardization of coal specifications for locomotives would seem necessary for the following reasons:

1 New locomotives, embodying fireboxes of different designs and different types of coal-burning equipment, etc., have replaced the older locomotives.

2 The older locomotives have been changed by the addition of stokers, different-type grates, arches, front-end design, etc., so that the former sizes of coal are not satisfactory.

3 Experimental and testing work on locomotives has proved that the size of coal should be changed for greater efficiency and economy.

The age and class of service of locomotives now in use are shown in Table 4. About two thirds of the steam locomotives in service were built before 1919 which would seem to rule out condition No. 1. These tabulations show that approximately 40 per cent of the present locomotives are equipped with stokers, although the relative age of the stoker-fired locomotives is not given.

That a great deal of work has been done on grate designs, arches, and "the combustion train" generally, is shown by the literature. There is no indication that these improvements have necessitated the change in sizes or types of coal, but every indication is that they have resulted in burning the present sizes and types more efficiently. The reduction in consumption of coal per ton-mile shown each year is due partially to these improvements, although increased steam temperatures and pressures, improved bearings, roadbed improvements, and fewer and longer trains have also helped.

The experimental and testing work in locomotives quoted in the literature is all of considerable age, although the types and sizes of locomotives tested are still in service. Undoubtedly other work has been done that has not been reported so that the answer to our third condition is not readily apparent.

The Preparation of Coal

The basic mining and preparation practices which have developed over a period of years in the various coal-producing districts have a vital bearing on the extent to which locomotive coal could be standardized without a serious dislocation of the coal industry's economy. Coal is a natural product and varies in quality, size consist, and chemical make-up, not only according to the mining district, but according to the mine and even from the part of the mine from which it comes. It can be improved or modified to a certain extent by mining methods and after it is mined, but no major changes can be made.

In 1943 there were, in the United States, 6620 mines, each having a production capacity of more than 1000 tons of bituminous coal per year. It is therefore difficult to treat the product of so many mines as one subject. No one mine, seam, or coal-producing district could conceivably furnish all the railroad coal in the country, so that of necessity the same locomotive will have to burn coals from widely separated geographic sources, and with different qualities and burning characteristics. The growing practice of running a locomotive over two or more divisions may result in using coals of widely different sources and characteristics on one trip.

If coal is loaded by hand, face preparation is practiced, that is, rock, bone, slate, and other impurities are not loaded with the coal but are rejected by the miner. This is much easier to do with hand-loaded coal than when the coal is mechanically loaded, inasmuch as a loading machine has no judgment as to what it picks up. The same is largely true of "strip" coal, for in strip-mining there is less chance to inspect the coal.

Once the coal is delivered to the tipple, it can be cleaned in two ways: The first and most universal practice is hand-picking, or removing the visible impurities manually. Sizes as small as 1 in. bottom size can be picked, although it is not generally economical to pick sizes under 3 in. The reduction of impurities by hand-picking depends to a large extent on how the impurities are found in the coal, the number of pickers employed, and their efficiency. In many mines hand-picking does not lower the ash content to a great degree, while in many others, where conditions are favorable, a large reduction in ash content is made. It also tends to waste considerable coal if the pickings are not cleaned by washing afterward (7).

It is difficult to pick run of mine and resultant, and it is the practice in some mines to separate into the various sizes, hand-pick or otherwise clean these, and then recombine them.

The second method, mechanical cleaning of coal, has been steadily increasing in popularity. One method of mechanical cleaning consists simply of "dedusting" or screening out a part of the fines and it is practiced in mines where a large percentage

of impurities are found in the extremely fine coal. The top size of coal removed may be as small as 60 mesh and as large as $^1/_4$ in. Of the 590,177,000 tons of coal produced in the United States in 1943, 145,575,849 tons, or 24 per cent, were mechanically cleaned. Of this total, 85.8 per cent or 124,374,775 tons were cleaned by wet-washing, whereas 21,201,074 tons, or 14.2 per cent, were cleaned by pneumatic methods. Forty-seven per cent of the washed coal was cleaned on jigs and 29.6 per cent was cleaned on launders and upward-current classifiers. Of the total coal cleaned, 30,326,426 tons were mined from strip pits, 125,313,687 tons were mechanically loaded underground, and 67,258,305 tons were hand-loaded underground.

Both the wet-washing and air-cleaning processes depend on the difference between the specific gravities of the coal and the impurity. The specific gravity varies for both the coal and the impurity. The "gravity" of the cleaning medium must be adjusted to remove as much of the impurity as possible without wasting excessive amounts of coal.

Although it is impossible to remove organic sulphur or some of the finely divided pyritic sulphur from coal by washing, it is sometimes possible to reduce other types of sulphur in high-sulphur coals by more than 50 per cent, depending on the efficiency of the washer, the nature of the coal, etc. The ash in high-ash coals may be reduced nearly 50 per cent, depending on the distribution of the ash-producing constituents, friability of the coal, etc. (8).

Air-cleaning of coal is usually confined to the sizes which are too small to be satisfactorily wet-washed because of the increase in moisture content likely to result from washing fines. For example, at a given mine sizes larger than 5 in. may be hand-picked, sizes between 5 in. and $^3/_8$ in. washed, and the $^3/_8$-in. $\times$ 0-in. cleaned pneumatically. When all the coal from, for example, 2 in. down is air-cleaned, it is carefully separated into its component sizes before being air-cleaned. For example, if it is desired to clean 2 $\times$ 0-in. coal, it is divided into 2 $\times$ 1 in., 1 $\times$ $^3/_4$ in., $^3/_4$ $\times$ $^1/_2$ in., $^1/_2$ $\times$ $^1/_4$ in., and so on, and run over separate air-cleaning apparatus. It is difficult to air-clean sizes with wide variations, such as 2 in. $\times$ 0 in., in one cleaner.

Mechanical cleaning is necessary and most beneficial to coals containing in the raw state a high percentage of ash and foreign matter and to mechanically loaded coal. Ordinarily when operations in an underground mine are changed from hand loading to mechanical loading, there follows an increase in the percentage of ash in the coal. When a cleaning plant is put in at the same time, the net over-all effect would be to perhaps lower the ash content by one to three per cent over the average encountered in hand-loading methods. The fact that a coal has been washed or air-cleaned is no indication of its quality, as the raw coal from one seam may be superior to the washed or cleaned from another.

The average cost of cleaning is given by Campbell (9) as about 20 to 25 cents a ton. There is an economic limit to the amount of ash and sulphur reduction which may be made by cleaning. Above this, reject losses increase to such a point that it is no longer economically justified to clean it.

Screening Coal

The equipment of mines for screening and preparing coals varies from mine to mine. This depends somewhat on the size of the mine, the amount of coal in reserve or the prospective life of the mine, and on the ultimate use for which the coal is best suited, or those sizes which will return the greatest over-all realization. The problem of screening the coal as it comes to the tipple into those sizes which are most in demand and provide the greatest return for the combination of sizes made is complex and requires both careful study and extensive planning. For example, if the coal from a given mine possesses the peculiar characteristics which make it suitable for domestic stoker coal, it may be profitable to crush some larger sizes to make stoker coal, and not at all profitable to make locomotive fuel.

Mines with no screening facilities of necessity make run-of-mine coal; and mines equipped with bar screens only are able to make two sizes such as 5-in. lump and 5-in. resultant. These mines may be equipped with crushers so that coal can be crushed to a given top size. The bulk of the mines which sell coal for many different uses are equipped with shaker screens and in many cases vibrating screens. A large number of sizes may be made by changing the size of the screens, although the number of sizes which may be made at one time at a given tipple depends on the number of tracks and their usage. The average mine probably makes and loads three or four sizes at a time. The elaborate modern tipples, particularly where mechanical cleaning is employed, may make 12 or more sizes simultaneously, part of which are delivered to bins and may be remixed or blended from the bins before loading. Mines which make only one or two sizes of coal are more likely to be found in the eastern area, that is, in Districts Nos. 1, 2, 3, 4, 6, and 8, than it is in Districts in the Middle West, Nos. 9, 10, and 11.

The practice of furnishing a double-screened washed coal such as 6 in. $\times$ 1 in., or similar size, to the railroads, is probably followed to a greater extent in Indiana, District No. 11, (10) (79.3 per cent of locomotive fuel furnished by this district in 1944 was double-screened with a top size of more than 2 in.) than any other producing district. Districts No. 9 (Western Kentucky), No. 10 (Illinois), and No. 4 (Ohio) also furnish considerable coal of this type. It would probably be most difficult for District No. 1 (Eastern Pennsylvania), to furnish sizable amounts of double-screened coals to the railroads due to the relatively soft structure of the coals. District No. 2 (Western Pennsylvania), No. 3 (Northern West Virginia), and No. 8 (Southern high volatile), could furnish a larger amount of double-screened coal and washed coal for locomotive use than is the usual practice. This, however, would only serve to emphasize the difference in quality between these coals and the others which are often used interchangeably on the same locomotive. If these districts were to furnish a double-screened coal, particularly of a smaller top size, it would mean a tremendous investment in screening equipment for many of the mines not now so equipped, and it is doubtful that the resulting small sizes could readily be sold. Many of the smaller mines which now make railroad coal exclusively could not afford to put in screening equipment, because the acreage available to them is limited. Furthermore, if at a normal rate of production a mine will work out within the next four or five years the cost of screening equipment could not be justified.

The remainder of sizes in a commercial mine which customarily produces block or lump, egg, nut, and 2-in., $1^1/_2$-in., or $1^1/_4$-in. nut and slack is not seriously disrupted by making run-of-mine or a resultant size such as 5 in. $\times$ 0 in. for railroad fuel, as the total production is utilized in the one or two sizes made at this time. If the railroads insisted on a given size, such as 5 in. $\times$ 1 in., the mine would undoubtedly have too much of either the large or small sizes at times and would be unable to operate.

Segregation and Degradation

Perhaps the greatest change that can take place between the time coal is mined and used is the segregation of the coal as it is handled. Each time it is dumped or conveyed the larger pieces tend to roll the farthest, as to the outside of the pile, and the smaller pieces tend to stay near the point of dumping. Naturally, as the difference between the top size and the bottom size of the coal increases, segregation also increases under the same conditions of handling. Segregation can be greatly reduced by careful handling, by using nonsegregating chutes and hoppers, by

layer-loading, etc., as well as decreasing the difference between the top and the bottom size.

Segregation is often confused with degradation, and with bad preparation. The user often finds it hard to believe that the fines he discovers in the middle of his coal pile were there when the coal was unloaded, and are not a result of the coal "slacking" or breaking up.

According to Fraser (11) the greatest degradation is in the handling of plus 4-in. lump coal and this amounts to about $4^1/_2$ per cent. There is very little degradation in the handling of run-of-mine, resultant, or screenings, perhaps due to the cushioning effect of the fine sizes. The same authority shows that the percentage of fines in the mine-run coal as delivered from the hard-structured seams found in Eastern Kentucky and Illinois coals is well within the limit set as desirable by several authorities for locomotive coal. All percentages are under 40 per cent. Subsequent handling, such as filling the fuel bins at coaling stations and coaling the engines would seem to increase the percentage of fines but little.

Briquetting

The briquetting of coal for railroad use in this country has been suggested and extensive tests were run by the U. S. Bureau of Mines in 1908 (12). The first tests were run on No. 3 Pocahontas coal both in the mine-run form and briquetted. As would be expected from the burning characteristics of this coal, the briquettes worked out much better than did the mine run, primarily due to the smaller amounts of fines. Later, briquettes were tried from a wide number of coals and general improvement in burning rates were reported. Apparently then, as now, the higher cost of the briquettes as compared to the cost of raw coal made their use uneconomical. Further research in briquetting methods may in time reduce the processing cost to a more practical figure.

Conclusions

Railroad locomotive fuel is such a large part of the total coal production of the United States that one standard size could not be screened out of the coal as mined without changing the sizes of coal furnished to all other coal users. The mere elimination of all the fines below a certain size would immediately lead to the accumulation of tremendous tonnages of this coal which could not be sold in a normal market.

The present practice of furnishing mechanically cleaned or double-screened coal for locomotive use is confined primarily to those districts where the practice of cleaning has grown over a number of years, due to greater mechanization of the mines, and to some extent to the large amount of impurities in the raw coal. Mechanical cleaning of coal has been developed to a greater extent in some districts and this naturally has resulted in a greater proportion of the locomotive fuel being cleaned. It is true that the trend is in the direction of more careful sizing and cleaning of coal, but it is equally true that all coal mines cannot be changed over immediately to produce a standard size of locomotive fuel. In many cases the raw run-of-mine or resultant sizes from one district will be equal or will be superior to the cleaned or double-screened sizes from another so that there may not be an actual advantage in having the size standardized.

No other type of coal-burning equipment is capable of being operated with such a wide range of size and quality of coals as the coal-fired locomotive. This has helped both the railroads and the coal industry but has also allowed in some cases the buying and furnishing of inferior quality and preparation when better grades were available.

In view of all the factors involved, it would seem that extensive tests should be made, using the different types and sizes of coal in the various sizes and types of present-day and new locomotives so as to determine the best performance under road conditions with as many sizes and qualities of coal as is possible. The published data are so inconclusive that such a procedure would seem prerequisite to making major changes. There is no question that the coal industry is willing to do anything within its power to arrive at the solution of this very important problem of locomotive fuel.

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Discussion

E. D. Benton.[3] The value of coal as a locomotive fuel is dependent upon several factors, (1) size consist, (2) calorific value, (3) price per ton, (4) per cent of ash, (5) ash-softening temperature.

Which one of the foregoing factors belongs in No. 1 position depends on the limitations of the other four. The fact that a locomotive can burn almost anything that is black is legend. This, in turn, has worked to the disadvantage of the steam engine and in favor of the Diesel. The availability or distance which a locomotive can operate is governed in no small part by the fuel it consumes. If the coal is high in ash there is a definite mileage limit when it must either be taken out of service and another engine substituted or it must suffer lengthy terminal delays while its ashpan and fire are being cleaned. If the fusion characteristics of its ash tend to promote slagging of the tube sheet, the engine is limited in mileage and must be taken out of revenue service, its fire killed and the honeycomb removed. It is often possible to favorably alter slagging difficulties by the use of double-screened coal or limiting the amount of smaller sizes and extreme fines. However, double-screened coal, after passing through storage piles and the resulting multiple-handling, suffers considerably in size consist by the time it reaches the stoker conveyer screw.

Motive power is a tool and its value is measured by the amount of revenue-tons of freight or passenger-car-miles hauled. This sounds like a trite statement, but let us expand it a little further. In the case of coal traffic, the tonnage moved from the mines depends on the ability of the coal mine to produce and sell its product. In normal markets, production capacity is usually in excess of market demands, resulting in mines not producing because of "no bills" on certain sizes. If the railroads are able to step in

[3] Assistant Supervisor, Fuels Division, Battelle Memorial Institute. Columbus, Ohio. Mem. A.S.M.E.

during stress periods and purchase such sizes as to relieve the bottleneck of no bills, then the sizes in demand move to market and the locomotive produces revenue. By and large the railroads take advantage of the steam locomotive's ability to burn a wide variety of coal by purchasing coal sizes not readily marketable. Thus the steam engine as a tool may be evaluated in two ways. There are, of course, limiting circumstances which may alter the ability of any railroad in the purchase of certain sizes.

The number of hand-fired engines and coaling facilities are the principal reasons why mine run or resultant mine run are purchased in such large quantities. Many railroads are unable to separate coals from various producing districts or certain sizes for specific services. Undoubtedly all producing districts would like the top size decreased but appear to be unwilling to do anything about the bottom size. Skimming the cream off the top simply decreases the availability of steam engines, promotes increased maintenance costs due to cinder cutting, tube-sheet slagging, etc., because when the per cent of small-size coal increases, it in turn causes increased carry-over and finally results in an increase in pounds of coal per ton of freight or passenger-car mile. Whenever you increase cost and decrease availability you simply give the Diesel the "green light."

Because of the way railroad fuel must be purchased and the general policy of storing coal during times of slow market demand, individual coals lose their identity in storage piles and its size consist suffers. But if there are not excessive fines due to the top size being sufficiently large when the coal is received for storage, then the coal from storage to the engine tank will contain less fines in spite of the degradation in storage. One thing which is not changed in storage is the amount of impurities in the coal. It would seem to the writer, since ash is the principal "hurdle" which coal must overcome, that rapid attention be given cleaning plants. If coal had no ash there is considerable doubt whether the now competing fuels would be much of a factor. As the per cent of ash increases, more and more difficulty is encountered until, as a practical matter, one of the measures for the distance coal is able to move from the mine is its per cent of ash. If the coal industry could supply the railroads with coal of not over 8 per cent ash it would have a marked influence on increase in availability and decrease in cost of operation. Another thing which the coal industry might well undertake is the active research of an economical method of drying washed coal. Wet coal in cold weather is particularly troublesome and if the removal of surface moisture could be accomplished economically the coal industry would make an outstanding contribution, not only for the railroads but industry in general, and the coal indusdustry in particular.

E. C. Payne.[4] Mr. Hardy has made some good suggestions in his paper and the writer appreciates this opportunity to emphasize those recommendations which will improve the competitive position of the coal-fired steam locomotive.

It has been proposed that extensive tests should be made using different types and sizes of coal for optimum performance on various types of locomotives. In the writer's opinion such a program should be undertaken as soon as possible so that present locomotives will reach the ultimate in efficiency and reliability and the railroads may find it less attractive to consider other types of motive power. If it should develop that prepared sizes are superior to run-of-mine, it would require a gradual change in the distribution of bituminous coal. Mines without screens would be forced to make substantial investments for tipple improvements, but this would certainly be in keeping with the times, in order that a producer meet all types of competition with coals of similar character. Considering the standardization of sizes for locomotive fuel, it is quite probable that top and bottom sizes should be adjusted to compensate for differences in physical and performance characteristics of the coals from different seams and different mines. The test program should establish the proper size for best performance. The old policy of supplying the railroads with run-of-mine and surpluses of any screen size, as a matter of convenience to the producer, undoubtedly has been a contributing factor in the failure to get the best results from the steam locomotive. Even the most expert fireman cannot obtain the best results unless he is furnished a reasonably uniform quality and size with similar burning characteristics for day-to-day operation.

The statistics cited certainly show that the locomotive has been quite versatile in the utilization of practically every size of coal produced by the bituminous industry. This versatility has been accomplished undoubtedly with a substantial sacrifice in efficiency and reliability. The locomotive with all of its limitations of weight, combustion space, and variable performance conditions should not be required to use just any size or grade of coal. It would seem much more feasible to utilize the stationary power plant for wide-range size and quality-coal application instead of using the railroads for balancing the production at the mines. Industrial steam-generating equipment has wider utilization flexibility and there is a definite trend in new power-plant construction toward the use of an even greater range of coal size and quality. Naturally, any change in the distribution of sizes produced by the bituminous industry should be very gradual, but the coal industry should co-operate to the fullest extent in improving the coal-burning steam locomotive.

It would also seem desirable that the coal industry co-operate with the builders of the modern steam-fired locomotives, and the railroads, to give wide publicity to the superior performance results that are being obtained by the modern coal-fired engines. The public should realize that modern coal-fired engines are also pulling modern streamlined trains and that some railroads believe that comfort, reliability, and economy are obtainable with the modern "iron horse." The coal industry must take other active steps to preserve the competitive position of the steam locomotive, analyzing this railroad-fuel business on an over-all economic basis rather than expect any so-called interdependence of the railroads and the coal industry to perpetuate the use of bituminous coal.

Author's Closure

Mr. Benton has brought up a very good point in that railroad fuel loses its individuality after it leaves the mine, so that any change in standards would have to be made system-wide as far as the railroad is concerned, and industry-wide as far as the coal industry is concerned.

I believe that both Mr. Benton and Mr. Payne have made an important point in that all coal preparation should be better, and that more and more coal will have to be cleaned for railroad use. This of course does not get over the hurdle as to the difference in the quality of the coal in the various districts, and the fact that these coals are mixed indiscriminately and used interchangeably many times on the same trip by one locomotive.

Perhaps when and if the suggested series of tests are made, a standard of preparation for each type of coal will be evolved that will somewhat equalize the differences in quality and burning characteristics. Thus the quality of the coal might be standardized to some extent from a performance standpoint although the size actually used would vary widely as to the district and seam involved.

Let us again re-emphasize that the blind changing of standards without such a series of tests would prove nothing but would merely cause a tonnage dislocation without guaranteeing an improvement in performance.

[4] Consulting engineer, Consolidation Coal Company, New York, N. Y. Mem. A.S.M.E.

A Failure-Sequence Indicator for Static Test Specimens

By R. W. POWELL,[1] BURBANK, CALIF.

An instrument is described which has been used to determine the location of initial failure in static tensile-test specimens. Its particular usefulness in aircraft structural development is discussed and technical details of its operation are presented.

In airframe design there are basically two major problems, the aerodynamic design, and the design of the structure. The structure must be strong enough to withstand the highest expected loading and still be as light as possible. It has been variously estimated that 1 lb of weight-saving in an airplane is worth from 10 to 600 dollars in additional pay-load capacity. Certainly weight is at a premium, and for this reason structural tolerances and factors of safety or ignorance must be small.

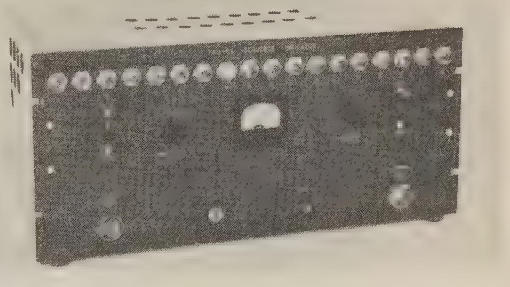
The usual aircraft structure is quite complex so that much of the data for stress analysis must come from empirical tests; and after an analysis has been made it is usually checked by additional tests.

For many years the aircraft industry has used strain gages of various types to obtain much of these data. However, strain gages are useful for measuring relatively small strains over a limited area only, and since the strain distribution in the structure under test is usually the least known factor, it is difficult to be sure that at least one of the strain gages is measuring the highest localized strain. Even if a strain gage is in the proper location and properly oriented, it may not give a good indication for extremely localized strain since a strain gage will average the strain over its active area. Of course, failure, when it occurs, is usually recognizable. However, even after failure, the location of initial failure is not definitely established because an initial failure of a weak section will usually cause progressive failure in adjacent stronger sections due to the resultant redistribution of the applied load.

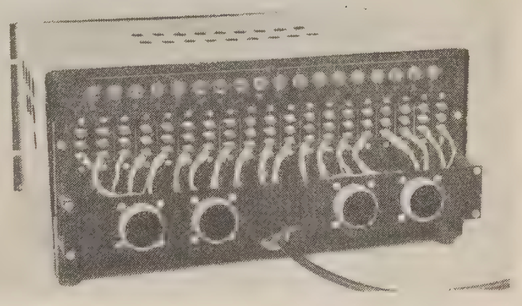
In compression tests the failure is usually in buckling, and in many cases it occurs slowly enough and with enough warning to be easily observed. However, in tension tests in which fracture or actual separation occurs, the failure often gives no warning and can progress, perhaps, at the speed of sound in the material.

For example, in a specimen of 24S-T dural, 3 ft wide, a failure initiated at one side may be complete in as little as 500 microseconds. It is highly desirable to locate the origin of such failures exactly. Ordinary techniques of obtaining data of this nature with the use of high-speed cameras or dynamic strain equipment are not very satisfactory because the interval in which the failure can occur may be from several minutes to several hours, and real difficulties involving film speed and length of film are encountered.

An ideal instrument for this type of measurement should start to record at the instant of initial failure (or slightly before), should be reasonably portable, should be simple to operate and install, and should have no effect on the strength of the structure. The



Front View



Rear View

Fig. 1 Failure-Sequence Indicator

device, Fig. 1, described in this paper incorporates most of these features.

Theory of Operation of the Pickup

The pickup or activating elements of the instrument are composed of small copper wires which are cemented to the surface of the structure. It has been determined experimentally that such wires, when cemented to aluminum alloy, will fail in tension at almost the same time as the structure, Fig. 2, for example. This is true even if the wire is more ductile than the structure, because at failure the wire is constrained by the cement bond at both sides of the rupture, and consequently the strain in the wire is highly localized.

The wire must be small in diameter so that the cement bond will be strong in comparison to the wire. It must be large enough and of proper material so that its incremental elongation will exceed that of the structure. Number 40 annealed copper wire is suitable for incremental elongations up to about 50 per cent, Fig. 3. In practice No. 40 Formex insulated wires are cemented to the structure with Goodyear MN28C cement and are spaced usually not closer than 4 in.

Theory of Operation of the Instrument

There are several possible standard methods of recording sequence of failure of the wires but for the most part they are not suitable because of the indeterminate time of failure or because

[1] Research Engineer, Lockheed Aircraft Corporation.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

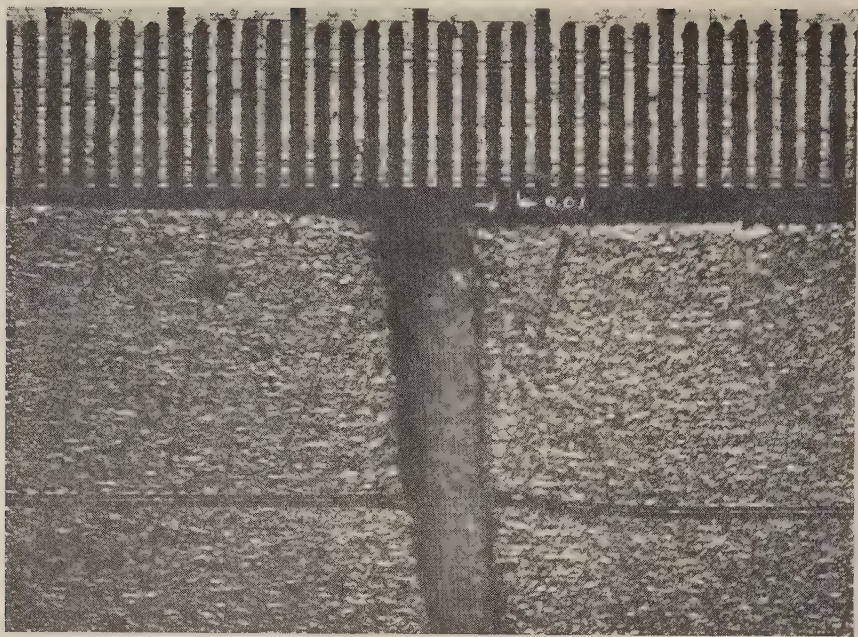


Fig. 2 Typical Failure of Failure-Sequence Wire When Cemented to 24S-T86

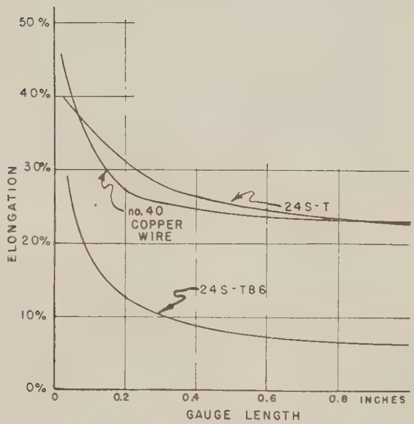


Fig. 3

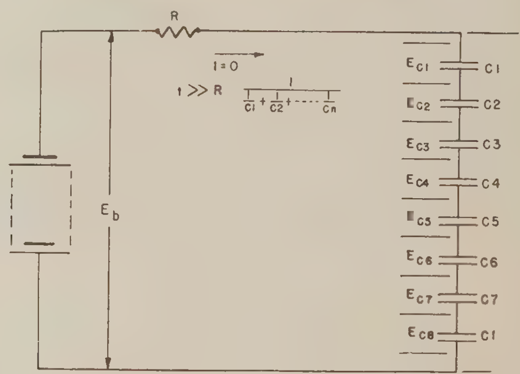


Fig. 4

of the short time involved during failure. The following method, however, has proved practicable:

Consider first several condensers which are connected in series with a battery and resistance, Fig. 4. The sum of the voltages on the condensers is equal to the battery voltage at any time after closure that is considerably greater than the time constant of the circuit. The voltage across each individual condenser is determined by its capacity, leakage, and initial state of charge.

If each condenser is short-circuited by a failure-sequence wire, Fig. 5, the charge on each condenser is zero until the first wire fails, Fig. 6. When this occurs the voltage across the first condenser is

$$E_{c1} = E_b\left(1 - e^{\frac{-t}{RC_1}}\right)$$

At the time of the second wire failure, t_1, it will have a voltage equal to

$$E_{c1} = E_b\left(1 - e^{\frac{-t_1}{RC_1}}\right) = E_1$$

After t_1 the circuit shown in Fig. 6 is modified by the addition of condenser C_2 as shown in Fig. 7. The charging rate of condenser C_1 is changed by the addition of the second condenser. After t_1 the expression for E_{c1} is

$$E_{c1} = E_b\,\frac{e^{\frac{-t_1}{RC_1}}\,C_2}{C_1 + C_2}\left[1 - e^{\frac{(t_1-t)(C_1+C_2)}{RC_1C_2}}\right] + E_1$$

At this same time the second condenser will begin to charge so that

$$E_{c2} = E_b\,\frac{e^{\frac{-t_1}{RC_1}}\,C_1}{C_1 + C_2}\left[1 - e^{\frac{(t_1-t)(C_1+C_2)}{RC_1C_2}}\right]$$

If the condensers are equal in capacity the expressions will be simplified to

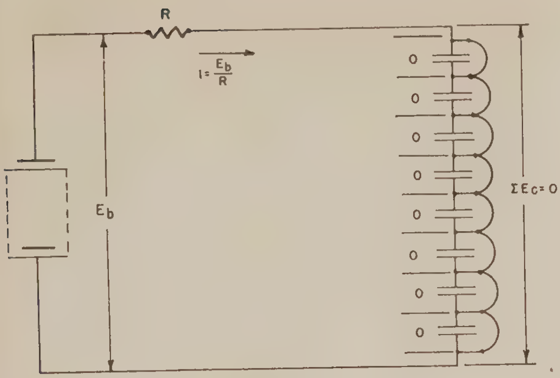


Fig. 5

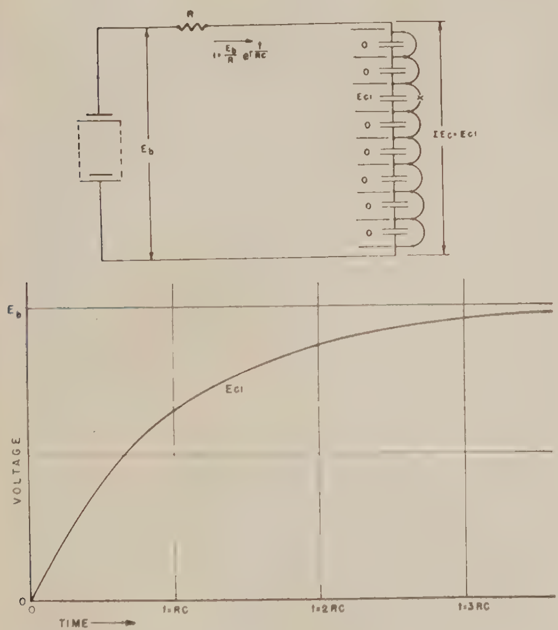


Fig. 6

$$E_{c1} = E_b \frac{e^{\frac{-t_1}{RC}}}{2}\left[1 - e^{\frac{2(t_1 - t)}{RC}}\right] + E_1$$

$$E_{c2} = E_b \frac{e^{\frac{-t_1}{RC}}}{2}\left[1 - e^{\frac{2(t_1 - t)}{RC}}\right]$$

The voltage on each condenser at t_2 when the third wire breaks is equal to

$$E_{c1} = E_b \frac{e^{\frac{-t_1}{RC}}}{2}\left[1 - e^{\frac{2(t_1 - t_2)}{RC}}\right] + E_1 = E_2 + E_1$$

$$E_{c2} = E_2$$

and the equation of charge of each of the three condensers is now equal to

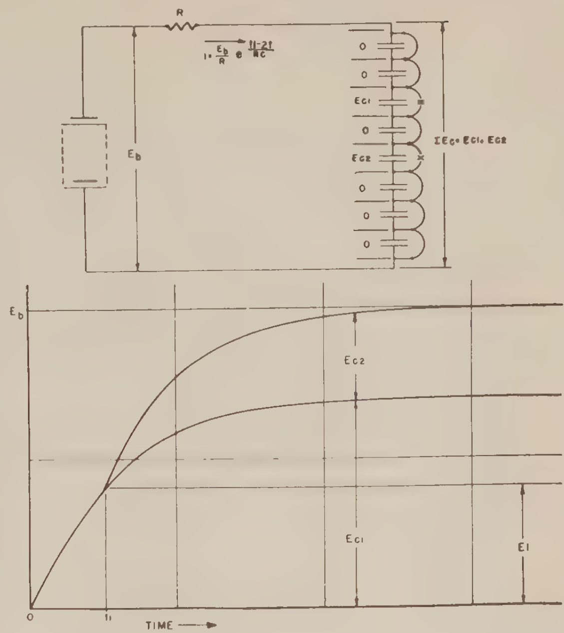


Fig. 7

$$E_{c1} = E_b \frac{e^{\frac{t_1 - 2t_2}{RC}}}{3}\left[1 - e^{\frac{3(t_2 - t)}{RC}}\right] + E_2 + E_1$$

$$E_{c2} = E_b \frac{e^{\frac{t_1 - 2t_2}{RC}}}{3}\left[1 - e^{\frac{3(t_2 - t)}{RC}}\right] + E_2$$

$$E_{c3} = E_b \frac{e^{\frac{t_1 - 2t_2}{RC}}}{3}\left[1 - e^{\frac{3(t_2 - t)}{RC}}\right]$$

and so on (see Fig. 8). It is easy to see that if the condensers are of equal capacity the charge of any one is always greater than that on condensers starting to charge at a later time, and always less than that on condensers that started to charge at an earlier time, Fig. 9. If the leakage is small, the charge on each condenser may be measured after failure by a ballistic galvanometer or a vacuum-tube voltmeter, and the sequence of, and interval between, the failure zones determined.

Unfortunately, it is difficult to keep these leakages small if the condensers remain attached to the specimen and to the battery, Fig. 10. Therefore 10 milliseconds after the failure is complete, all circuits to the condensers are opened automatically so that the slight remaining leakage will merely decrease the charge, leaving the ratios of charge between the several condensers relatively unchanged. The condensers are carefully picked for low dielectric leakage and for uniformity of capacity.

Even with the foregoing precautions the time of recording data manually for a large number of sequence wires could allow excessive leakage. The instrument is therefore provided with an automatic stepping relay that scans the condensers and indicates first failure with a numbered light. Immediately following failure, the scanning relay automatically goes into operation connecting a vacuum-tube voltmeter, which does not discharge the condenser, to each condenser successively. If any condenser has a charge of 80 per cent or more of the applied voltage a relay is actuated, the scanning is stopped, and the corresponding light indicates the location of the first failure.

In case of initial simultaneous wire failures the switch will scan

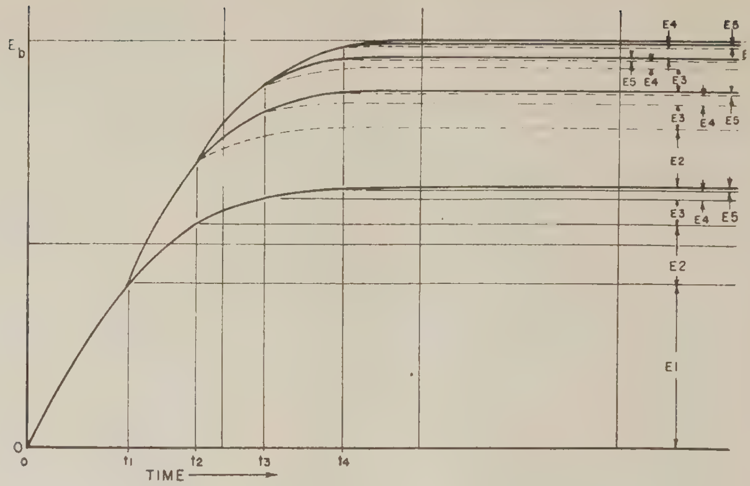


Fig. 8

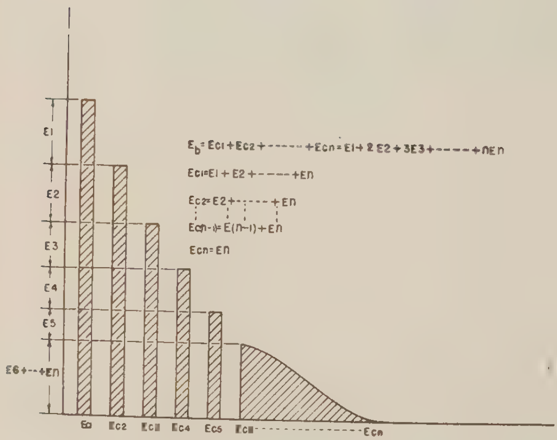


Fig. 9

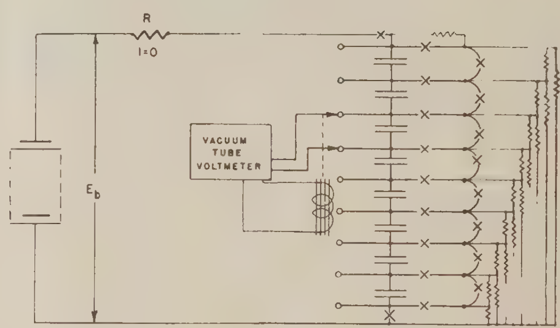


Fig. 10

through once and the operator may increase the voltmeter sensitivity until a stoppage of the scanning relay is effected. The number of the wire is indicated as before by the lights, and the charge on the condenser is indicated by the meter on the front panel. When these data have been recorded the operator may continue the scanning to another stoppage by pressing the "step" switch to restart and adjusting the sensitivity control.

As the instrument is currently used, any number of failure-sequence wires up to 18 may be attached (if necessary, it may be designed for any number of wires). It has an initial time constant of 500 microseconds, which means that a difference in time of failure of 20 microseconds of the first two wires may be detected, or that the second wire to fail may be determined if it occurs as much as 2 millisec after the initial failure. This time range may be changed to other values by changing the size of the resistor or the condensers.

Conclusion

The device has been used on static tests of aircraft structures involving 24S-T, 24S-T86, and plexiglas, and in each instance the data obtained were wholly consistent and apparently valid. In several cases structural redesign was based, in part, on the indications obtained by this method, and improved strength and joint efficiency were obtained.

Laminated Edge Attachment for Acrylics

By E. H. SNYDER,[1] BURBANK, CALIF.

As it had been observed that pressure loads encountered on such large acrylic parts as canopies, astrodomes, blisters, noses, windshields, and the like, caused frequent replacements, particularly at attachment points, it was determined that an improved method of attachment must be developed. In this paper is described an improved method of edge attachment for the canopy of the P-80 airplane in which aerodynamic and cockpit pressure loads have imposed higher loads than usual in the acrylic canopy material. In tests, laminated edging integrally attached to acrylic sheeting consistently approached and in certain cases developed the full strength of the cast acrylic sheeting.

Introduction

An examination of a number of service reports on large acrylic parts such as canopies, astrodomes, turrets, blisters, noses, and windshields revealed that replacements were comparatively frequent. Most replacements were necessitated due to crack formations originating at attachment points. The method of edge attachment assumed a role of increased importance with the introduction of high-performing aircraft such as the P-80 airplane. Aerodynamic and cockpit pressure loads encountered in the P-80 imposed still higher loads in the acrylic canopy material.

The present conventional method of attachment requires the drilling of a considerable number of holes along the edges of the canopy. These holes have to be large enough to permit the installation of rubber grommets and steel liners which serve to minimize the possibility of crack formation. A new and improved method of edge attachment was recently developed which consists of bonding integrally a laminate of relatively high strength to the canopy edges. Attachment can then be made directly through the laminated edge strips without the use of grommets and liners.

Until recently the manufacture of such a reinforced edge has not been practical. However, with the development of catalysts which promote rapid polymerization or solidification of the methyl-methacrylate liquid monomer in the presence of ultraviolet light, the bonding of edge strips to the canopy has become feasible.

Preliminary tests showed that this type of attachment was at least twice as strong as the conventional edge attachment. Thus a series of tests was instigated to develop a design and technique which was commercially applicable to regulation canopies. The final objective of this program was to apply the new attachment method to actual airplane parts such as canopies and windows and determine their strengths when tested under conditions encountered in the actual flight of the airplane.

Discussion of Laminating Resin

The acrylic sheeting used in present-day aircraft is made by casting methyl-methacrylate monomer (a water-white liquid) between two glass plates. The addition of a catalyst and the application of closely controlled heat cause the monomer to polymerize or cure into a solid state. Polymerization is accompanied by an exothermic reaction, which, if not properly regulated, would cause the rate of polymerization to progress beyond control and result in the formation of bubbles or perhaps violent boiling of the monomer. This is reviewed in order to point out that polymerization must proceed at a rather slow rate. Several days or a week, depending upon the thickness, are necessary for the curing of the usual cast sheet.

When attention was first directed toward a laminated edging for acrylics, it was conceivable that layers of cloth could be cast integrally around the edges of a cast acrylic sheet. This, however, was beyond the scope of the average aircraft plastics laboratory. It was hoped that a quicker and more practical method could be developed to attach integrally a laminated edging to methyl-methacrylate sheeting.

A light-sensitive catalyst,[2] which caused strips of cloth, impregnated with prepared acrylic monomer to "jell" in 5 to 10 min. and cure into a hard mass in approximately 1 hr of exposure to sunlight, appeared to be the solution to the laminated edge attachment.

Methyl-methacrylate monomer is waterlike in consistency. To make it more suitable for laminating operations, it was thickened or bodied so that it would not run off the laminated or bonded item before the polymerization or solidification reaction was complete. The monomer was bodied or partially polymerized by adding a very small amount (0.2 per cent by weight) of the light-sensitive catalyst and refluxing until the suitable viscosity was attained. Care was exercised in determining the amount of catalyst necessary so that none in excess of the specified amount would be added. Too much catalyst would cause the exothermic reaction which occurs during refluxing to proceed so rapidly that uncontrollable boiling results in complete polymerization. When this bodied resin cooled to room temperature, an additional 3 per cent by weight of the catalyst was added to make it ready for application. In this condition it must be used within a few hours or stored in a refrigerator to prevent complete solidification or polymerization.

Fabrication of Joint Specimens

The test specimens used to determine the strength of the various types of edge attachments were made from $^5/_{16}$-in-thick acrylic sheet stock $4^1/_2$ in. long and 3 in. wide. The thickness and width were thus chosen to approximate 1 sq in. of cross section. The laminates used for the edges of the new type joints consisted of Fiberglas or other fabrics laminated with the same resin that is used for making the actual blown canopy.

Type I—External V-Joint. Some of the first acrylic tension specimens, containing an external "V" on the ends (see Fig. 1) were prepared "wet;" that is, laminae of glass cloth which had just been dipped in the acrylic resin were laid symmetrically on either side of the scarfed ends and the whole assembly sandwiched between cellophane-covered glass plates. After exposure to sunlight for approximately 1 hr, the resin polymerized or cured, and the specimen was then ready for trimming and mounting. It has been found that the time necessary for curing can be substantially reduced by using a good strong source of ultraviolet light instead of natural sunlight.

Since cut Fiberglas cloth tends to fray and become unwoven at the ends, it was practically impossible to keep a few loose glass

[1] Research Engineer, Lockheed Aircraft Corporation.

Presented at the Rubber and Plastics Session of the Aviation Division Meeting, Los Angeles, Calif., June 3–5, 1946, of The American Society of Mechanical Engineers.

[2] Manufactured by C. C. Sachs, No. Hollywood, Calif.

threads from floating out upon the clear portion of the specimen and being cured in place.

To avoid the problem of these loose Fiberglas threads and also the undesirable handling of the wet laminae in the final preparation of specimens, laminates were cured into flat sheets in the following manner:

Each piece of fabric (about 23 in. square) was dipped into a pan of the prepared resin and laid in place on a 2-ft-sq piece of cellophane-covered tempered glass plate. As each piece was stacked on the glass plate the trapped air bubbles were worked out with a suitable squeegee. When all layers were in place, micarta strips $^1/_2$ in. wide and of proper thickness were nested around all four edges to serve as spacers and also to prevent too much resin from escaping. A second 2-ft-sq piece of cellophane-covered tempered glass plate was then placed over the stack of impregnated fabric and clamped in place with a number of standard C-clamps. After curing, the laminate was machined into strips for bonding to the acrylic-specimen edges. These flat laminated sheets were made in thicknesses equal to one half that of the acrylic component in order to facilitate beveling of the edges of strips which were machined from the flat sheets. Figs. 1 and 2 illustrate how the beveled strips were placed together to form a mating "V." The successful rapid machining of these strips was accomplished by using cutting tools with cemented-carbide tips. The prepared resin was placed on all mating surfaces and the strips cured in place by exposing the assembly to sunlight.

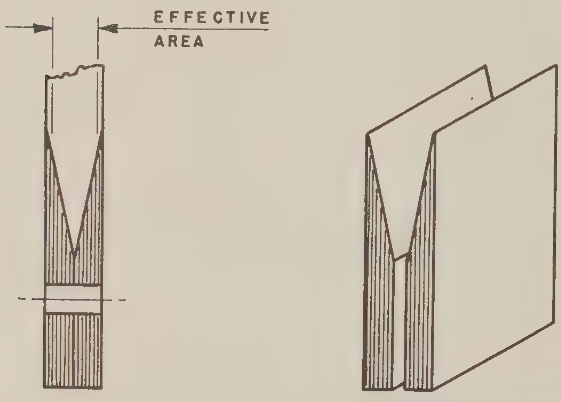


Fig. 1 External V-Joint

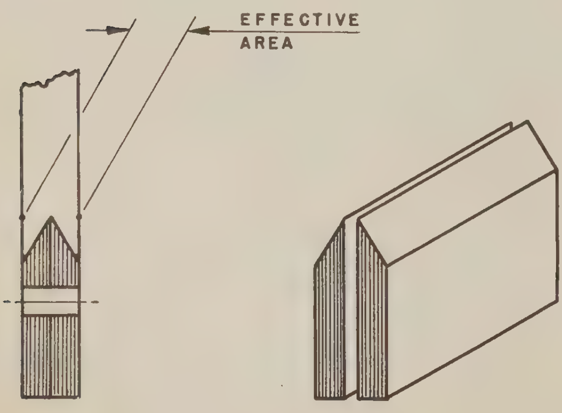


Fig. 2 Internal V-Joint

Tests made on this type of attachment showed that it had a strength equivalent to that made with individual wet laminae but it, too, had some distinct disadvantages: (1) Due to the shrinkage problem which is difficult to handle and because of the fact that resin cast in this manner has a lower strength than the original acrylic sheet, the scarf of the edge strips had to be of a length which would allow a transition gradual enough to produce an attachment of optimum strength. The length of scarf necessary caused the opaque laminated edge to extend about $^1/_2$ in. above the usual canopy mount, thus decreasing visibility. (2) Any excess resin which was present tended to be forced over the clear adjacent surface. This necessitated laborious polishing to restore the original optical qualities.

Type II—Internal V-Joint. This type of joint was also made by bonding the cured laminated strips to the prepared acrylic edge, but in this case the scarfed edges of the laminate were fitted into an internal "V" in the acrylic edge (see Fig. 2). Tests made on this type of specimen showed that it was stronger than an equivalent external type of joint; and that an internal "V" $^1/_4$ in. deep was as strong as any $^3/_4$ in. external V-joint. In addition, the disadvantages of the external type of joint were eliminated, because it was now possible to keep the opaque edge flush with the canopy mount, and any excess resin present when the edge strips were bonded in place was forced over the opaque laminate instead of the clear adjacent acrylic sheet.

Figs. 1 and 2 illustrate both types of joints and clearly indicate the advantages of the internal type of attachment.

Results of Tests

Tension Tests. As a result of stress concentrations, the efficiency of edge attachments in direct tension vary from 30 per cent (for bolted attachments, using rubber grommets and steel liners) to approximately 85 per cent for laminated edge attachments. In terms of average stress at failure, the conventional bolted attachment developed approximately 2500 psi and the laminated edge attachments ranged as high as 6500 psi.

Bearing Tests. Since the specimens containing the laminated Fiberglas edges were failing outside of the actual joint and causing no appreciable elongation of the attachment holes, it was decided that a laminate containing all laminae of Fiberglas was too strong. It was hoped that a laminate of cotton or linen cloth would solve the problem and eliminate the undesirable machining qualities of laminated Fiberglas, but the linen and cotton cloth-filled laminates failed in tension before there was any evidence of bearing failure. A number of laminated bearing specimens were thus fabricated using a combination of cotton cloth and Fiberglas cloth. Elongation of the attaching holes was thereby controlled by substituting cotton cloth for certain Fiberglas laminae.

Fatigue Tests. Test specimens of the conventional rubber grommet-and-liner type and those containing the internal V-type laminated edges were made identical to those used for tension tests. These were tested dynamically at various loadings and frequencies until failure occurred. The fatigue life of the laminated attachment was found to be over 1000 times that of the bolted type.

Applications of the Laminated Edge Attachment

(*a*) *Aircraft Canopies.* From the promising results obtained from the small test specimens it was decided to prepare a full-size canopy of the type used on the P-80 airplane with the new type of edge attachment and tested under simulated flight conditions.

The P-80 canopy is of the one-piece "bubble" type, which lends itself to "free-blowing" or vacuum-forming. In this process of forming, a heat-softened acrylic sheet is clamped to a vacuum pot and free-drawn to a controlled depth. The shape of the

vacuum-pot opening gives the general outline required for the canopy edge. After forming is completed the enclosure is finish-trimmed for mounting.

The P-80 production canopy is fabricated from $^3/_8$-in-thick acrylic sheet. However, the new canopy which was to contain the laminated edging was formed from a $^5/_{16}$-in-thick acrylic sheet.

Following is a step-by-step description of the detailed operations required in the preparation of the laminated strips and their subsequent attachment to the canopy edges:

1 Preparation of laminates: As a result of the bearing tests made on various types of laminates, a combination muslin-and-Fiberglas cloth laminate was chosen, which would show an incipient bearing failure in the attaching holes at a load slightly below the average breaking load of the acrylic specimens. This would allow a more uniform distribution of the loads.

The laminated-edge-strip material, which was considered as the optimum for this type of attachment, consisted of two layers of coarse-woven muslin with a layer of ECC-11-148 Fiberglas cloth on each outer surface. These laminates were made in the same manner as previously described, and their nominal thickness was chosen as $^1/_{64}$ in. less than $^1/_2$ of the minimum thickness of the canopy edge, in order to make it possible to produce a completed edge strip with surfaces flush with the canopy surfaces. The laminated panels were cut into strips $1^1/_8$ in. wide and then beveled on one edge to mate with the canopy edge and also on each end to mate with adjoining strips.

It has been found that the variation in thickness of commercial canopies was as much as 0.100 in. due to manufacturing tolerances; in addition, the thickness variation may further be increased by the canopy drawing or stretching operation. However, by placing a strip of wet uncured resin-impregnated fabric between two cured laminated strips, it was possible to compensate for any variation in thickness normally found.

2 Preparation of canopy edges: The edges of the special blown acrylic canopy were first trimmed shorter than a conventional canopy to allow for the addition of the laminated edge. The edges then were machined to form an internal "V," since the internal V-joint had proved to be superior from the standpoint of strength and ease of fabrication. The internal "V" was machined quite readily by the use of a portable hand router and a special bit.

Transparent contoured fixture strips, which were necessary to hold the laminated edging to the canopy and at the same time permit light to reach the joint to effect curing, were obtained by sawing off the edges of scrap production canopies. These contoured strips were then clamped on either side of the canopy edges so that they extended approximately $1^1/_2$ in. beyond the machined edges of the canopy. They served their purpose effectively for the forming of the laminated strips and also as transparent fixtures for holding the formed laminated strips in place during the curing cycle.

3 Forming of laminated edge strips: The prepared laminated strips were formed to mate with the ever-changing contour of the canopy edges. Since the resin used to make the laminates was a "thermoplastic" (softens under heat and hardens when cooled), it was only necessary to heat the strips for several minutes in an oven at 300 F and then hold them in place in the grooved canopy edges until they cooled. This was facilitated by the transparent fixtures (edges from scrap canopies) clamped to the prepared canopy edges. The hot pliable laminated strips were forced down onto the machined edges of the canopy and clamped to the overhanging transparent fixture strips until they cooled enough to retain their new contour.

4 Curing of preformed laminates to canopy: The next step in the fabrication of the edge attachment was to sandwich a wet resin-impregnated Fiberglas lamina between two cured formed laminated strips to build up the thickness of these edge strips to equal that of the canopy edge. Excess resin from the wet laminae flowed into any spaces created by the varying edge thickness of the canopy and was thus cast and cured into place around the canopy edges to form a smooth, clear, and continuous transition from the opaque laminated edge to the clear acrylic canopy material.

To simplify tooling, the attaching operation was accomplished in two steps. In the first step the cellophane-covered transparent fixture strips were clamped only to the inside edges of the canopy. The chamfered edges of the inboard cured laminated strips were "doped" with resin, forced into contact with the canopy edges, and clamped in position to the fixture strips for curing. The second curing step was taken after removal of the inboard transparent fixture strips. The wet impregnated Fiberglas lamina was then placed on the previously attached inboard laminated strip. Next, the chamfered edges of the outboard cured laminates were doped with resin and carefully placed in their correct position on top of the wet lamina. The cellophane-covered outboard transparent fixture strips were then placed on top of this assembly and clamped in place to the canopy. As a final operation to assure correct positioning, the outboard cured laminates were forced into intimate contact with the machined edges of the canopy before finally clamping in place to the outboard fixture strips for curing.

The canopy described, having the laminated edging, was subjected to pressurization tests along with production canopies containing the rubber grommet-and-metal liner type of attachment. The laminated-edge-type canopy, blown from $^5/_{16}$-in-thick acrylic material, withstood pressurization loads 25 per cent greater than those which produced failure in the $^3/_8$-in-thick production canopies. This $^5/_{16}$-in-thick canopy, containing the laminated edging, was not tested to destruction, but was saved for possible investigation of other variables.

This canopy was actually one of three fabricated with the laminated edging for testing purposes.

The first one, formed from $^5/_{16}$-in-thick acrylic sheeting, failed prematurely at a load higher than that withstood by any previous production canopy, when part of the mounting structure failed.

After reinforcement of the mounting structure, the second $^5/_{16}$-in-thick canopy was tested as previously described and then saved for possible future investigations.

The last and third canopy was formed from $^1/_4$-in-thick acrylic sheeting. Considerable speculation awaited the test results of this particular canopy. Unfortunately, however, it was cracked during installation, so further testing was discontinued.

5 Fabrication of canopy from a flat developed pattern: Near the end of the war an ingenious method was developed by a West Coast plastics fabricator[3] for the free-blowing of aircraft canopies from a flat developed pattern. This method consisted of machining a sheet of acrylic to the exact flat developed pattern of the canopy. After heating the flat blank in a specially constructed oven, it was carefully laid on a blowing jig and securely clamped around all edges to effect a perfect seal against air leaks before the blowing operation was started. Positive air pressure was used rather than pulling a vacuum. This enabled the operator to check the part during the blowing process and control the air pressure throughout the operation.

The canopy for North American's P-51 "Mustang" fighter plane was "free-blown" from a flat developed pattern using this method. With North American's permission a P-51 canopy was secured by Lockheed in the flat pattern, and Lockheed's laminated edge attachment was cured to the edges of this pattern.

[3] Stacks Plastics, Culver City, Calif.

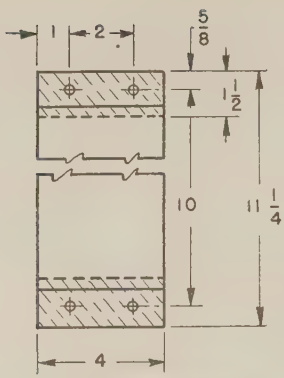


Fig. 3 Creep Specimen

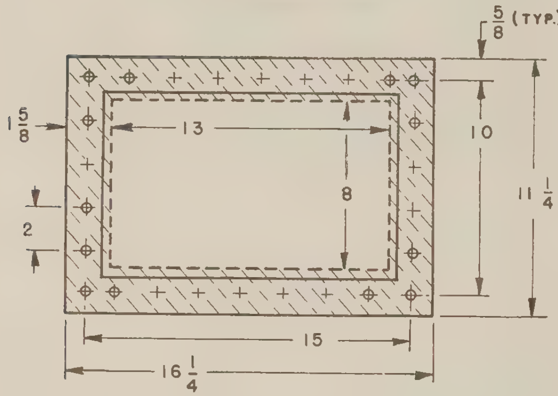


Fig. 4 Shear Specimen

Since the curing of the laminated edge strips to a flat sheet obviously entails less difficulty than curing strips of compound curvature to canopy edges of compound curvature, it was desired to determine whether or not the addition of the laminated edging to the flat pattern would alter the forming characteristics of this particular free-blowing method.

No difficulties were encountered in the final operation of blowing this flat developed pattern into a finished canopy with laminated edges.

The hand methods used in fabricating the special laminates and their subsequent attachment to the canopy edges were necessarily laborious. However, when production methods are considered, it is apparent that certain operations could be eliminated or greatly simplified. The method of blowing canopies from a flat developed pattern would further facilitate production if applied to the laminated-edge-type canopy.

(*b*) *Acrylic Panels.* Since joints of 85 per cent efficiency were consistently being produced of laminated strips to acrylic sheeting and developing stresses as high as 6500 psi, the possibility of using stressed acrylic panels was given consideration.

One of the characteristics desired to be known about an acrylic panel, which incorporated a laminated edge attachment, was its behavior when subjected to shear loads both at normal and extreme temperature ranges.

1 Shear tests. The method used in fabricating the shear test specimens was essentially the same as that used for fabricating the canopy laminated edge attachment. Three test specimens were made from $^3/_8$-in-thick acrylic sheeting according to Fig. 4. These panels were mounted in a steel shear frame having pin-connected corners, and the assembly was attached to a rigid floor structure. The shear loads were applied to the panel frame by means of hydraulic jacks and transferred to the plastic panel through the $^1/_4$-in-diameter attaching bolts. Total shear deflections were measured by means of a dial gage. A temperature-controlled enclosure was built around the test panel so that various temperatures could be accurately maintained.

The first panel was loaded at a temperature of 65 F in increments of 50 lb per in. (see Fig. 5). The application, removal, and reversal of shear load was repeated for 1000 cycles at 150 lb per in.; 1000 cycles at 300 lb per in.; and 3000 cycles at 500 lb per in. This repetition of loading showed no change in the shear deflection noted at the beginning of each cycling test. A maximum static load of 850 lb per in. was applied at room temperature without any serious permanent effects.

When the foregoing tests were completed the temperature was lowered to —55 F, and a load of 300 lb per in. was applied and maintained for 2 hr. Upon removal of all load there was no indication of permanent set. The load was then gradually increased to a maximum value of 650 lb per in., whereupon the specimen suddenly shattered.

The second panel was installed and the temperature lowered to —40 F. Loading was applied in small increments, and again when the value of 650 lb per in. was reached, the panel suddenly shattered.

The third and last panel was tested at 130 F. Loading was applied in 50-lb-per-in. increments until a load of 600 lb per in. was attained. Failure occurred approximately 90 sec after application of the 600-lb-per-in. load.

The amount of deflection at various temperatures is compared in Table 1.

2 Tension creep tests. The purpose of these tests was to determine the behavior of an acrylic panel incorporating laminated Fiberglas edges under prolonged tensile loading at various temperatures.

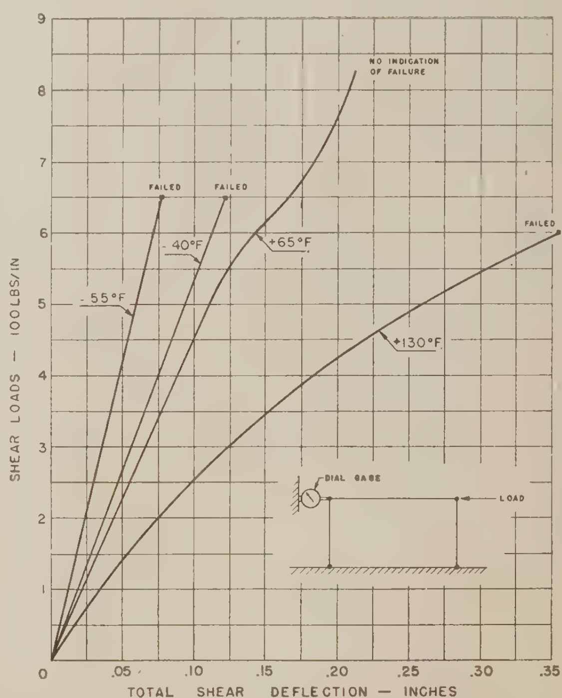


Fig. 5 Shear Load Versus Deflection for Laminated Edge-Acrylic Panel

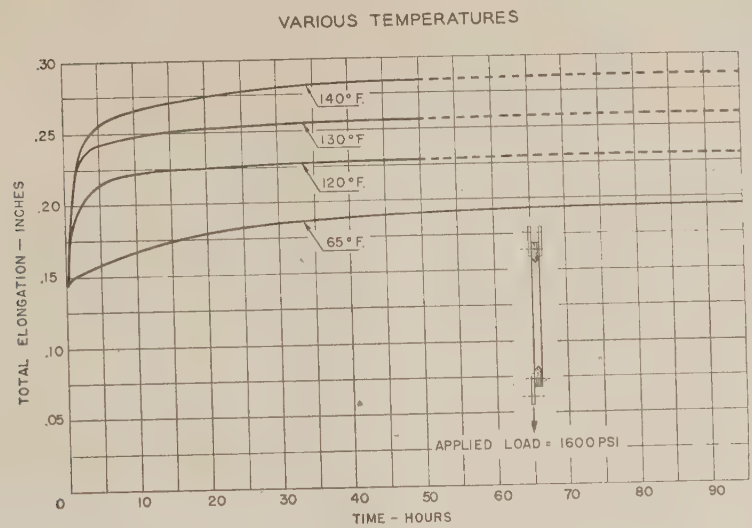


Fig. 6 Acrylic Creep Tests; Elongation Versus Time at Various Temperatures

TABLE 1 DEFLECTION OF TEST PANEL AT VARIOUS TEMPERATURES

Temperature, deg F	Shear load, lb per in.	Total deflection, in.
— 40	650	0.120
70	650	0.155
130	600	0.354

Test specimens were made from $^3/_8$-in-thick sheet stock with the Fiberglas edge attachment at each end, as shown in Fig. 3.

Specimens were suspended in a controlled-temperature enclosure by means of a double-shear-type fitting. At the lower end, arrangement was made for applying 2400 lb (600 lb per in.) through a single shear-type fitting, so that there was an eccentricity of $^9/_{32}$ in. between the suspension and the applied load. Two dial gages, calibrated in thousandths of an inch, were installed, one on either side of the specimen, to indicate the total elongation in the 10-in. gage length. Time versus elongation curves for different temperatures are presented in Fig. 6.

The tests indicated that when the load was first applied the rate of creep was high, but that it decreased rapidly over the first several hours.

Creep effect is greatly influenced by temperature, as may be seen by referring to Fig. 6. Considering 10 hr as a reasonable period for "long-duration" loading, it is apparent that under 600-lb-per-in. load and at room temperature, the observed creep constituted only an additional 12 per cent above the elastic (instantaneous) elongation, while at 140 F the deformation would almost double itself. Under this load and at any temperature up to 140 F, 10 hr appears to be sufficient time for a reasonable stabilization of the elongation.

At the conclusion of the tests no permanent elongation of the holes in the Fiberglas-reinforced edge was apparent. It is improbable that any appreciable amount of the observed creep effect occurred in the laminated edges.

Summary

In summarizing, it is pointed out that a laminated edging integrally attached to acrylic sheeting consistently approached and in certain cases developed the full strength of the cast acrylic sheeting.

Where further reduction in load concentrations is necessary at attachment points, various combinations of filler material may be used in the laminated edging to produce controlled bearing failure in the attaching holes.

Rotational Drop-Testing of Airplane Main Landing Gear

By W. H. GAYMAN,[1] BURBANK, CALIF.

The immediate physical objectives of drop-testing are presented, together with a general method for conducting rotational tests that have proper contact velocity and energy relationships coincidental with the absence of undesirable dynamic reactions at the pivot axis. Application of the method to tests of the main landing gear of the Models PV-1 and PV-2 patrol bombers is discussed. Conclusions drawn from dynamic-load measurements emphasize the possibility of attaining excessive loads in the drag-resisting structure when the impact surface is horizontal and the gear is inclined to simulate resultant loading of any particular design condition. It is implied that the practice of dropping the test gear on an inclined platform during translational drop tests may result in unconservative loading unless deliberate compensation is afforded.

Introduction

DROP-TESTING of a newly designed airplane landing gear is commonly included in the structural-test program carried out with a static-test airplane. These tests have as their immediate purpose the demonstration of the degree of compliance with design specifications regarding, fundamentally, the energy-absorption characteristics and related dynamic behavior of the landing gear. While the tests are generally preceded by landing-gear-jig drop-tests, conducted incident to developing or checking the shock-strut design, their specific objectives are the determination of landing-gear functional characteristics with effects of airframe elasticity included, and the checking of the structural integrity of the gear and its supporting members. The latter consideration is in general complementary to other required structural tests.

In the case of an airplane with conventional landing gear, it has been customary to conduct rotational drop-tests independently for both tail gear and main gear. In tests of the former, the airplane is usually permitted to "rotate about the main gear axles" as the tail is raised and dropped. In tests of the main gear, the tail-wheel axle or some other convenient item is utilized as the pivot axis.

The author has failed to discover any literature justifying the basic concept of rotational drop-testing of main landing gear as commonly executed; he presumes that its acceptance has been due to its advantages over translational drop-testing of the entire airplane because of its inherent freedom from additional test variables arising from the following:

1 The problem of assuring simultaneous or at least reproducible contact conditions of main wheels and tail wheel.

2 Interactions of main landing gear and tail landing gear attributable to airplane-pitching accelerations due to different stiffness rates and strokes of the two basic types of landing chassis.

This is not to imply that translational drop tests are conducted infrequently; the advent of tricycle landing gear has given impetus to development of this technique.

However, it is not the purpose of this paper to present a discussion of methods other than rotational drop-testing, or of the philosophy of drop-testing in general. Rather, the intent is to present, from a test engineer's point of view, a discussion of methods that have been under development since the beginning of a program for drop-testing the PV-1 patrol bomber.

1 GENERAL DYNAMIC CONSIDERATIONS

Drop-Test Problems

There appear to be no standard interpretations of drop-test criteria as they reflect on the major aspects of the physical problem. Consequently, the test engineer must formulate his own interpretations from the pertinent general specifications, or perhaps depart from the letter of the specifications in order to rationalize some of the apparent physical incompatibilities. For example, while the drop-test specifications of various agencies are specific enough in defining "drop-height" and "load-factor" requirements for landing gear of various types of aircraft, in general they either leave too much latitude in the test-airplane weight distribution or impose limitations that have no fundamentally rational basis. In the manner of support of the test airplane at its pivot attachment, a specification may impose restrictions, which, if rigidly adhered to, can unwittingly cause local failure of the aft fuselage section in the attempt to obtain tests of the main landing gear, or conversely, failure of the main landing gear or its supporting structure in rotational drop tests of the tail gear.

That instances of such failures exist is a matter of record. In some cases a "too literal" interpretation of the pertinent specification may have been the cause; in other cases lack of precedent, combined with insufficient appreciation of the dynamic aspects of the problem may have been a factor. In any case, however, possible danger to test personnel as well as direct and indirect costs occasioned by such accidents have warranted the direction of efforts to evolve a practical rotational drop-test technique that satisfies all of the basic requirements. Because of the length that would be required to cover specific problems arising in rotational drop-tests of auxiliary gear, the present discussion is necessarily limited to rotational drop-testing of main gear, although the basic dynamical approach is applicable to either.

Rotational drop-testing is simply a convenient means of obtaining landing-gear performance characteristics. Because of the variation of linear acceleration with distance from the center of rotation, this type of test is not intended to constitute a structural test of the entire airplane, but rather is limited to a relatively localized region embracing the test gear.

Test Setup Parameters

In order to impart a maximum of physical meaning to the interpretation of the test results it is important that the test setup

[1] Formerly, Research Engineer, Lockheed Aircraft Corporation.

Presented at the Aviation Division Meeting, Los Angeles, Calif., June 3–5, 1946, of The American Society of Mechanical Engineers.

parameters be consistent with the basic design conditions. Hence for a given test:

(*a*) The magnitude of the static reaction on the test gear should be the same as that established by the design condition. (A value based upon an "equivalent-mass" analysis may be used where applicable.)

(*b*) The wheel contact velocity for a given free-drop height, as measured at the axle (or bottom of the tire), should be the same as that to be obtained in a hypothetical translational drop from the same drop height.

(*c*) The energy to be absorbed by the test gear, resulting from a drop from a given height, as measured at the axle, should be the same as that part of the airplane's total potential energy that would be absorbed by the test gear in a purely translational drop-test from the same drop-height, effects of interactions of main gear and auxiliary gear being discounted.

(*d*) The direction of the impact load on the gear, relative to the shock-strut axis, should be the same as that established by the particular design condition.

Requirement (*c*), as presented here, is really repetitive since ideally the satisfaction of conditions (*a*) and (*b*) automatically fulfills (*c*).

In addition to the attainment of these basic objectives, the test engineer has a few of his own requirements to meet, the foremost, aside from safety for test personnel, being that he avoid structural damage to the test airplane resulting from dynamic loading at the pivot point.

An analytical evaluation of the degree to which all of these criteria may be satisfied may be made by consideration of the dynamics of the compound pendulum.

Consider the case of the rigid pendulum subjected to constrained rotation, Fig. 1. If the pendulum has its motion arrested suddenly by a stop, the nature of the instantaneous loading at the pivot point is dependent upon the following factors:

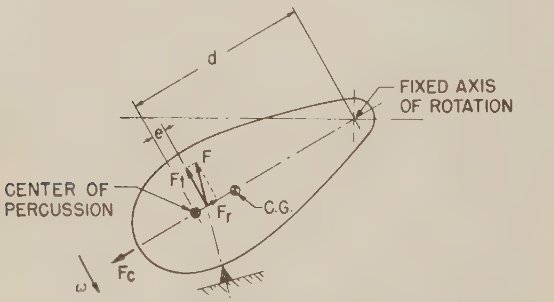
(*a*) The instantaneous value of the angular velocity.

(*b*) The magnitude of the force exerted by the stop.

(*c*) The direction of the stop force relative to the line containing the center of percussion, the center of gravity, and the center of rotation.

(*d*) The displacement of the line of action of the stop force from the center of percussion of the pendulum.

If the line of action of the stop forces passes through the center of percussion, the tangential component produces no reaction at



F = INSTANTANEOUS IMPACT FORCE EXERTED BY STOP
F_t = TANGENTIAL COMPONENT OF F
F_r = RADIAL COMPONENT OF F
F_c = CENTRIFUGAL FORCE
RESULTANT REACTION AT AXIS OF ROTATION, $R = \sqrt{\left(\frac{e}{d}F_t\right)^2 + \left(F_r - F_c\right)^2}$

FIG. 1 "FIXED ROTATIONAL" COMPOUND PENDULUM

the pivot axis. However, the radial component adds algebraically to the simultaneous value of centrifugal force.

If, in the case of a semielastic body, the motion is arrested by a sprung stop, the natural frequency of which is very low compared with the natural axial and flexural frequencies of the body, the behavior of the system is quite comparable with that of the inelastic pendulum.

The rigid-body type of analysis applied to a rotational drop-test setup indicates the nature of the problem of avoiding undesirable loading at a fixed pivot point. Consider a case for tests of main landing gear. If the airplane wheels are dropped on a well-greased horizontal surface, it is generally assumed that the landing reactions act vertically. This being granted, it becomes necessary to orient the pitching attitude of the airplane so that requirement (*d*) noted previously is fulfilled, that is, so that the direction of the shock-strut axis relative to the impact load conforms with the design condition.

This factor precludes the probability of avoiding substantial loads at the fixed-pivot axis even if the test airplane's weight is distributed so that the center of percussion is directly over the line of contact of the main-gear wheels. With the airplane in attitudes intended to simulate either "level-landing" or "three-point-landing" conditions, it is not possible to distribute the required mass so that the line containing the center of percussion and the fixed axis of rotation is normal to the landing reactions.

In the case of the Model PV-1, a 26,500-lb normal-gross weight airplane, calculations for drop tests in three-point attitude, with the center of percussion over the main-gear axles and the tail-gear axle a fixed center of rotation, indicate that allowable impact loads on the main gear could produce a tail-gear drag-load component in excess of 24,000 lb. An estimated value of angular velocity at the instant of peak impact load indicates an additional 3000-lb component, due to centrifugal force, giving instantaneously a magnitude of total drag load on the tail gear greater than that of the airplane gross weight.

In order to avoid the imposition of large dynamic loads at the pivot point, it has sometimes been a practice to provide elastic restraint of the pivot by use of shock cord. This technique may well prevent damage to the airplane structure but still may result in violation of requirements (*b*) and (*c*) relative to contact velocity and energy absorption, depending upon the degree of restraint afforded.

The ideal anchorage for the airplane in rotational drop tests is one whereby the axis of rotation may be placed on a horizontal line through the airplane center of gravity (cg).

The "free rotational" compound pendulum represents a system of this type, Fig. 2. If the pivot axis is allowed to move freely in frictionless horizontal guides that offer complete vertical restraint, no horizontal forces can be applied to the body during its free fall, and the center of gravity will drop vertically. During this period the only force existing at the pendulum support is that required to produce angular acceleration of the body about the instantaneous center of rotation, which falls with a vertical velocity the same as that of the center of gravity. Thus the line containing the instantaneous center of percussion (icp) and the instantaneous center of rotation passes horizontally through the center of gravity. If the instantaneous center of percussion is located directly over the stop, the stop force produces no impact at the moving pivot axis.

EQUILIBRIUM EQUATIONS

Several useful and interesting relationships may be obtained from the equilibrium equations of this system during its free-fall period. By referring to Fig. 2 it may be seen that the position, velocity, and acceleration of the center of gravity (cg) relative to the reference axes are described by

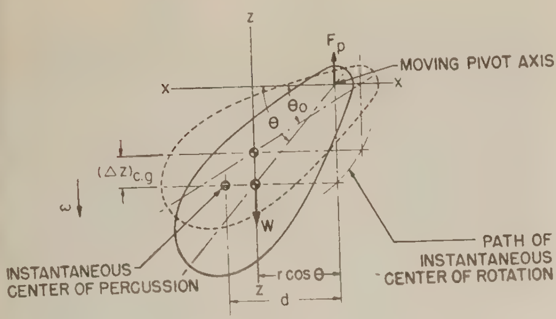


Fig. 2 "Free Rotational" Compound Pendulum

$$(z)_{cg} = r \sin \theta \qquad [1a]$$

$$(v_z)_{cg} = r\omega \cos \theta \qquad [1b]$$

$$(a_z)_{cg} = r\alpha \cos \theta - r\omega^2 \sin \theta \qquad [1c]$$

For translational equilibrium

$$\left(\frac{W}{g}\right)(a_z)_{cg} = W - F_p \qquad [2]$$

For rotational equilibrium

$$\left(\frac{W\rho^2}{g}\right)\alpha = F_p r \cos \theta \qquad [3]$$

From Equations [1c], [2], and [3],

$$(r^2 \cos^2 \theta + \rho^2)\,\alpha - (r^2 \sin \theta \cos \theta)\omega^2 - gr \cos \theta = 0 \qquad [4]$$

Equation [4] may be put in a more convenient form by using $\omega \dfrac{d\omega}{d\theta}$ for α and letting $\omega^2 = 2u$, resulting in

$$\frac{du}{d\theta} - \left(\frac{2r^2 \sin \theta \cos \theta}{r^2 \cos^2 \theta + \rho^2}\right) u = \frac{gr \cos \theta}{r^2 \cos^2 \theta + \rho^2} \qquad [5]$$

Solution of this equation yields

$$\omega = \sqrt{\frac{2\, gr\, (\sin \theta - \sin \theta_0)}{r^2 \cos^2 \theta + \rho^2}} \qquad [6]$$

where θ_0 is the position angle corresponding to $\omega = 0$.

The term, $r(\sin \theta - \sin \theta_0)$ is simply the change in height of the center of gravity, that is, $(\Delta z)_{cg}$.

By use of the concept of the instantaneous center of rotation the relationship between the drop height of the center of gravity and that of the instantaneous center of percussion is apparent

$$(\Delta z)_{cg} = \left(\frac{r \cos \theta}{d}\right)(\Delta z)_{icp} \qquad [7]$$

From basic mechanics $d = \dfrac{k^2}{r}$, where k is the radius of gyration about the axis of rotation, and r is the distance from the center of rotation to the center of gravity.

Here $d = \dfrac{r^2 \cos^2 \theta + \rho^2}{r \cos \theta}$, so that the vertical velocity of the center of percussion in terms of its drop height is

$$(v_z)_{icp} = d\omega$$

$$= d\sqrt{\frac{2g\,(\Delta z)_{icp}\, r \cos \theta}{(r^2 \cos^2 \theta + \rho^2)\, d}}$$

$$= \sqrt{2g\,(\Delta z)_{icp}} \qquad [8]$$

Since the vertical velocity attained by the instantaneous center of percussion in falling from a given drop height is the same as that attained by a body falling freely from the same height

$$(a_z)_{icp} = g$$

and, from Equation [2] and the counterpart to Equation [7]

$$F_p = W\left[1 - \frac{(a_z)_{cg}}{g}\right]$$

$$= W\left(1 - \frac{r \cos \theta}{d}\right) \qquad [9]$$

If the system is prevented from rotating by an external restraint acting along a vertical line through the instantaneous center of percussion, the static reaction at the pivot axis is

$$R_p = W\left(\frac{d - r \cos \theta}{d}\right)$$

$$= F_p \text{ for the same values of } \theta$$

Free Rotational Compound Pendulum System

The "free rotational" compound pendulum system ideally permits fulfillment of all of the basic rotational drop-test requirements for main gear if it is physically feasible to satisfy the following conditions:

1 The inclination of the shock strut relative to the impact load (assumed acting vertically) may be made to conform with the design condition.

2 With the airplane in the attitude consistent with item 1, a distribution of weight may be made such that, for the entire loaded airplane, (*a*) simultaneous values of $r \cos \theta$ and ρ satisfy a value of d predetermined from a choice of moving-pivot location; and (*b*) simultaneous values of $r \cos \theta$ and W satisfy the requirement for test-gear static reactions.

It is to be noted that the test-airplane gross weight and center-of-gravity position, as such, need not conform with those of the design condition; nor need the disposition of weights in the fuselage be based upon the actual distribution of fixed equipment or useful load. It is, of course, important to avoid local overloading of any part of the structure; the probability of this occurrence may readily be determined by analysis of the weight distribution with regard to anticipated local accelerations.

In practice, deviations from the theoretical performance exist; these may be summarized qualitatively as follows:

(*a*) *Eccentricity and Nonverticality of Impact Load.* Because of the inclination of the shock strut the fore-and-aft position of the axle relative to the airplane varies as the strut is compressed under load. Thus at only one position may the test-airplane instantaneous center of percussion be directly over the axle. The effect is usually very small, however, the maximum eccentricity from this source being of the order of 1 per cent of the value of d. Another contribution to eccentricity may result from change in effective mass. Prior to impact, the airplane may be considered as a relatively rigid body; after impact the semisprung mass of tire, wheel, axle, and piston behaves altogether differently from the rest of the airplane. Here, again, the magnitude of these effects is generally small since the percentage change in "rigid" mass is small.

Potentially of more importance than the horizontal eccentricity of axle relative to the instantaneous center of percussion is the drag-load component resulting from forward or aft motion of the tire on the impact surface. Even with a liberally greased contact surface it is not inconceivable that the direction of resultant load may vary by several degrees from the vertical, depend-

ing upon the coefficient of friction and the moment of inertia of the wheel-and-tire mass. This factor is of little practical consequence in so far as the magnitude of dynamic loading at the pivot axis is concerned; it is more significant that the nature of the effect is generally such as to be additive to the design drag-load component, simulation of which is intended by prior inclination of the shock-strut axis. More will be said about these effects in the discussion of the test results.

(*b*) *Nonrigidity of Airframe.* Potentially, a major reason for differences between the idealized and the actual behavior might appear to lie in the fact that the airplane is not a rigid body. Quantitatively, an analysis of the effects of elasticity is a formidable task (1, 2).[3] Moreover, it would appear to be a misdirection of effort to attempt such an analysis merely for purposes of of drop-testing

Qualitatively, it may be noted that the load-time history of a landing event is usually very long in comparison with the natural period of vibration of any of the significant weight items. Hence pure resonant effects may be discounted. However, both the release shock and the landing shock excite in the airplane components transient oscillations which may have a measurable effect, not merely at the pivot axis, but also on the test-gear dynamic loading. It might be expected that elastic effects would be most noticeable for critical metering pin-and-orifice combinations. Cases are known wherein a particular metering pin, prescribed as a result of landing-gear tests in a translational drop-test rig, was found not to give optimum performance in airplane drop tests. However, it should be noted that differences in overall test techniques, including setup parameters and instrumentation, may well produce differences of comparable magnitude.

Fig. 3 Tail Wheel Mounted in Horizontal Guide Block During Tests Simulating "Level Landing With Inclined Reactions"

(Thrust line was inclined 18 deg, nose down, and tail gear was swiveled 180 deg to avoid interference between fork and guide block.)

2 APPLICATION TO LOCKHEED PV PATROL BOMBER DROP-TESTS

Basic Test Arrangement

The moving tail-pivot test arrangement has been used in drop-tests of the main landing gear of PV patrol bombers. These tests were conducted for both "level-landing" and "three-point landing" conditions to determine energy-absorption and load-factor data throughout a range of drop heights.

As a matter of convenience, the tail-gear axle was chosen as the moving-pivot axis. The tail-gear strut was locked in a fully compressed position by steel straps attached to the torque-knee lugs. The tail wheel, less tire, was set on a steel track laid in the base of a horizontally slotted guide block, which was shimmed up from the floor to the height required to give the specified inclination of shock-strut axes for each landing condition. The installation is shown in Fig. 3.

The weight of the test airplane, consisting primarily of basic structure less outer wing panels, constituted about 20 per cent of the normal gross weight just prior to the start of the test setup. At this time it was desired to add the required mass in such a manner that the proper static reactions were obtained and that the center of percussion of the entire test airplane was placed over the line of contact of both main-gear wheels. From basic weight data the weight, location of the center of gravity, and pitching radius of gyration about the center of gravity were calculated for the empty test airplane. Weight and center-of-gravity location were checked by actual weighing.

As additional weights were placed in the airplane, a running tabulation of the weight-distribution parameters was made until, by successive approximations, the calculations indicated that the desired conditions had been satisfied. For overload conditions weights were added in equal increments both forward and

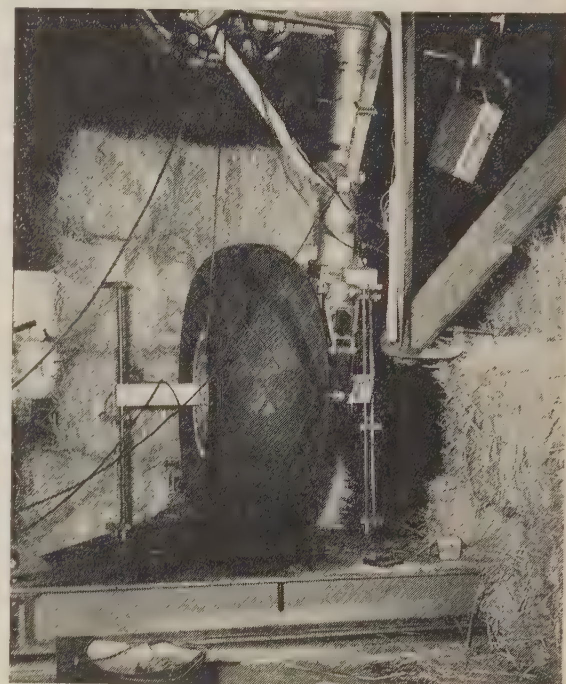


Fig. 4 PV-1 Main Landing Gear Drop-Test Setup

(View shows dynamic "weighing" platform, tire-deflection-measurement slide wires, and strain-gage installation on drag strut. Baled straw was used for protection of personnel and test airplane in event of tire or gear failure.)

aft of the center of gravity at distances equal to the computed radius of gyration about the center of gravity for the normal-

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

gross-weight condition. In this manner the location of the center of percussion was held constant, and the increases in main-gear static reactions were made proportional to the corresponding increases in gross weight of the test airplane.

In the loading of the test airplane no particular effort was made to obtain exact correspondence of fuselage load distribution with the actual airplane weight distribution, although provisions for large items of fixed equipment or useful load greatly influenced the operations. It was considered important, however, to reproduce power-plant weights and centers of gravity locations inasmuch as each engine-mount support structure was closely associated with the landing-gear supporting structure. As a matter of economy the wing outer panels were not installed, having been damaged in previously conducted static tests. However, it is considered that the over-all behavior of the landing gear and wing center section would have been somewhat more representative had it been feasible to effect a normal distribution of wing weight. Moreover, it would undoubtedly have been more rational from the standpoint of wing torsional moments to have reduced "power-plant" dead weight by the ratio of the distance from the instantaneous center of rotation to the instantaneous center of percussion and the distance from the instantaneous center of rotation to the power-plant center of gravity. Yet in all probability such a reduction would have involved overloading the fuselage forebody in the three-point-landing condition because of the necessity of having enough weight forward to locate the instantaneous center of percussion properly.

Drop-Test Instrumentation

Several months prior to the start of the tests it was recognized that a considerable amount of data would need to be taken in as short a period of time as possible because of the pressure of other testing schedules. Accordingly, special instrumentation was devised to permit rapid recording and reduction of the data.

In the PV-1 tests simultaneous and continuous recording was made of the following quantities as a function of time:

1 Load on bottom of tire.
2 Tire deflection.
3 Shock-strut piston travel.
4 Airplane travel from the instant of release.

These measurements were made concurrently on both left-hand and right-hand gear. In addition, the following quantities were recorded on one side only:

5 Landing-gear drag-strut load.
6 Longitudinal deflection of the lower end of the shock-strut outer cylinder relative to the plane of the wing main beam.

In tests of the PV-2 facilities were available for measurements of side-strut and drag-strut axial loads on both sides, as well as tail-pivot loads.

All "load" measurements were made by means of SR-4 strain gages installed on structural members and incorporated in 1000 cycle-per-sec alternating-current bridge circuits, the output voltages of which were amplified, rectified, and impressed on the galvanometer elements in a recording oscillograph. Tire and strut deflections, as well as airplane travel, were recorded in a basically similar manner through the use of specially constructed slide-wire potentiometers. The outer-cylinder deflections were detected by a bending-beam-type of strain-gage pickup.

The measurement of axial loads in tubular struts posed no new problems, since the techniques of dynamic strain measurement and associated calibrations to obtain corresponding loads are fairly well established. However, the determination of dynamic loads on the shock strut required a somewhat different approach, inasmuch as there were no locations on the gear itself where anticipated strains sufficiently large for accurate measurement could be considered as absolutely indicative of load on the member. Accordingly, it was decided to drop the wheels on platforms, each of which was supported by three tubular pedestals to which strain gages were attached. By virtue of identical load-strain sensitivities of the pedestals and of the manner of incorporating the gages in the bridge circuit, the load indication of each platform was made independent of the load center relative to the three supports and also of any local pedestal bending strains that might be developed from horizontal-load components.

A typical oscillogram is shown in Fig. 5. It is to be noted that traces labeled "inboard" and "outboard" deflections of each tire appear. It was found necessary to use two slide-wire potentiometers on each wheel because of cumulative rotation due to bending of wing center section, shock strut, and axle. Thus a means of knowing true deflections at the tire center line was afforded, as well as the magnitude of the rotation. Each oscillogram trace has its own calibration factor in terms of its physical counterpart, the calibration being facilitated by the static conditions prior to release of the airplane and immediately after the airplane came to rest.

Test-Data Interpretation

In the interpretation of the oscillograms it was necessary to correct the platform-load indications for the effects of semisprung mass of wheel and axle, in order to obtain magnitudes of load experienced by the shock-strut outer cylinder and supporting structure. In general, the peak loads on the tire occurred at a time when the shock-strut stroke-time trace indicated very little differential acceleration of sprung and semisprung airplane masses. Accordingly, the peak loads on the outer cylinder were obtainable quite accurately merely by taking the appropriate fraction of the platform-load indication. As a matter of convenience, this same fraction was usually applied to all values of platform load in the process of constructing load-stroke and load-mass travel graphs from the oscillograms. Inasmuch as the semisprung gear weight amounted to only a few per cent of the static platform reaction, the error involved was usually of no practical consequence.

The recording of dynamic loads and deflections on both main gears made possible a determination of over-all energy relationships. By comparing the potential energy of the airplane prior to release with the energy absorbed in both main-strut and tire units up to the time of minimum center-of-gravity height, an estimate of the minimum amount of energy absorbed by the airframe could be made. The term "minimum" is used, since the nominal center-of-gravity travel was used in computations, without regard to the incremental energy accruing from structural deformation.

In the earlier series of tests, analysis of the oscillograms showed that peak drag-strut loads were considerably greater than were anticipated. This condition was true for both three-point-landing and level-landing tests. Two readily apparent reasons for the differences were (*a*) that the peak load on the gear was developed earlier in the stroke than had been the case in previously conducted jig drop tests and (*b*) that the effects of structural deformation had not been included in the preliminary stress analysis. However, even with allowances for these factors, the drag-strut loads appeared to be too large. Qualitatively, the residual discrepancy was attributed to nonverticality of impact load due to inertial or frictional effects.

The PV-2 test instrumentation permitted an indirect evaluation of the degree of main-gear load eccentricity relative to the center of percussion. With the airplane in a "tail-high" attitude intended to simulate "level landing with inclined reactions," the line of action of the hoisting gear very nearly intersected the line

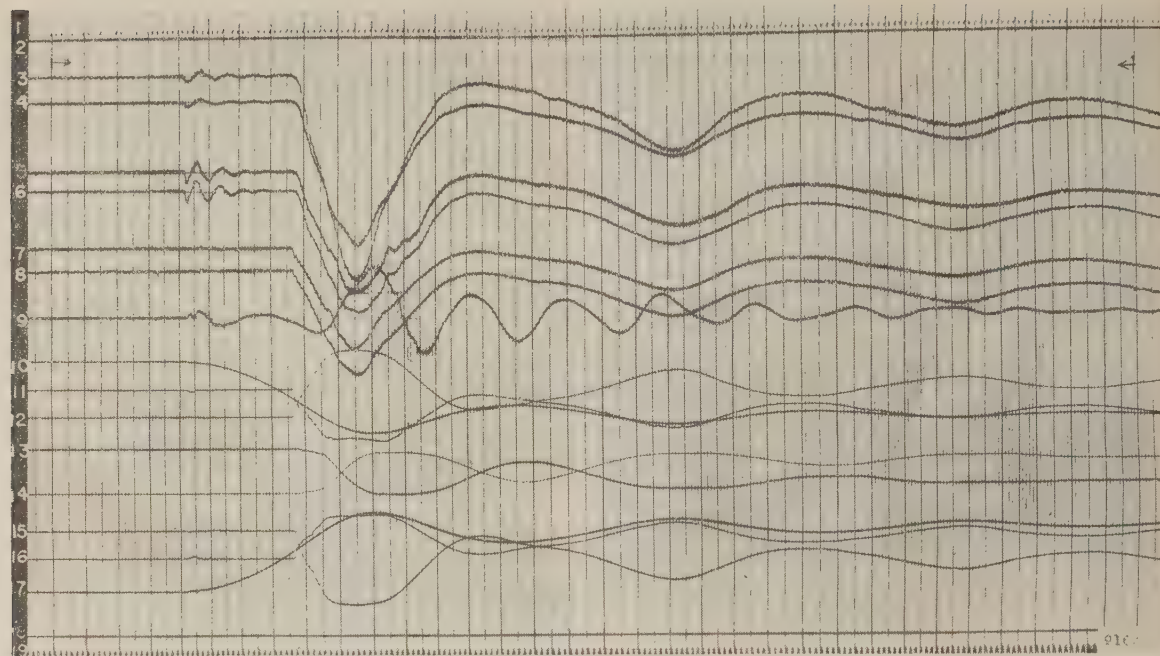


Fig. 5 Typical Oscillogram, With Numbered Traces Corresponding to Physical Quantities

(1 and 19, sixty-cycle timing waves; 2 and 18, reference line; 3, left drag-strut load; 4, right drag-strut load; 5, left side-strut load; 6, right side-strut load; 7, left-platform load; 8, right-platform load; 9, tail-gear load; 10, left-wing travel; 11, left tire, outboard deflection; 12, left tire, inboard deflection; 13, left strut stroke; 14, right strut stroke; 15, right tire, inboard deflection; 16, right tire, outboard deflection; 17, right-wing travel.)

of contact of main-landing-gear tires. Hence if the airplane behaved as a rigid body, and if the impact force acted vertically through the center of percussion, no change should have been expected in the tail-gear static reaction other than that occasioned by a slight change in airplane attitude.[3] Actually, a rather significant change in tail-gear reaction occurred, as can be seen by reference to the oscillograph trace in Fig. 5. Statically, prior to release, the bottom track of the horizontal guide applied an upward force on the tail wheel. Shortly after the instant of initial main-gear impact this force diminished to zero, remaining at zero for an instant while the tail wheel traveled freely about 1/16 in. before encountering the restraint of the upper guide. The peak downward load on the tail gear was attained shortly after the peak loads had been reached on the main gear. This behavior was typical for this series of tests, the magnitude of dynamic tail load increasing with main-gear drop height, and the time of maximum occurring at or shortly after maxima on the main gear. The persistence of tail-load oscillations after the main landing event was also characteristic.

An obvious deduction to be made from the tail-pivot loading is that substantial drag forces were developed as the semisprung gear elements moved forward due to compression of the inclined shock struts. Thus the diving moment induced by these drag forces was balanced by a downward dynamic load on the tail pivot. The validity of this conclusion is strengthened by the fact that the recorded peak loads on the landing-gear drag struts are in good agreement with values calculated by means of the instantaneous attitude of the airplane, the measured vertical loads on the gear, and values of drag load required to balance the simultaneous tail-pivot load. The data indicate that at the instant of peak vertical load the unwanted drag load may have been larger than 12 per cent of the vertical load. Thus the direction of impact load relative to the shock-strut axis was considerably more severe than that prescribed by the design condition.

[3] Reference to Equation [9], et seq.

Drop-tests for simulation of "three-point landing with vertical reactions" produced effects in the opposite direction, that is, the aft acceleration of wheel-and-tire masses demanded by the compression of the shock struts led to the development of peak resultant loads acting somewhat forward of vertically upward. Again, the net effects on the gear and its supporting structure were more severe than were intended.

Conclusions

The author feels that there are two definite conclusions to be drawn from the experiences accumulated both during the drop-test programs discussed herein and during more recently completed rotational-jig drop tests employing the same principles:

1 Drag-load errors arising from wheel inertia or from friction between tires and impact surfaces are not necessarily negligible in any type of drop-testing, whether simulation of the proper drag component is sought either by inclining the shock-strut axis or by inclining the contact surface. It is significant that, in general, the former expedient produces effects in a direction to aggravate loads in the drag-resisting structure, whereas the latter tends to alleviate them, resulting in an unconservative test unless deliberate compensation is afforded. While the control of these factors may be unduly costly, the measurement of their effects on the airplane structure appear to be justified as a standard procedure. The degree of refinement to be attained in instrumentation and in adjustment of test setup parameters is indicated by an over-all view of the problem. Drop-testing to simulate a particular "instantaneous" design condition is one thing; drop-testing to simulate actual critical landing conditions is quite another.

2 The "free-rotational" drop-test method described herein is believed to be an improvement over previously used rotational test methods. Moreover, from the standpoints of safety, accuracy, and ease of airplane control, it appears to offer advantages over translational airplane drop-testing in cases where simulation of dynamic landing loads on the entire airplane structure is not intended.

Acknowledgment

Acknowledgement is due a number of Lockheed engineers for their contributions, especially toward the initial work in 1943. Particular note should be made of the assistance given by G. E. Nichols and W. Bush in the testing and data reduction; of the helpful suggestions made by J. S. Edison, P. C. Mortenson, and R. Contini; of the development of the initial electrical instrumentation by R. E. Rawlins; of the guidance by V. S. Upton under whose direct supervision the work was carried out; and of the basic contributions by W. C. Hurty, whose interest and encouragement made possible the evolution of the new techniques described herein.

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Some Suggested Specifications for Thermal Ice-Prevention System for Aircraft

By L. A. RODERT,[1] INDIANAPOLIS, IND.

The general conclusions from research and development of the airplane thermal ice-prevention system are given and additional comments in discussion of present knowledge are made. Details of design are discussed whereby some faults of early applications of the thermal system may be avoided. Specifications for the thermal system are given as an aid to operators and others desiring to provide the most effective protection against the formation of ice on airplanes to be designed for commercial or military use.

Introduction

THE need for closer adherence to scheduled operations becomes increasingly urgent as the military and commercial use of the airplane is extended. Attention therefore is being given to the improvement of equipment and operational methods which will enable the airplane operator to achieve the regularity which is necessary to meet the demands of the public and competition.

The airplane is handicapped during inclement weather by obstructions to visibility and the formation of ice. The maintenance of schedules also is affected at some terminals by congestion of air traffic. While visibility and traffic problems may be solved by signal devices and operating procedures, the ice problem requires an intimate consideration of the airplane.

Research and development activities in many countries have been directed in search of a solution to the ice problem. These activities have, in general, concluded that the problem can be solved by the use of heat. Although it is fortunate that an answer has been found, heat is not entirely a fortunate solution because the internal parts of the airplane are adversely affected and complicated by its application. Engineering skills which can cope with the problems of flight and the internal mechanism therefore have to be directed to the task of applying the thermal system.

The locale, intensity, and distribution of heat and the mechanical, electrical, and aerodynamical details must be considered when the airplane ice-prevention specifications are prepared. The reports of the researchers have been limited in many instances to a discussion of the practicability and thermal requirements of the system, and rightly so, since these were the objectives. During the conduct of the tests, which involved the operation of several airplanes in natural icing conditions, practical design and operational information was also obtained. In the present paper, the practical considerations of design will be considered on a more equal plane with the theoretical objectives of the researches. It is hoped that this treatment of the knowledge will be helpful during the interim between the present and the later date when the manufacturers and operators will have had adequate experience with applied aircraft to establish the needed empiricisms.

Ice-Prevention Data

After the preliminary exploration in the United States and Germany, from which the practicability of the thermal system of ice prevention was established, many projects were undertaken in order to test the application to various parts of the airplane and to determine the intensity, distribution, and extent to which heating must be applied in order that ice might be alleviated. In this country, the application to all vulnerable parts of the aircraft has been recommended (1, 2, 3, 4, 5, 6, 7);[2] in Canada, the National Research Council has advanced and endorsed the use of heat on the propeller; and in Germany, the thermal system has been applied to lifting surfaces and windshields (8, 9). British researchers, collaborating with and actively participating in the work in this country (due to the difficulties imposed by hostilities over the British Isles), have also approved the thermal system as superior to other methods for all vulnerable parts of the airplane. These results all pertain to aircraft of conventional configuration, size, power loading, and power-plant installation.

Aerodynamic Lifting Surfaces. When heat is delivered to the leading edge 10 per cent region in sufficient quantity to raise the average leading-edge skin temperature 100 deg F above the ambient air temperature, while flying in dry air at 18,000 ft pressure altitude and at 60 per cent rated engine power, adequate heat is provided to prevent or remove ice formations from wings and empennage surfaces. This conclusion has been reached from tests with full-scale aircraft in flight in natural icing conditions. Conclusions based upon the data are considered to be applicable to conventional airfoils of from 4 to 20 ft chord whose pressure distribution is similar to that of the N.A.C.A. 230 airfoil series and whose thickness is from 6 to 18 per cent of the chord. The surfaces of the wings were normally clean but not aerodynamically smooth. The condition of the surfaces was more likely to be hydrophilic than the hydrophobic. The data were established from flights in several aircraft over a period of 5 years' time, over a wide geographical area, in various types of meteorological conditions, and with different observers (1, 2, 3, 5). Encounters with what may be called severe icing conditions have been successfully experienced.

The chordwise heating intensity over the leading edge 10 per cent chord region gave a temperature rise which was uniform within about 15 deg F in one airplane which had satisfactory protection in all conditions.

Ice does not form in the vicinity of the controls or control hinges unless, by peculiar rigging, a part of the control surface is made to protrude beyond the normal boundary layer of air which passes over the wing. Hinges, balances, or other protuberances which may occur from faulty manufacturing techniques will collect ice when the leading edge is heated sufficiently to keep ice off from the stationary part of the airfoil. When ice is allowed to refreeze on the regions rearward from the leading edge by reducing the heat below specified values, the formations have always occurred forward of the control-hinge regions.

Flight in clouds containing unusually large quantities of water at air temperatures below 32 F caused the wing leading edge to be

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[1] Research Engineer, Stewart-Warner Corporation, South Wind Division; Chairman, N.A.C.A. Subcommittee on De-Icing Problems.

Presented at the Aviation Division Meeting, Los Angeles, Calif., June 3–5, 1946, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

substantially wetted back to the 5 to 7 per cent chord point on the upper surface. Observations were not made directly on the lower surface, however, the wetted area is thought to be about the same as on the upper side. The area rearward from the 5 to 7 per cent chord region has been observed to be only partially wetted. No ice formed on areas rearward from about the chord point of maximum ordinate on wings of large chord, and, from this observation, it is concluded that no water is in contact with the skin in this region since the temperature of the skin was below the freezing point of water. This conclusion is not thought to be valid for airfoils of 4 ft chord or less.

In flights through cumulo-nimbus clouds of major development, large quantities of water in the form of snow, sleet, and rain, held aloft by strong up-currents have been encountered which caused the deposit of a slushlike formation along the stagnation-pressure region for short periods of time. These formations were as much as 2 in. wide and $^1/_2$ in. thick in some cases of extreme severity.

The delivery of heat to the aerodynamic lifting surfaces has been accomplished by the following means:

1 Exhaust tube within the wing leading edge (1).

2 Circulation of heated air within the leading edge and discharged from the airfoil interior in the vicinity of the trailing edge or control-surface hinge line (5).

3 Circulation of heated air within the leading edge and discharged from the airfoil interior into the boundary air near but to the rear of the transition point on the low-pressure side, high-pressure side, or both (2, 3).

The exhaust tube within the wing was discarded in favor of the air-circulating systems because the latter has been found to be more easily constructed and maintained and to be more readily employed on most airplanes.

Attempts to deliver heat to the wing leading edge by the spanwise flow of heated air along the wing leading edge have not been successful. All successful leading-edge designs have been adaptations of the Junkers leading heat-intercooler design (8) which employs a spanwise plenum, chordwise flow for heat exchange, and either a chordwise or spanwise flow for the outgoing air.

The wing-heating systems have been made without serious deleterious effects on the strength, construction, weight, or maintenance of the aerodynamic surfaces.

Windshield. The vision of the pilot and copilot of airplanes at speeds under 200 mph, indicated, has been maintained during flights in icing conditions by the transmission of about 1000 Btu per sq ft per hr through the outer surface of the windshield glass (2, 4). The prevention of ice on the windshield has also been accomplished on one airplane by the passage of a layer of heated air over the exterior from a slot at the forward edge (5). About 10,000 Btu per sq ft per hr was employed in the externally discharged air-heating system. However, the mechanical design was inferior, and it is believed that less heat would be required in a better arrangement.

Greater amounts of heat than that just indicated have resulted in excessive warmth in the pilot's cabin particularly in airplanes in which the pilot's face is near to the windshield. Heating the windshield to a limited degree increases the resistance of the glass to the impact of bird strikes according to tests conducted by the Civil Aeronautics Authority. Heating the windshield to a temperature above 140 F seriously weakens the glass and has a deleterious optical effect upon the plastic binder of safety glass.

Heat has been transmitted in the necessary quantities by the following:

1 Circulation of heated air between the panels of a double-glazed windshield.

2 Electric power through the use of embedded heating wires in the glass.

3 Circulation of heated air over the exterior surface.

4 Electric power through the use of external heating wires.

5 Electric power through the use of transparent conducting surfaces.

It is very important to remember that vision through the windshield is the important objective in windshield ice prevention.

The solution to this very important part of the ice problem must consider ice prevention on the outside, frost removal on the inside, resistance of the panel to the impact of flying birds, and the other more obvious construction and service requirements. Hope has long been held that a transparent and durable electrical conducting film for the coating of glass could be discovered whereby the windshield panels could be heated with electric power. Such a product has been discovered and developed by the engineers of the Pittsburgh Plate Glass Company. This discovery, which has been given the name of "Nesa," promises to solve the windshield icing problem in an optimum manner for all the factors involved wherever electric power is available in the required quantities.

The improvement of the optical qualities of the doubly glazed windshield has been attempted by the treatment of the glass surfaces for the reduction of the surface reflectivity. Coating the glass surfaces with magnesium fluoride by the evaporative process was not successful in one attempt. The use of the doubly glazed windshield has several disadvantages. Tests have shown the increased reflectivity of the greater number of surfaces to be detrimental to the pilot's vision, particularly at night in the landing maneuver. Water precipitates form on the inner side of the outer glass occasionally on some installations during taxi and take-off runs due to the circulation of high-humidity air past the cold outer panel. The fogging clears soon after the ducts are cleared of the moist air and the temperature of the air rises. The collection of foreign particles on the interior surfaces of the glass necessitates frequent clearing in dusty regions.

Propeller Blades. Ice can be removed or prevented from forming on propeller blades by the provision of heat to the blade leading-edge region or to the entire blade section. The application of the thermal system to the propeller blade has been accomplished by the use of electric power and by passing heated air through the interior of blades having an open passage from root to tip. The minimum quantity of heat required for a propeller blade has not been reliably established; however, sufficient data have been obtained whereby practical and reliable propeller protection can be achieved for blades currently in use. The results indicate that blades for a 12-ft-diam propeller will require from 1000 to 1200 watts of electric power per blade when turning at about 1100 rpm propeller speed and flying at about 180 mph indicated airplane speed (6). Other results indicate that blades of a 15-ft-diam propeller under about the same operating conditions require from 1000 to 1500 watts of electric power per blade (10).

Flight tests in natural and simulated icing conditions by independent researchers have shown that the thermal system for propellers may be operated intermittently with satisfactory protection for the propeller and with a great saving in electric power. When operated intermittently, the power is applied to a propeller for a short time during which the ice is removed and then the heating is interrupted. During the interruption, heating is applied to another propeller on which a small amount of ice has been permitted to form. Intermittent operation is thought to require a slight increase in blade heating in order to minimize the time required for ice removal and reduce the size of ice particles thrown off.

The electric heating of propeller blades has been accomplished by neoprene blade shoes in which an electric resistance of either conducting rubber or wire was constructed. The blade shoes have been about 60 inches long in some successful installations. Some blade shoes have given several hundred hours' service in varied conditions without excessive abrasion while others were severely damaged by one flight in snow and heavy rain.

The shoes were applied to the blade by adhesives. The attachment changed the contour of the blade profiles from 0.050 to 0.065 in. at the leading edge and about 0.010 in. at the 20 per cent chord point. The blade shoes covered about 25 per cent of the blade chord on the camber face and slightly less than 20 per cent on the thrust face on one installation which gave favorable results. The heating intensity of the electrically heated shoes was twice as great over the leading-edge part of the shoe as over the areas immediately to the rear of the leading edge on camber and thrust faces (6).

Power has been successfully supplied to the electrically heated blades by propeller-hub-type alternating-cuttent generators and through slip ring and brushes from an electric power source, such as the regularly installed generator.

Air for use in the air-heated hollow steel propellers was supplied from heat exchangers in one successful demonstration of the system. The energy employed in the propeller air-heated system was excessive and all ice was not prevented in every test, although it should be noted that by baffling the blade interior the heat would have been better employed and better protection would have been provided.

Miscellaneous Parts. Although the results of tests on wings, windshield, and propellers are indicative of how ice may be prevented on other vulnerable parts of the airplane, direct solutions to the miscellaneous problems have not been established. Individually, the collection of ice at points on the airplane other than on the wings, windshield, or propeller may not seriously affect the operation of the craft. Collectively, however, the miscellaneous problems, unless solved in part, may so handicap the airplane that the advantages of the improved protection to the wings, etc., will not reward the operator with the improvement in reliability which would otherwise be achieved.

The formation of ice on exposed frontal parts of the airplane such as the fuselage, engine cowls, and propeller spinner has been observed to be of sufficient size and shape to reduce the performance slightly, create a hazard when the formations become dislodged and fly rearward, and in the case of the cowls create an interference with the propeller blades which might interfere with blade-angle movement in an extreme case.

The formation of ice on radio antenna wires has broken the wires on most airplanes which have been operated in severe icing conditions unless steel cables were used in place of the copper wire which is customarily employed. Ice formations on radio antenna insulators have caused a reduction in the distance over which radio signals could be transmitted.

The formation of ice on the mast to which the static-pressure orifice for the airspeed and altimeter system is mounted causes an error in the meter readings and increases the drag of the airplane. In extreme cases the masts have been broken away from the airplane fuselage.

The formation of ice inside and on the edge of inlets to air ducts will reduce the inlet area and increase the air-pressure losses in the duct. Turning vanes in air scoops will collect ice and cease to be an advantage to the circulation of air. When ice dislodges from the opening and passes into the duct, due to flight into warmer air temperatures, the formations have seriously stopped the flow of air into a duct for a period until the ice has completely melted.

The formation of ice on exposed moving mechanisms has rendered such equipment inoperative. The exposed cable and pulley wheel or similar devices on propeller governors which are mounted on the front face of the engine, and the alternate air-intake door and operating arms for carburetor air intakes are examples of equipment which has been affected by ice formations. Slight accretions have been observed on control-surface hinges and balance weights but not of sufficient size to cause interference since these parts are ordinarily shielded by the forward part of the aerodynamic surface.

Engine Air-Induction System. The established practices of heating the engine air in this country and of heating the parts of the carburetor and induction system walls in the United Kingdom will prevent and remove the formations of ice in the engine air-induction system (7). Most airplane service installations of the air-heating system are unsatisfactory because of high maintenance requirements and the use of unsatisfactory mechanical arrangements for adjusting the carburetor air temperature. The air-temperature rise which is required for the removal or prevention of ice in the induction system is known by most airline and military operators and will not be discussed here.

Equipment

The application of the thermal ice-prevention system has necessitated the invention and development of new and unique mechanisms. The success of the detail design of the system is dependent upon the judicious choice and use of these devices. In the conduct of full-scale research, some experience was gained on some of the mechanical, thermal, and electrical equipment which must be used to implement the thermal system. These data are not a substitute for long service experience but may be helpful in the current phase of the development.

Sources of Heat. Heated air has been provided for wings, empennage, windshield, and propeller by exhaust gas-to-air heat exchangers in service and experimental airplanes, and by combustion heaters in experimental airplanes in this country. In Germany, the heat exchanger, combustion heater, and a mixture of exhaust gas and air have been employed as heat sources. Adequate heat capacity can be achieved with any of these types of heat source although several airplane systems have been designed which failed to meet the design requirements in this regard. Heat exchangers have been employed in service in large numbers which have given no cause for maintenance after thousands of hours flight time. Other heat exchangers have not given satisfactory service and have been replaced. Combustion heaters have been employed in service in sufficient volume and for a long enough time (for cabin-heating purpose) to indicate that this type of source can be made satisfactory. Minor service complaints have been experienced but none has been found which involves unsolvable problems.

Installations of the exhaust-gas exchanger have been made which provide adequate heating under all necessary operating conditions (11). Whereas the use of the combustion heater obviously involves an economic loss due to the use of gasoline for heat, the continuous passage of air and exhaust gas through the heat exchanger also involves a power loss which must be compensated for in increased power of the engines in order to maintain unaltered climb and speed performance (12).

Electric power has been provided by propeller-hub-type alternating-current generators for propeller ice prevention and the airplane's electric system. Service difficulties have been experienced in the use of the hub generators; however, the problems do not appear beyond solution.

Air-Distribution System

The distribution of air from the heat sources to the heated areas has been accomplished with aluminum, aluminum strength

alloys, mild steel, stainless steel, and glass-fabric tubes. The difficulties experienced with the duct systems were as follows:

1 Excessive air-pressure loss due to small size ducts, circuitous paths, internal protuberances, collapsed fabric flexible connectors, sharp corners, and improperly located or shaped guide vanes.

2 Excessive maintenance due to use of soft metals which were easily dented, use of thin metal which fluttered and failed in fatigue, duct connections which made repair of any part very difficult, use of insulation materials which deteriorated under vibration and wear, and control valves which failed elastically or in fatigue due to the use of small metal gages.

3 Off-gassing of pungent volatile materials when hot.

4 Creation of a fire hazard by the absorption of inflammable liquids in duct-insulating materials.

5 Air losses at duct joints and seams.

Controls. An electric control consisting of motor-actuated air valves, overheat turn-off switches, off-on signal lamps, on-off switches and a signal test circuit has been found satisfactory in one experimental system which employed heat exchangers (5). So-called safety devices and alarms failed more frequently than did the equipment against which failure the safety device was to provide.

Instrumentation. The definition of the properties of a thermal ice-prevention system has been accomplished with great difficulty. The use of the very elaborate and expensive measuring equipment and testing methods has not altered this situation (5). The measurement of pressures is handicapped by the turbulent condition usually existing in the duct system. The measurement of pressures is difficult because the instrument usually altered the condition being measured. With care, air flow rates can be measured to within plus or minus 10 per cent. Attempts to invent and develop ice-rate indicators have not resulted in success.

Temperatures can be measured with accuracies as given in Table 1.

TABLE 1 PROBABLE ACCURACY OF TEMPERATURE MEASUREMENT IN FLIGHT

Item	Range, deg F	Accuracy, deg F
Metal structure	0 to 300	±8
Metal structure	300 and above	±20
Ambiant air	—20 to +40	±1[a]
Duct air	0 to 300	±10
Leading-edge skin	0 to 200	±10
Engine exhaust gas	1000 to 1800	±25

[a] With greatest care and with correction for kinetic effect on a wetted thermometer bulb in a moving stream.

The stresses induced in a heated wing have been measured in tests from which apparently reliable results were obtained. The accuracy of this work is stated to be ±400 psi for the stress-change data involved in the heated wing (13).

The measurement of the temperature of the ambient air has been studied with care for the case of the thermometer bulb being located in a moist moving stream (14).

Meteorology

The measurement of the meteorological factors, i.e., water content, droplet size, and air temperature, has been undertaken by several capable engineers and the work continues, but the results thus far have not been encouraging. The best information on water content now available is that measured at mountain top weather bureau stations and that resulting from analytic studies which are based upon fundamental concepts of the physics of the air. Limited flight data substantiate data collected by ground stations and from analytical studies. The data are more satisfactory for stratus-cloud structures than for frontal or vertical developments. The great variations within clouds of vertical development and frontal phenomena make the measurement in these conditions particularly difficult.

The water content and droplet size in stratus clouds have been estimated to approach the values given in Table 2.

TABLE 2 ESTIMATED PHYSICAL DATA ON WATER IN ATMOSPHERE

Air temperature, deg F	Droplet size, $\mu\left(\frac{1}{1000}\text{ mm}\right)$	Water content, g per cu m
0	10	0.50
5	13	0.55
10	16	0.75
15	19	1.15
20	23	1.65
25	26	2.30
30	29	2.85
32	30	3.00

The values given in Table 2 are established by a faired line which has been drawn on a graphical presentation of weather-bureau data (15). The faired curve was drawn so as to give values which in all cases exceeded the data thus far observed and therefore is probably conservative. Our knowledge of the weather is obviously inadequate.

Operational Factors

Flight tests have been made by various observers over the United States from the Pacific Coast to the Great Lakes region, and over Canada from the Lakes to the Atlantic Coast and beyond. No variation in the nature of ice due to geography alone should be expected and none has been observed. Most severe and frequent icing conditions have been observed where a combination of large water content and subfreezing temperatures are encountered. The movement of Pacific air masses over the mountains along the Pacific Coast and the movement of frontal disturbances from the central mid-continent in an easterly and northerly direction cause severe icing conditions. Air mass or local weather does not usually produce dangerous icing conditions except over terminals where traffic is held aloft for long periods at fixed altitudes.

As has been reported by many investigators, formations can occur at air temperatures from 32 F to as low as —40 F or lower. Flight tests have been made in which icing has been encountered up to 18,000 ft pressure altitude. Reports have been recorded from flight operations over the North Atlantic of icing conditions which extend to about 27,000 ft pressure altitude. Icing at altitudes above 20,000 ft is found only in clouds of vertical development and therefore occurs over limited areas or along narrow lines.

Changing the velocity of an airplane will affect the formations of ice in accordance with the kinetic equation (14). Present transport aircraft do not fly sufficiently fast to prevent ice on the wing except at air temperatures near 32 F. The velocity of the outer blade radius sections of the propeller, however, is heated by the kinetic effect to a degree which will prevent the formation of ice under most conditions. In order to take advantage of the velocity effect in the design of propeller ice-prevention equipment, it is necessary to consider the thermal conductivity and other factors involved in the blade-heating problem (16).

The tests in icing conditions have involved sustained flight for over 2 hr in intermittent, moderate, and light icing conditions. Tests in severe conditions have been for short periods only because of the limited size of the regions in which such conditions have been found. Several experiences with severe conditions have been encountered for periods of about 20 min.

The performance of a system cannot be judged by an observation at any one instant because some ice accretions on the surface of an airplane wing which is heated less than that required for complete prevention will accumulate and intermittently be blown away, the bond having been broken by heating which was not adequate for complete prevention.

Operations have been made in icing conditions during engine warm-up, taxi, take-off, climb, cruise, let-down, landing, single engine cruise, and single engine[3] let-down with adequate protection provided. Frost has been removed from the leading-edge regions of the wing and empennage in about 15 min. This operation is usually accomplished in less time than that required for the warming-up of the engines.

The removal of ice from the wings, after accretions have been allowed to form, sometimes is accomplished satisfactorily, other times with some refreezing on the rearward and unheated portions of the wing.

The installation of the thermal system will result in a loss of climb and speed performance at all times when air is passed through a ram-actuated system. The internal and external losses will have almost a negligible effect on the performance of the airplane if care is used in providing the utmost in aerodynamic cleanliness. A poorly designed installation, on the other hand, can cause a loss of as much as 6 mph in speed at 55 per cent power cruise. The use of alcohol blade shoes on propellers has not resulted in a measurable loss in performance, although it is reasonable to believe that a slight loss in propulsive efficiency has been experienced when the design blade section ordinates are changed to the extent necessary in the heater installation. The blade shoes could be expected to have a greater effect on new and well-designed propellers.

The passage of heated air through a propeller and the discharge of air at the propeller tip cause a slight loss in propulsive efficiency. These losses will depend upon the shape, size, and location of the tip outlet, and the nature of the air passage.

Service experience with the heated wing indicates that the construction involved complicates the problems of repair of the leading-edge region. When damaged by hail, rough ground handling, or bird strikes, the replacement of the double-skin regions is particularly difficult. Normal means of corrosion prevention, i.e., alclad and dichromate primers, have retarded corrosion on three experimental airplanes after two years of operation. Metallurgical inspection of structural parts has not revealed any deterioration in the allowable strength, yield strength, elongation, or cyrstalline structure.

Discussion and Specifications

The results of all tests and experience in this country and abroad indicate that the thermal system can be applied to most commercial and military airplanes in a manner which will retain normal operational efficiency and dependability during conditions of icing.

In order for the thermal system to make a contribution in improved dependability, the airplane must be able to operate in icing clouds or freezing rain at any of the various operating conditions. It is not sufficient that the efficiency of ice prevention in the cruising altitude, for instance, be better than present equipment. An improvement in the protection during cruising would improve the safety, but unless the conditions of take-off, taxi, and landing are equally provided for, the operator will not be able to improve greatly his flight regularity.

The engineer therefore must give care in making the thermal ice-prevention installation in order to assure that protection has been provided for all normal altitudes and conditions of flight. A thorough knowledge of applied operations is a necessity in meeting this requirement successfully.

The evolution in design will alter the application of the thermal system, but it will not change the natural truth of ice being melted by heat. The fundamental seems to have been established.

[3] With a twin-engine airplane.

Preliminary to the statement of detail specifications, an expression (or at least understanding) of the general objectives should be formulated. The general specification should define the degree of perfection, and certain specializations which result from the use to which the airplane is to be put, the climate and region in which it will operate, the class of service, and the civil or military codes, compliance to which is mandatory. These general qualifications frequently are stated in part through other sections of the specification for an airplane but not in sufficiently complete detail to define the general requirements of the ice-prevention equipment. A specific statement on perfection is desirable because the results of tests and experience thus far observed show that the costs of perfect protection for an unlimited flight duration in the most severe conditions, and under conditions of a dual emergency (such as engine failure in addition to icing conditions), may penalize the usefulness of the aircraft to an impractical degree. Economically there is a limit to the perfection of ice protection in somewhat the same manner that there is a limit to the load factor or power loading which can be provided in an economically employed vehicle.

In addition to a statement on the degree of protection desired, the following general specifications are recommended:

0:1 The formation of ice on the airplane in any manner or location which shall reduce the operational efficiency, interfere with regularity of service, or reduce the safety of flight shall be prevented.

0:2 The installation of the equipment shall result in the minimum increase in weight and maintenance, and minimum decrease in speed and climb performance.

0:3 The safety, structural capacity, hazards from fire or gaseous poisoning of personnel, and versatility of the airplane shall not be adversely altered in any degree by the installation of the thermal system.

Aerodynamic Lifting Surfaces. Complete protection for the wings, empennage, and control surfaces will preserve the lift, drag, and pressure distribution of the basic airfoil, protect the operation of mechanisms, maintain flow through leading-edge duct inlets, protect the illuminating qualities of lighting equipment mounted thereon, and prevent damage to or interference with other parts of the airplane due to ice on the wing or to ice which has become dislodged from the wing and flies rearward.

The complete prevention of ice requires that the leading edge be maintained at a temperature above 32 F over all regions on which liquid water may reside. An ideal solution in the evaluation of the heat required might consist of determining how much water will strike the leading edge and over what area of an airfoil at the various water contents and droplet sizes given in Table 2. By assuming all or a specified portion of the water evaporated, the remainder to be abraded away by the airstream; by assuming a specified transition from laminar to turbulent flow as affected by the presence of water on the wing; by assuming the surface to be hydrophobic or hydrophilic, or to what degree of each; and by making other simplifying assumptions of aerodynamic nature, the required external heat-transfer coefficient at various chord points may be approximated. These calculations should be done at several air temperatures and therefore at several water-content conditions as given in Table 2 in order to establish the condition of maximum heat requirement and the optimum distribution for all conditions. In order for this method to be completely rigorous, the calculations should also be repeated for several combinations of wing and power loadings. This process does not seem practical because the data of Table 2 are not and have not been purported to be authoritative, nor have other weather data more authoritative been established. The process is of dubious accuracy because nothing is known about the relation of mass

transfer to abrasion losses or the distribution of this relation along the chord of the wing.

A rigorous solution to the external heating requirements on an airfoil will not be possible in the design of the thermal ice-prevention equipment until the weather conditions, i.e., water-content and droplet size variations with air temperature, have been established and recognized by regulating agencies and until the physical and thermal processes on the wing leading-edge exterior are better understood.

The degree of protection given a wing will depend upon the intensity of heating, the extent, and distribution of the heat, the pressure distribution of the airfoil, surface smoothness and wetability, and other factors. The specification should allow freedom in design which will permit the engineer to take advantage of all artifices and fortuitous circumstances. For example, in the propeller-wake region, it may be permissible and desirable to discharge the heated air from the leading-edge system into the boundary layer at about the 15 per cent chord point and thus greatly improve the thermal effectiveness of the system.

The experience gained thus far with the thermal system has shown that airplanes equipped with exhaust-heat exchangers will have been provided with protection for the lifting surfaces if sufficient heat is delivered to the 10 per cent chord leading-edge area to produce 100 deg F skin temperature rise at 18,000 ft altitude, 0 F ambient air, long-range cruise power, and in clean air. In the airplanes with which this experience was gained the heated air was passed through the after region of the wing after it had served its purpose in the leading-edge system. The effect of passing the air through the after section may not be large. It is obvious that some benefit will accrue, however, just as it is obvious that a greater heated chord coverage than 10 per cent will give a more efficient use of the heated air.

Practical limitations of wing leading-edge construction require an internal exchange system similar to the Junker's leading-edge intercooler design and therefore predicate within certain limits the type of distribution of heat which can be achieved with air or gas as the heating medium. While the distribution of heat may not be optimum as transmitted to the outer skin by the Junker's intercooler design, neither is the distribution as obtained particularly wasteful of heat. By tapering the passage of the air gap, a nearly uniform outer-skin temperature can be obtained. The conductivity of the skin will even out peaks or valleys which result from large or small transfers on the inner or outer surface at particular chord points.

A specification for the aerodynamic lifting surfaces such as the following is recommended:

1:1 A thermal system of ice protection shall be provided which shall preserve all of the aerodynamic, mechanical, and control characteristics of the wings, horizontal stabilizer, fin, and all control surfaces.

1:2 The degree of protection shall be equal to or better than that afforded by that quantity of heat which will raise the leading-edge 10 per cent heated region average temperature 100 deg F above static ambient air temperature for the condition of 60 per cent engine rated power at 18,000 ft pressure altitude and maximum load, less fuel required to climb to 18,000 ft pressure altitude.

1:3 All air-duct inlets, outlets, breathers, exposed mechanical devices, and illuminating fixtures on or in the surface shall be provided protection equivalent to that afforded the leading-edge areas as specified in 1:2.

1:4 The intermittent operation of the thermal system as presently designed for the aerodynamic lifting surfaces shall not be allowed except during emergency let-down and landing operations.

Windshield. The practical application of the heated windshield has been extensive and therefore the preparation of specifications for this part of the ice prevention does not involve as many questionable issues as other parts with which service experience is limited. The double-glazed windshield, which is heated by the passage of heated air between the panels, has provided protection but is subject to many faults. Freedom in design is desirable therefore in order that new and better solutions to the problem may be developed in service. The wording of the windshield specification should allow this freedom. Since the air-heated system is the principal method now available, the specification should also seek to stimulate the improvement of this method by manufacturers who desire to continue its use.

The external air film, infrared electric heating, electric conducting transparent film, and embedded resistance-wire electric heating all show promise and may replace the double-glazed arrangements.

Specifications for windshield ice prevention are recommended as follows:

2:1 Means shall be provided through the use of heat whereby the vision of the pilots through the cabin transparencies will not be reduced by the formation of ice, frost, or fog on inside or outside, in any maneuver in flight or on the ground.

2:11 The provision of adequate vision at all times must be achieved without the use of knock-out panels or the opening of windows.

2:2 The heating of the pilot's windshields shall not impair the comfort in the cabin of the airplane or cause irritation to the eyes, nose, or throats of personnel in the pilot's cabin.

2:3 The prevention of ice on the windshield should be accomplished without the resort to water repellents on the outer surface of the glass.

2:4 The operation of the thermal ice-prevention equipment shall not reduce the protection which is afforded by the windshield to the cabin occupants against bird impacts and similar hazards.

2:5 The construction of the heated windshield shall not seriously impede the cleaning of the windshield glass.

2:6 The heating of the windshield shall not cause an optical distortion of the pilot's view.

2:7 The heating of the windshield shall not cause an accelerated deterioration of the plastic binders of safety glass or of elastic or plastic frame seals of the panel in the airplane.

2:8 The protection against ice on the windshield should be equal to or better than that which is afforded when 1000 Btu of heat per sq ft per hr is passed through the outer surface of the transparency at an indicated airspeed of 150 mph.

Propeller Blades. Although the quantity of heat required for the protection of the propeller blades has not been evaluated by analysis, experimental data have been collected whereby adequate specifications can be prepared. As in the case of other equipment previously noted, freedom of design should be permitted because there are several ways in which the thermal system can be employed with apparently equal results. Ice on the blades of a constant-speed propeller reduces the efficiency of the propeller at all blade angles and also reduces the blade angle in accordance with the increase in drag of the blade section. Protection of propellers therefore should be as nearly perfect as possible.

The Army Air Forces Specification No. 29245, November 15, 1945, may well serve as a guide in the detail specifications of the blade-heating equipment. The specifications as follows are also recommended:

3:1 Means shall be provided whereby the propeller blades shall be heated in a manner and to a degree which shall prevent

the loss of propulsive efficiency or excessive propeller unbalance by ice formations or damaging shedding of ice fragments at all operating conditions in the air or on the ground.

3:11 The prevention of ice on a propeller which is feathered shall not be required.

3:2 The protection on the propeller blade should be equal to or better than that which is afforded by the continuous dissipation of 3.2 w per sq in. average over the leading-edge 20 per cent chord region and 75 per cent of the span measured from the hub face, or by the intermittent dissipation of 3.5 w per sq in. over the same area.

3:3 The construction of the blade-heating system shall provide for a heating intensity on the leading-edge one-third of the heated area which is double that on the rear two-thirds.

3:4 The propeller-blade heating installation or its operation shall not cause a reduction of the normal propeller efficiency.

3:5 The construction of electrically heated blade shoes shall allow for trimming of the tip end when this region has become abraded by wear.

3:6 The passage of heated air through a propeller blade shall not cause an apparent increase in the propeller noise level.

3:7 The operation of the propeller ice-prevention system during reverse pitch braking is not required.

Miscellaneous Parts. Specifications for the provision of protection on regions and parts, the formation of ice on which does not create at once a direct hazard, are recommended as follows:

4:1 The formation of ice on the front of the fuselage, engine nacelles, propellers, spinners, or hubs shall be prevented if ice formations on these parts, individually or collectively, cause a significant reduction in the performance of the airplane, or if the formation causes danger to, or malfunctioning of, the affected part or any other part of the airplane.

4:2 The formation of ice shall be prevented by thermal means on the fuselage, and other places not otherwise specified, from which dislodged ice formations may cause damage to other parts of the airplane.

4:3 Radio antennas and masts should be located so as to minimize the formation of ice thereon, and the strength of these parts should provide for carrying maximum ice formations without structural failure.

4:4 Provision shall be made for the intermittent removal of ice from all protuberances, antenna wires, masts, and extending mechanisms when such formations reduce the specified performance of the airplane or adversely alter the operational usefulness of the airplane.

Specifications for the protection of minor functional parts are recommended as follows:

4:5 Provision shall be made for the prevention of ice formations by thermal means on air speed total- and stati-pressure orifices, on the mast supporting such instruments, and on the surface of the airplane where flush-type static orifices are employed.

4:6 Wherever possible, air-intake scoops should be designed so as to eliminate the vulnerability of the scoop to ice formations, otherwise provision shall be made for the prevention of ice by thermal means from air-inlet scoop and guide vanes within air scoops.

4:7 Provision shall be made for shielding all exposed mechanisms not otherwise protected from the formation of ice, the formation on which would cause damage or malfunctioning to the mechanism.

Engine Air-Induction System. The protection against ice in the engine air-induction system is considered to be of first importance by pilots and operators. No other phase of the icing problem can affect the performance and safety as suddenly and dangerously as does ice in the carburetor and other induction parts. The provision of satisfactory protection has been handicapped because of the circuitous channels of procurement and divided or questioned responsibility. Protection for the induction system involves considerations of engine, carburetor, airplane design, and airplane operations. In the past, the engine and carburetor have frequently been designed without consideration of the ice problem. In other instances, false claims have been made which have misled the pilots until authentic operating data were available.

The only satisfactory solution to the induction icing problem is one in which no ice is allowed to form during all operating condition. The operation of the system should not adversely alter the performance of the airplane and should not limit the operation of the engine when ice is being prevented. Whereas these requirements are important in consideration of reciprocating engines, they are even more important when turbine type engines are employed.

Recommended specifications for the protection of the engine air-induction system are as follows:

5:1 Provision shall be made for preventing the accumulation of ice in the engine air-induction system through the use of thermal means.

5:11 The thermal system of ice prevention shall provide for the heating of the walls, exposed parts, and protuberances in the induction system,to, or above, the freezing temperature of water; or

5:12 The thermal system of ice prevention shall provide for the heating of the engine air to a temperature above the freezing point of water at all points along the path of air flow.

5:2 The use of the thermal system shall not limit the power rating or manifold pressure at which the engine may be operated when the system is in use.

5:3 The use of the thermal system shall not reduce the power of the eingine (because of a reduced ram pressure in the engine manifold) to an extent which will reduce the operational efficiency of the airplane.

5:4 The operation of the thermal system shall not require adjustment by the pilot of the airplane nor shall any attention from the crew be required by the equipment while it is in use.

5:41 Required thermal modulation, where not inherent in the system, shall be provided by automatic devices.

Miscellaneous Mechanical Equipment. Good mechanical design of aircraft is a production of good engineering administration, a willingness on the part of the manufacturers to allow adequate time for design and tests of new developments, and an accurate statement by the operator who buys the airplane of the anticipated requirements. Some detail mechanical specifications are desirable, however, several items of which follow:

6:1 A source of heat or power for each component of the thermal system shall be provided which will have adequate capacity for all normal and emergency operating conditions and for all meteorological conditions to be encountered.

6:11 Operating conditions in which the thermal system shall be required to function are engine warm-up, taxiing, take-off, climb, climb with one engine inoperative, cruise at all loading and power conditions, cruise with one engine inoperative, 400 fpm let-down with 45 per cent or more engine power, 400 fpm let-down with one engine inoperative, and landing.

6:2 Provision shall be made to protect the airplane structure and equipment, which will be in contact with the circulated heating medium, against corrosion from active ingredients contained therein.

6:3 In so far as the output of the heating source is not inherently modulating, provision shall be made to limit the maximum temperature of the circulating medium to 350 F.

6:4 Provision shall be made, in systems employing exhaust gas-air heat exchanger or exhaust gas-air mixing equipment, to close the hot-air duct if a fire occurs in the engine nacelle.

6:41 In event of fire in the engine nacelle, the hot-air duct shall be closed with a stainless-steel valve by automatic means.

6:5 Provision shall be made for easy access to the equipment of the thermal system, for easy inspection, and for the repair of the wing and empennage leading edges, ducting, and valves by normal aircraft sheet-metal shop practices and without excessive cost.

6:6 An indication of good duct design shall be when the pressure losses in the duct do not exceed those occuring in the heat exchanger, combustion heater, or wing leading-edge internal heat-transfer system.

6:7 Provision shall be made to prevent the absorption of liquids such as hydraulic fluid or oil into the thermal insulation of hot-air ducts.

6:8 Only such insulating materials may be employed on the ducts which are fireproof, contain no volatile oils, and which do not deteriorate when subject to aircraft vibration.

6:9 Flexible fabric connectors may not be used in the air duct system.

Acceptance Tests. The approval and acceptance tests of the equipment should be given careful consideration in the specifications. Although the only permanent proof of acceptability will be satisfactory protection in ice over a long period of service, preliminary tests on one airplane can serve to show approximately the thermal qualities and therefore the ice protection to be afforded. It should be noted, however, that the only practical time to change the thermal ice-prevention system is when it is still on the drawing board.

Recommended specifications for the acceptance tests and instrumentation of the thermal system are as follows:

7:1 The capacity of the thermal system to produce and distribute the required quantities of heat to the protected components shall be demonstrated prior to flight tests and preferably prior to the assembly and fabrication of the components into the airplane.

7:11 The capacity of the thermal system shall be determined from airflow rates and air-temperature measurements.

7:12 The flow rate to each component of the thermal system shall be within 5 per cent of the designed flow rate when the rate through the air inlet is in accordance with the design value.

7:13 The quantity of heat delivered to each component of the system shall be within 10 per cent of the design value.

7:2 The capacity of the thermal system to modulate the air temperature within specified limits shall be demonstrated in accordance with the performance demonstration as specified in 7:1.

7:3 The measurement of temperatures may be by thermocouples and manual or automatic recording potentiometers.

7:4 Installations employing exhaust-gas exchangers or gas-air mixing devices shall demonstrate that the back pressure on the engine is within allowable limits, that specified temperature modulation is achieved, and that the capacity of the heat source is adequate for all conditions as specified in 6:11.

7:50 A demonstration of the control system shall be made prior to flight wherein all normal and emergency contingencies which may be encountered will be simulated.

7:51 The control system shall cope with all operational contingencies with the safety of the airplane and personnel maintained.

7:60 The performance of the thermal system shall be demonstrated in flight in dry air and in stratus-type clouds.

7:61 Flight-performance tests shall demonstrate the normal and emergency operation of all controls, burners, heaters, valves, safety devices, and equipment of the thermal system.

7:62 Flight-performance tests shall demonstrate that the heat is properly distributed to the various components of the heated system.

7:621 The demonstration of heat distribution shall be made in dry air, and stratus clouds at kinetic temperature above 32 F, and in cloud conditions as obtainable at kinetic temperatures between 0 and 32 F.

7:70 Provision shall be made and procedures outlined in printed form whereby all components of the thermal ice-prevention system may be observed in operation and inspected when the airplane is at rest on the ground.

7:80 The thermal system should be equipped with a means of indicating in flight that heat is or is not being supplied to the wings, empennage, and propellers. The indicator shall be within the full view of the pilot's, copilot's, or flight engineer's station.

Conclusion

The use of the thermal ice-prevention system involves many aspects of heat transfer. The progress which has been made in establishing a solution to the problem has, in large part, been due to the interest which has been given to the application of the thermal system by engineers who have given special study to this branch of engineering.

The use of the thermal system, however, has not been different from most other engineering applications of heat in that the problem involves many considerations which require a knowledge of other branches of engineering. In our present case, the aerodynamics of the wings, the propulsion of the propeller, the metallurgy of the structure materials, the economics of operational aeronautics, and many other branches of knowledge, which are important fields of specialization, must be considered.

The specifications and comments enumerated above cannot be expected to be complete in all detail. They do call attention, however, to many features of the thermal system that might otherwise not have been considered by engineers who have not had the opportunity to follow closely the research and development work on this problem. Giving to the design engineer in the factory and the operations engineer in the air line the most complete statement of the experiences of the research work should enable us to obtain the best equipment, at the least cost, and in the shortest time.

Acknowledgment

In the use of all information in this paper, acknowledgment is given to the N.A.C.A., by which the data were established, to the Army Air Forces and the Bureau of Aeronautics which sponsored many of the projects, and to the air lines and manufacturers who have faithfully encouraged and supported the work on the ice problem.

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The Escher Wyss-AK Closed-Cycle Turbine, Its Actual Development and Future Prospects

By CURT KELLER,[1] ZURICH, SWITZERLAND

This paper is the first comprehensive presentation in this country of details of the gas-turbine process employing a closed cycle. In spite of grave handicaps entailed by the war, the development of this turbine proceeded from the first experimental installation in 1939, to the advanced designs which are now available for industrial application.

Introduction

UNLIKE combustion turbines little has so far been reported in the United States about the gas-turbine process employing a closed cycle. The first plant of this kind was completed in Switzerland during the summer of 1939, just before the outbreak of war, namely, an experimental installation of 2000-kw useful output, Fig. 1. The enforced seclusion of Switzerland during the last 6 years, and more especially the interruption of communications with America made it impossible to discuss this new AK-plant, developed by the Escher Wyss Engineering Works in Zurich according to proposals made by Ackeret[2] and Keller.

However, in spite of the many difficulties arising from the war, it nevertheless proved possible to try out the experimental plant until normal industrial operation was reached and to investigate fully all its components.

About a year ago Prof. H. Quiby (the successor of Professor Stodola at the Swiss Federal Institute of Technology in Zurich) carried out exhaustive official performance trials on the new plant. A report (8)[3] on these trials was published in June, 1945. These official trials represent the termination of the first phase of internal scientific development, and the results obtained have justified in every respect both the theoretical and practical expectations. Projects embodying such a closed-circuit plant for power generation or ship propulsion can now be realized, on the basis of the preliminary studies extending over a number of years, without unwarrantable technical risks being involved. The questions that arise in this connection and the constructional solutions which we consider suitable from the technical point of view will now be mainly dealt with.

Since 1939 a number of original publications have been made regarding the theoretical and physical basis of the AK-process, which are now also accessible to American engineers (several of these articles have already been printed in English). The Bibliography of the chief papers regarding this field, together with the short index at the end of this paper, are quite comprehensive.

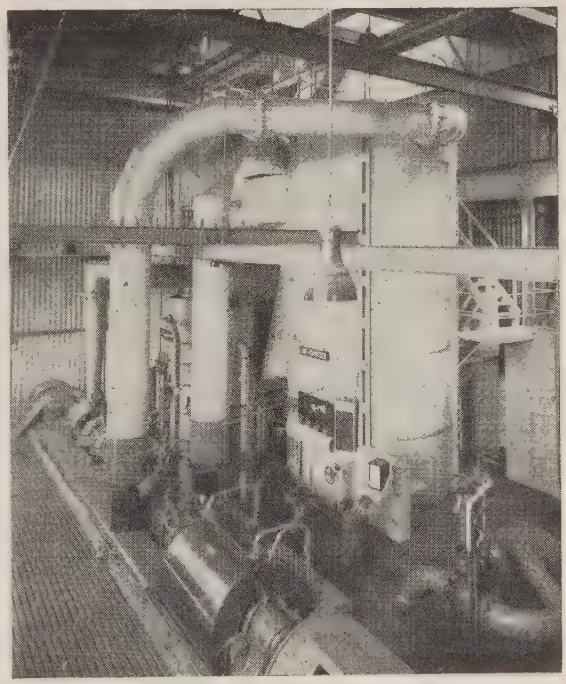


Fig. 1 General View of the First 2000-Kw Experimental Plant at Escher Wyss Works in Zurich, Switzerland

It will suffice therefore to give a brief summary of the chief characteristics of the new process as an introduction. Some information, but not all the data, was provided in a preliminary report which was read by R. T. Sawyer and discussed by S. A. Tucker during the June, 1945, meeting of the Society.

In recent months we have been favored by a number of visits from American engineers who came to the Escher Wyss Works, where information concerning the development work in connection with these AK-plants was given, and the trial installation itself was explained. As a consequence, our work in this field has become known among a wider circle of specialists.

The publications hitherto made with regard to the basis and development of the AK-process will be supplemented for the first time in the present paper by referring in greater detail to studies and designs, so far not published, for closed-circuit units which from the technical point of view are now practicable, and by remarks concerning the later uses and further possibilities of development. Since the research work and the whole field for such installations are very extensive, it is possible within the framework of the present paper to discuss only the principal ideas on the basis of examples.

[1] Director of Research, Escher Wyss Engineering Works.

[2] Professor of Aerodynamics and Flow Mechanics, Swiss Federal Institute of Technology, Zurich, Switzerland.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., November 26–29, 1945, of The American Society of Mechanical Engineers.

Summary of Theoretical and Physical Bases of Closed Circuit and Its Particular Features

As in the case of all other gas-turbine processes, it is our endeavor to approach as closely as possible the thermal efficiency of the ideal process for thermal prime movers, namely, the Carnot process, and furthermore with means which are simple in practice and at the same time economical. The closed gas cycle offers favorable possibilities for realizing this aim.

In the case of air or other technical gases the Carnot process proper, Fig. 2, is accompanied by numerous practical drawbacks, because for attaining the high initial temperatures T_1, which modern steels are capable of withstanding, very high pressure ratios P_1/P_2 from 200 to 300 have to be dealt with, especially during the adiabatic compression, which is difficult to realize in turbomachines.

The AK-process endeavors to bring about a cycle, the so-called double-isotherm cycle, which is thermodynamically equivalent to the Carnot cycle for gases. Fig. 3 illustrates this cycle in comparison with the Carnot cycle proper. The AK-process employs only small pressure ratios between A and B for isothermal compression of the air. Further adiabatic compression between B and C, according to Carnot, is intentionally dispensed with and replaced by an internal exchange of heat at constant final pressure of the preceding isothermal compression. The supply of heat from an external source by the combustion of fuel shall, as in the Carnot process, take place exclusively along the isotherm C-D. After this expansion with simultaneous development of power, the heat is reimparted to the gas along B-C by internal heat exchange in the circuit itself, whereby the pressure remains constant. This double-isotherm circuit has exactly the same theoretical efficiency as the Carnot cycle and is, consequently, dependent only upon the temperatures

$$\eta = \frac{T_1 - T_2}{T_1} \text{ (absolute temperatures)}$$

But the high pressure ratio, which is such a drawback, is entirely avoided; it can be reduced to a small fraction of the foregoing value, and one is not bound to a given relationship $P_{\max} : P_{\min}$ for attaining the maximum temperature T_1.

The hatched surfaces $ABCD$ in the entropy diagrams, Figs. 2 and 3, illustrate the work obtained for each unit weight of gas (air) for the Carnot circuit and for the double-isotherm circuit. This work is the difference between the turbine output $D'C'CD$ (= heat supplied from external source Q_{in}) and the compression work $A'B'BA$ (= heat carried away in cooling water Q_{out}). The relatively large proportion of compression work in the case of gas processes is clearly shown by the entropy diagram.

Complete isothermal compression and expansion can, of course, be only approximately realized in the machines that are available in practice. However, compression and expansion by stages, with intermediate cooling or heating (indicated by dotted lines on the AK-diagram) permit of this aim being put into practice to a considerable degree with only a few stages. The average temperatures T_1' for the supply and extraction of heat T_2' do not then deviate very considerably from the highest and lowest temperature of the ideal circuit. At the same time, the chief requirement of the thermal law is still largely fulfilled, namely, that the whole heat shall be imparted to the working medium at high temperature, and the remaining heat, after the development of power, be extracted at as low a temperature as possible.

An important new feature of the AK-process lies in the fact that the working medium is not taken from the atmosphere and afterward returned to the ambient air, as occurs in the case of all old reciprocating machines employing hot air, as well as in open-circuit gas turbines. Furthermore, since the AK-circuit is closed, it works with higher pressure of which the lowest pressure at the compressor inlet already lies considerably above atmospheric pressure. The chief characteristics, namely, a double-isotherm circuit with high-speed turbomachines and heat exchange, heat supply from an external source and raising of the

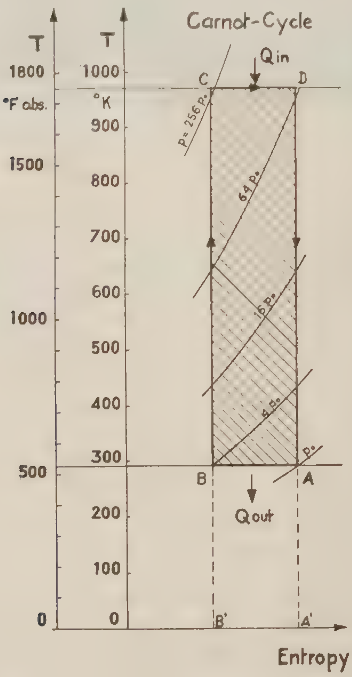


Fig. 2 Entropy Diagram of Carnot Process for Gases

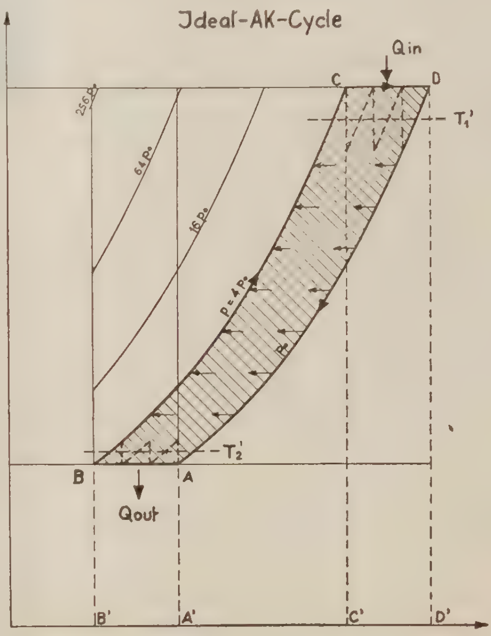


Fig. 3 Entropy Diagram of AK-Process With Complete Regeneration, Which Is Equivalent to Fig. 2

pressure level, have decisive consequences for the realization in practice, to which reference will now be made.

To illustrate the fundamental difference between the method of operation of the ordinary combustion turbine and the closed circuit of the AK-plant, simplified layouts for these different installations are compared with one another in Figs. 4 and 5. In

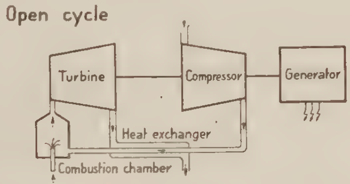


Fig. 4 Schematic Diagram of Open Circuit of Combustion Turbine With Heat Exchanger

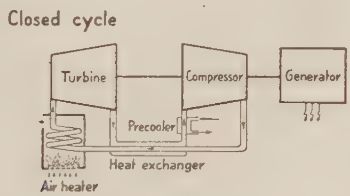


Fig. 5 Schematic Diagram of Closed Circuit of AK-Plant

place of the combustion chamber, the closed-circuit system has an air heater which is heated from an external source, corresponding to the boiler of a steam-turbine plant. In comparison with the open-combustion turbine which discharges its waste air to the open, the only additional component of the AK-plant is a precooler which cools the working medium after it has passed through the turbine and heat exchanger, to as low a temperature as possible before being reinhaled by the compressor. The principal features of our process and the characteristics resulting therefrom may be summarized as follows:

The Closed Pressure Circuit. Soiling of runner or impeller blading as well as of heat-exchanger surfaces by combustion residue or other foreign matter is completely eliminated, which provides a guarantee that the efficiency of the machines and the heat-transmission coefficients will remain unchanged. The working medium, which is always clean, permits the adoption of small cross sections for the elements of the heat exchangers.

The system is so supercharged that the pressure at the suction branch of the compressor already lies considerably above atmospheric pressure. As a consequence of this supercharging the dimensions of all parts of the plant, both machines and heat exchangers, are considerably reduced, on the one hand owing to the smaller specific volume of the working medium, and on the other hand because of the considerably increased heat-transmission coefficients at a higher pressure. Such supercharged operation permits of the unit outputs being increased almost without limit.

External Heating. By separating the circuit of the working medium from that of the combustion air it becomes possible to employ any kind of solid, liquid, or gaseous fuel, such as bituminous coal, lignite, oil, or gas.

Output Regulation. The output of the plant can be varied by changing the pressure level without altering thet emperatures and while maintaining the efficiency practically unaltered at all loads. By raising or lowering the pressure level (brought about by temporarily supplying or extracting working medium to or from the circuit), the plant can be suited to any desired part load, simply by changing the density of its working medium. In doing this the flow conditions in the machines, the pressure ratios, the velocities, and the angles of attack to the blades remain practically unchanged, the same being true of the internal efficiency. Hence the machines always operate at exactly the same point of their pressure-volume characteristic. Consequently, the efficiency of the plant is almost equally high at part loads as at full load, only the constant losses arising from bearing friction and heat radiation being proportionately more pronounced at part loads.

The fact that the temperatures remain unchanged at all loads is a particular advantage for the practical operation of such plant at high temperatures. No regulating valves, etc., are provided in the circuit proper, i.e., either on the machines or auxiliaries. The means for controlling the supply and withdrawal of air, which is stored in cold compressed-air accumulators, are united in a regulating set outside the circuit of the working medium and are traversed only by cold air.

The Use of Other Gases. Other working media than air can be adopted only in the case of a closed circuit with an external supply of heat. The employment of suitable light gases, such as helium, for example, opens up the possibility of increasing the output of the plant for the same dimensions, or of further raising its efficiency, as a consequence of the particular physical characteristics of such gases for special purposes (drive of transport means or aircraft).

Operating Cycles of the AK-Process

The simplest air turbines operate with direct expansion in the turbine itself and with two intermediate coolers in the compressor, as may be seen in Fig. 6. The most favorable pressure ratio

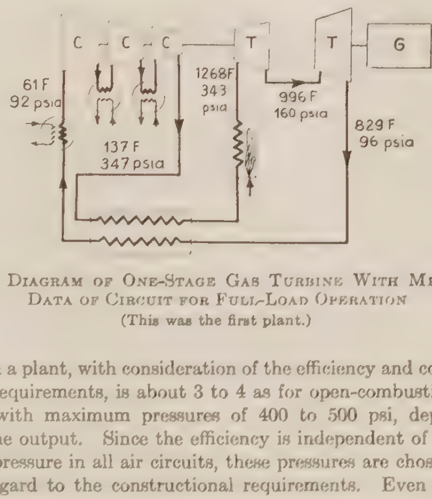


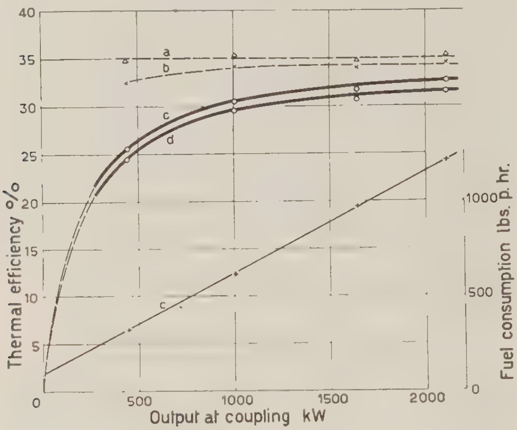
Fig. 6 Diagram of One-Stage Gas Turbine With Measured Data of Circuit for Full-Load Operation
(This was the first plant.)

for such a plant, with consideration of the efficiency and constructional requirements, is about 3 to 4 as for open-combustion turbines, with maximum pressures of 400 to 500 psi, depending upon the output. Since the efficiency is independent of the absolute pressure in all air circuits, these pressures are chosen only with regard to the constructional requirements. Even for the largest outputs no very high pressures are necessary, i.e., such as are encountered in steam turbines. For initial temperatures from 1200 to 1400 F, the discharge temperatures from the turbine are not higher than 750 to 950 F, so that no special alloyed steels are necessary for the heat exchangers. The example provided by the experimental plant proves that noteworthy thermal efficiencies can even now be attained with single-stage designs.

Fig. 7 shows the thermal efficiency which was measured during the official performance trials on the first experimental plant. In this connection it should be borne in mind that when this experimental plant was projected the chief intention was to try out the principle itself and test the co-ordination between the various components of the circuit. As a consequence of the in-

tended experimental work, the plant was laid out to insure easy accessibility to all parts thereof, so that measurements could be carried out without difficulty. Consequently, neither the arrangement nor the space requirements of the experimental plant may be taken as a criterion for later industrial installations which can be made much more compact.

The Sankey diagram, Fig. 8, shows the course of the heat quantities flowing through the various parts of the plant. Such a diagram applies in a similar manner to all single-stage installations. For each 1000 kw the AK-plant requires about 20 lb per sec working air in the circuit. In installations of larger output or to meet demands for higher efficiencies, we employ two-



a) Efficiency without insulating and bearing losses
b) Efficiency without bearing losses
c) Efficiency without auxiliaries
d) Net efficiency all losses included
e) Fuel consumption

FIG. 7 OVER-ALL THERMAL EFFICIENCY OF PLANT, AS DETERMINED BY OFFICIAL TESTS CARRIED OUT IN DECEMBER, 1944
(Curves indicate effect of relatively large bearing and insulation losses as well as power needed for auxiliaries)

stage expansion with intermediate heating which can be easily put into effect, as the following examples demonstrate:

This two-stage expansion represents a further and sufficient approach to the double-isotherm process which it has been endeavored to realize from the beginning with the AK-process (see Fig. 3). Intermediate heating, or double heating, as we refer to it in the case of the AK-plant, brings about a saving in fuel of 10 to 15 per cent. Considerations of a theoretical and practical nature lead, in the case of double heating, to the increased pressure ratio of about 10.

As a result of the greater pressure ratio and consequently the larger heat drop in the case of double heating, with due consideration of the losses in the circuit, not only a considerable increase in the thermal efficiency is realized, but at the same time a further reduction is accomplished in the dimensions of the machines and auxiliaries. This applies, in particular, to the heat exchanger because the weight of the circulating working air per unit of output is considerably smaller (approximately 12 lb per sec per 1000 kw), and because the greater pressure ratio allows of larger pressure losses without reducing the efficiency.

Fig. 10 shows the conditions in the case of direct expansion (A) and double heating (B), with due consideration of the losses. The circuit with double heating (B), can be thought of as realized in practice by placing two single circuits (A) next to one another. If in both instances one takes the same values for the limit temperatures T_1, T_2, with the same machine efficiencies η_T, η_K but with twice the pressure losses $\epsilon_B = 2\ \epsilon_A$ and temperature differences $\Delta\ t_B = 2\ \Delta\ t_A$ in the recuperator for the case (B), then the course of the efficiencies applies for (A) and (B) equally, if only the pressure ratio is taken as abscissa for (A) and for (B) the square of this ratio. The definition of ϵ is $\epsilon = \dfrac{\Delta p}{p_1}$, $\epsilon_2 = \dfrac{\Delta p}{p_2}$

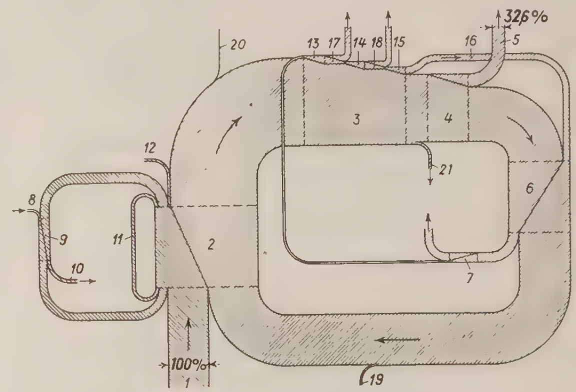


FIG. 8 SANKEY DIAGRAM FOR SINGLE-STAGE TEST UNIT, SHOWING QUANTITIES OF HEAT IN ACTION THROUGHOUT PLANT
(This is also typical for other installations of this kind.)

1 Calorific heat value of fuel
2 Air heater
3 High-pressure turbine
4 Low-pressure turbine
5 Output at turbine coupling
6 Heat exchanger
7 Precooler
8 Combustion air
9 Combustion-air preheater
10 Chimney
11 Circulating flue gases
12 Heat losses of air heater
13 Low-pressure compressor
14 Medium-pressure compressor
15 High-pressure compressor
16 High-pressure circuit air
17 Low-pressure intercooler
18 High-pressure intercooler
19 Heat losses on high-pressure side
20 Heat losses on low-pressure side
21 Mechanical losses

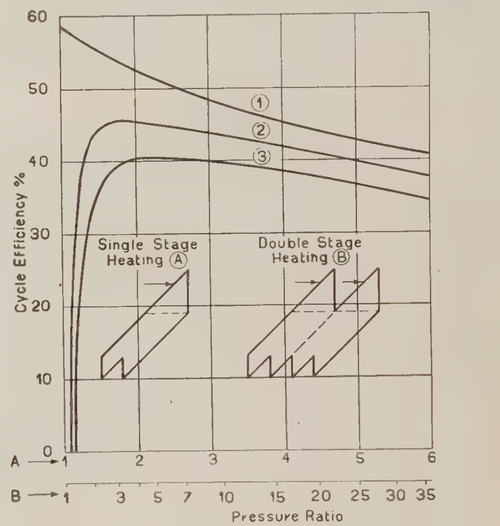


FIG. 9 CYCLE-EFFICIENCIES FOR SINGLE-STAGE AND TWO-STAGE INSTALLATIONS

Assumptions: $T_1 = 1200$ F, $T_2 = 80$ F $\eta_T = 91$ per cent, $\eta_c = 86$ per cent

Curve 2 valid for $\Delta t_A = 18$ F $\epsilon_A = 5$ per cent
or $\Delta t_B = 36$ F $\epsilon_B = 10$ per cent
Curve 3 valid for $\Delta t_A = 36$ F $\epsilon_A = 10$ per cent
or $\Delta t_B = 72$ F $\epsilon_B = 20$ per cent
Curve 1 valid for $\Delta t = 0$ $\epsilon = 0$
for both cases

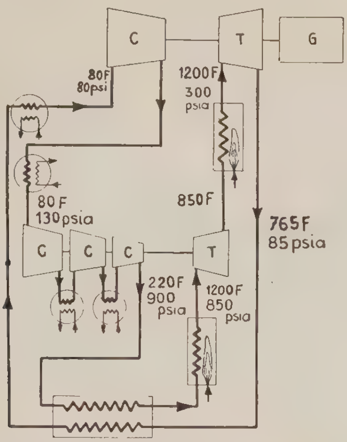


Fig. 10 Diagram of Two-Stage Plant With Intermediate Heating and Typical Data of Circuit

Δp = pressure drop on high-pressure side with pressure p_1, or low-pressure side, p_2.

With reference to Fig. 9, curves 2 and 3 have been plotted for two different pairs of values Δt, ϵ. Assuming the same difference in temperature, for example Δt = 36 deg F, and the same pressure losses ϵ = 10 per cent with direct expansion and double heating, the difference between curves 2 and 3 indicates the improvement by double heating. In the technically useful field of suitable pressure ratios, it amounts to 4 per cent absolute.

As already pointed out, the quantity of air in circulation becomes smaller owing to the greater pressure drop for double heating, so that the heat-exchanging surfaces are likewise reduced. If in the case of (*B*) one intentionally renounces to a certain extent an improvement in efficiency by allowing greater temperature and pressure losses as a consequence of increased velocities, the conditions will be those of the field between 2 and 3. However, in this way the surfaces of the heat exchangers as well as the other dimensions can be reduced considerably.

Installations with double heating operate according to the schematic diagram, Fig. 10. The corresponding Sankey diagram, Fig. 11, shows the reduced quantities of heat which are conveyed in comparison to Fig. 8. In actual installations, maximum pressures of 400 to 600 psia with back pressures of 40 to 60 psia will be adopted for small outputs, whereas for larger outputs above 10,000 kw, the pressure is raised to a maximum of 850 psia with about 85 psia back pressure. In this case also, the heat drops are so distributed that the heat exchanger can be built without requiring the use of special alloyed steels. The power stations and marine plants subsequently referred to operate according to this circuit.

Examples of Typical Components of Closed-Circuit Plants

Although the duty of the various machines and apparatus remains fundamentally the same in all installations, various designs nevertheless result which are dependent upon the different outputs required, the uses to which the installations will be put, the efficiencies and the available fuels. The closed cycle, operating at a pressure above atmospheric, leads to conditions of construction which deviate considerably from other kinds of gas and steam turbines. The consistent application of knowledge gathered from the laws of flow, from up-to-date aerodynamics, and also concerning the improved properties of metals subjected to high temperatures, coupled with proper harmonizing of the various components, has in recent years led to ever-increasing simplicity for our designs. They deviate in many respects from the layouts usually encountered for turbomachines. An AK-plant is not simply an assembly of known components; the arrangement of the various parts in relation to one another and the course of the working medium through the whole installation are the subject of careful study, whereby due attention has been paid to heat expansion. Only in this way can the pressure losses in the pipings and the other secondary losses be reduced to an admissible measure.

In this connection it has proved advantageous that all parts of an AK-plant such as the turbines and compressors, as well as the heat exchangers and air heaters, form part of the actual manufacturing program of Escher Wyss, which specializes in the construction of turbomachines of all kinds. Thus all the components have been developed and built in the company's own works, the experience gathered in various fields being duly coordinated and proper use made of the latest research results obtained in the company's own hydraulic and caloric laboratories.

It should also be borne in mind that AK-installations are quite suitable for standardization. Thus for example, the whole range of stationary installations from 3000 to 50,000 kw can be dealt with by a few types of machines and auxiliaries. The component parts of the heat exchangers themselves as well as the regulating means can, for installations of these various sizes, be put together in suitable combinations. This standardization which has been attempted but never realized in the case of steam turbines will have a favorable influence on the price calculations.

Turbines and Compressors. Since the specific volume is re-

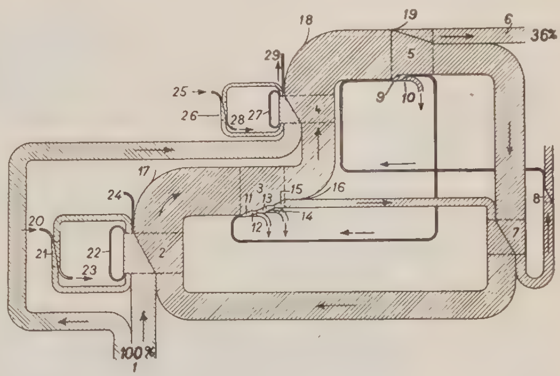


Fig. 11 Sankey Diagram for Two-Stage Plant Similar to Fig. 8

1 Calorific heat value of fuel
2 High-pressure air heater
3 High-pressure turbine
4 Low-pressure air heater
5 Low-pressure turbine
6 Output at turbine coupling
7 Heat exchanger or recuperator
8 Precooler
9 Low-pressure compressor
10 Low-pressure intercooler
11 Medium-pressure compressor I
12 Medium-pressure intercooler
13 Medium-pressure compressor II
14 High-pressure intercooler
15 High-pressure compressor
16 Mechanical losses of high-pressure set
17 Heat losses on high-pressure side
18 Heat losses on medium-pressure side
19 Mechanical losses of low-pressure set
20 Combustion air of high-pressure air heater
21 Combustion-air preheater of high-pressure air heater
22 Circulating flue gases of high-pressure air heater
23 Chimney of high-pressure air heater
24 Heat losses of high-pressure air heater
25 Combustion air of low-pressure air heater
26 Combustion-air preheater of low-pressure air heater
27 Circulating flue gases of low-pressure air heater
28 Chimney of low-pressure heater
29 Heat losses of low-pressure air heater

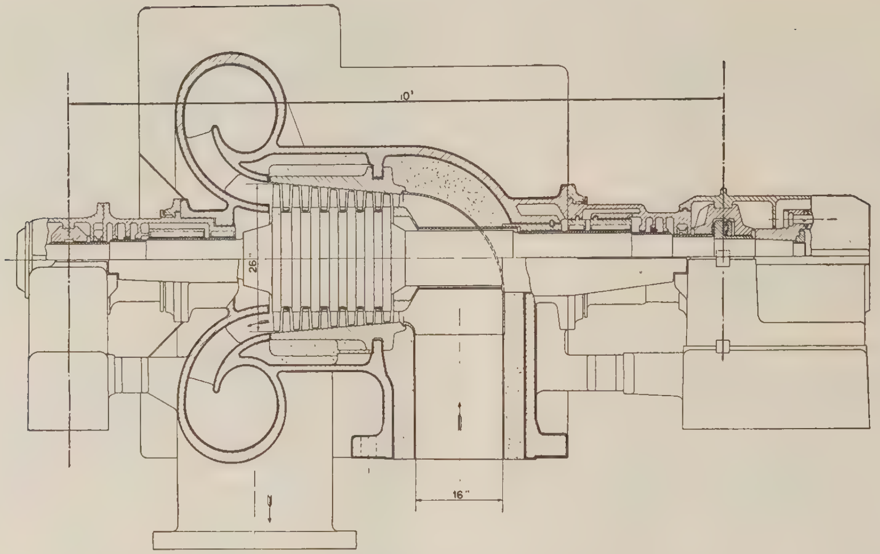


Fig. 12 Cross Section Through HP-Turbine of 25,000-Kw Set
(The effective output of this turbine is 28,000 kw.)

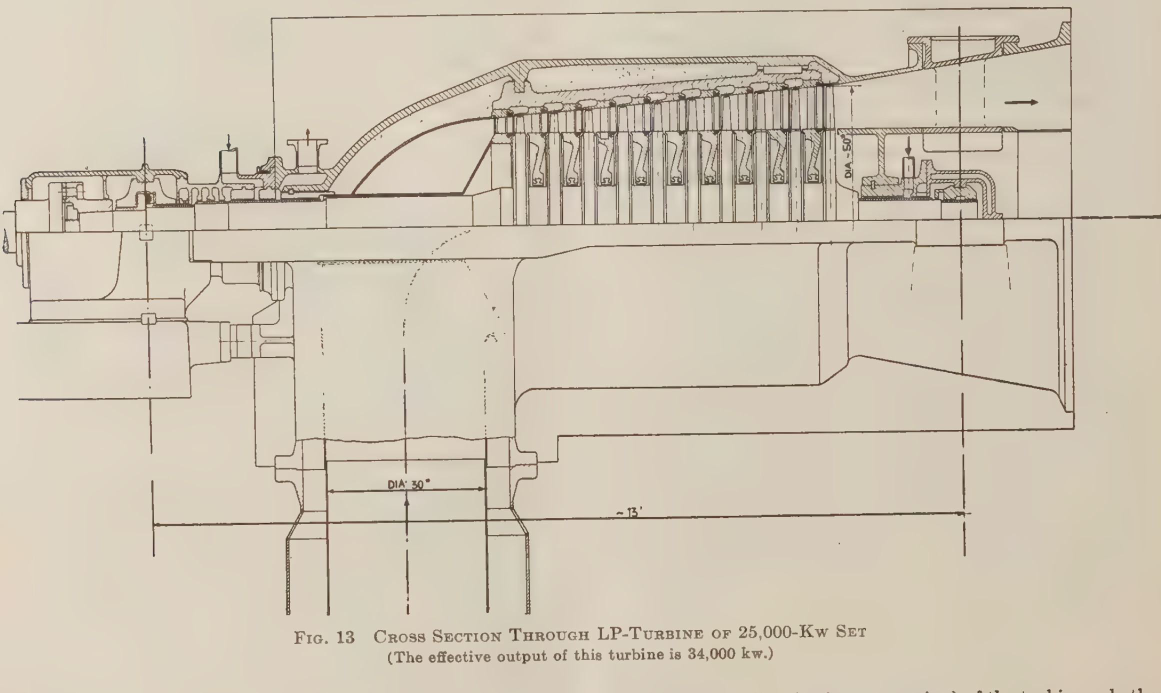


Fig. 13 Cross Section Through LP-Turbine of 25,000-Kw Set
(The effective output of this turbine is 34,000 kw.)

duced owing to the raised working pressure of the closed circuit, the machines are astonishingly small when compared to open-circuit combustion turbines. In addition, the heat drop that has to be dealt with is much less than in the case of steam-turbine plants, so that the closed-circuit turbine has not many stages.

For informatory purposes the main dimensions of the rotors for standard AK-plants of various outputs have been indicated in Table 1. These dimensions may vary slightly according to the speed and blading (action or reaction) of the turbine, whether an axial or radial compressor and of course also in accordance with the pressure that is employed. For the greater part, however, the sizes of the machines will not prove very different from the figures indicated in Table 1. The fact that no regulating valves or stop valves are fitted to the machines leads to favorable conditions for the construction of such hot-air turbines.

This permits the adoption of constructional forms for the turbines which differ considerably from those usually employed for

TABLE 1 APPROXIMATE FIGURES FOR MACHINE OF CLOSED-CYCLE POWER PLANTS OF DIFFERENT OUTPUTS (60 CYCLES PER SEC)

Net output, kw	6000	12000	25000	50000	100000
Maximum pressure, psi	600	600	850	850	850
Maximum diameter of:					
High-pressure turbine, in	17	19	26	35	47
Low-pressure turbine, in	31	39	50	70	87
Maximum diameter of axial high-pressure compressor, in	10	12	16	20	27
Maximum diameter of axial low-pressure compressor, in	21	30	33	53	65

steam turbines, Figs. 12 and 13. Special attention has to be paid to the inlet and outlet losses which, as a result of the small pressure and temperature drops that are utilized, play a relatively important part. The absence of regulating devices permits of the turbine being situated, literally speaking, immediately in the piping which conducts the working medium, an arrangement with which one is familiar in the construction of hydraulic machines.

It is possible, for example, to pass the working medium through an inlet branch supply pipe to the first runner wheel without employing annular channels. The outlet can be made symmetrical and a considerable part of the kinetic energy can be recovered, for instance in spiral outlet casings as in hydraulic turbines or pumps. Such forms are also used for the axial-flow compressors. The small turbines also permit the adoption of double casings which are designed according to the same principle as the hot-air piping previously referred to. Between the thin internal shell and the external casing there is a layer of insulating material; the internal casing serves only for conducting the hot current, while the cold external casing takes up the pressure. The whole guide apparatus is fixed to the casing at a point where temperatures are low. In this manner the external casing is, even for high inlet temperatures, subject at the most only to the discharge temperature of the last stage (approximately 925 F). Thus although the turbine-inlet temperature is much higher, heat-resisting steel need not be adopted for the casing but only for the small internal parts.

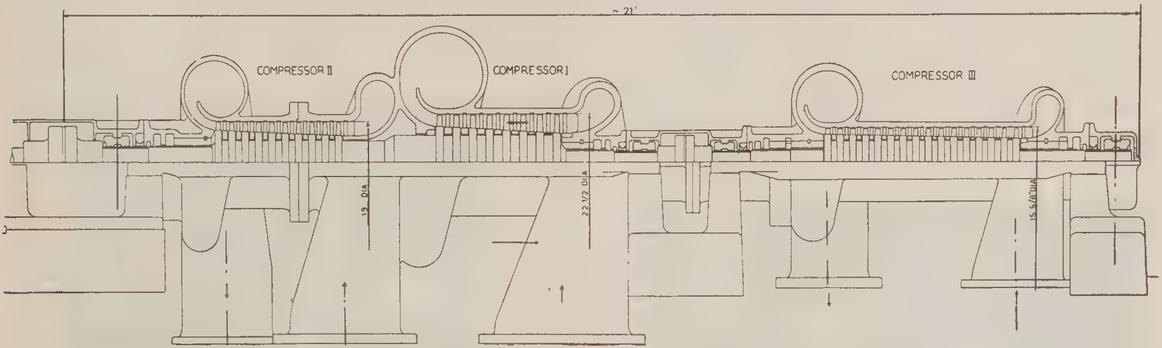


Fig. 14 Cross Section Through Axial-Flow Compressor for 25,000-Kw Set
(n = 6000 rpm.)

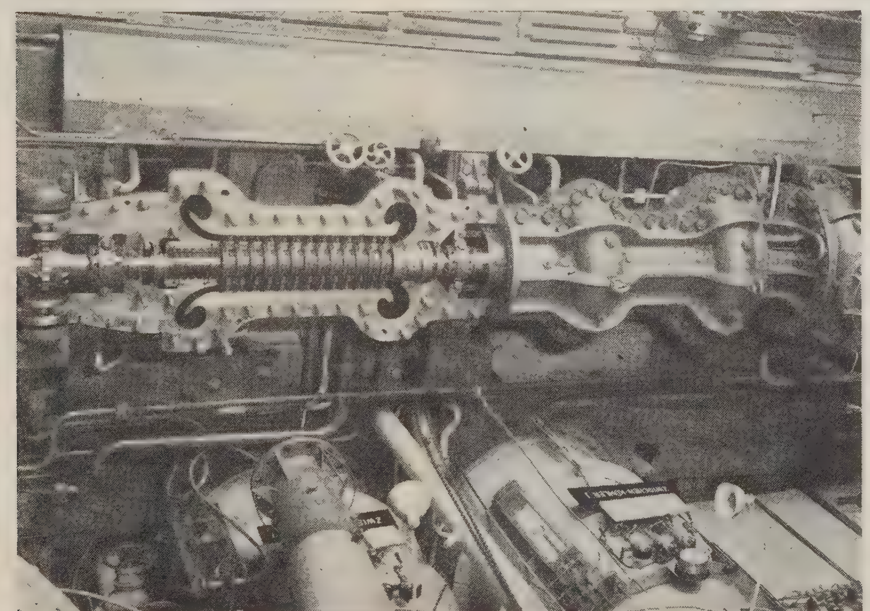


Fig. 15 View of Axial-Flow Compressor With Streamlined Inlet and Outlet Casing
(AK-Test Plant.)

Similar constructional principles, dictated by the latest knowledge in the field of flow techniques, apply also for the compressor and for the intermediate coolers forming part of the latter, Fig. 14.

Since relatively large quantities of air but small increases in pressure have to be dealt with, up-to-date axial-flow turbocompressors of the multistage type are particularly suitable for such duty. The high speeds needed for this type of compressor lie within limits which offer good constructional conditions also for the driving turbine. As a result of these high speeds the compressor set is of small dimensions even when dealing with the largest volumes. It has been possible to raise the efficiency of the bladings above the values attainable with radial compressors. This is again the outcome of systematic research in this particular field, Fig. 15.

As turbine and compressor always and at all loads work at the same operating point, high-quality blading need only be developed for this one point without compromises. For these conditions the course of the pressure-volume characteristic for part loads need not be considered. For the same reasons, for instance, in the case of small outputs of other gases, radial compressors can also be adopted thus leading to fewer stages.

Figs. 16 and 17 illustrate typical blading for AK-turbines and AK-compressors having stage efficiencies of more than 90 per cent. As a consequence of the raised pressure the Reynolds numbers of the machine bladings are of a considerably higher order, so that the percentage of friction losses becomes smaller. This holds good only for smooth and clean surfaces. Tests in our laboratories on a full-sized axial compressor inhaling ambient air, containing only usual workshop impurities, have proved that the blade efficiency dropped from 86 to 83 per cent during 12 hours continuous operation.

The rotating shafts of the machines are sealed from the ambient air by means of labyrinth glands or a combined system of labyrinth glands with liquid sealing, depending upon the size of the plant and the kind of gas employed. Good sealing is necessary in consideration of the losses, especially at smaller outputs or when employing special gases.

The glands illustrated in Fig. 18 have proved their merits in the case of the experimental plant. Sealing air extracted from the circuit is passed through a pipe to the labyrinth chambers. The pressure of this sealing air is at all loads always somewhat higher than the pressure inside the glands, corresponding to the point where it is bled from the circuit. In this way it becomes impossible for hot air to escape. From the point where sealing air is introduced another part branches off toward the exterior of the gland and flows into a collecting space which is connected to a point in the circuit where a somewhat lower pressure pre-

Fig. 16 Runner Wheel of HP-Turbine 12,000-Kw Set (Diameter 32 in. approximately.)

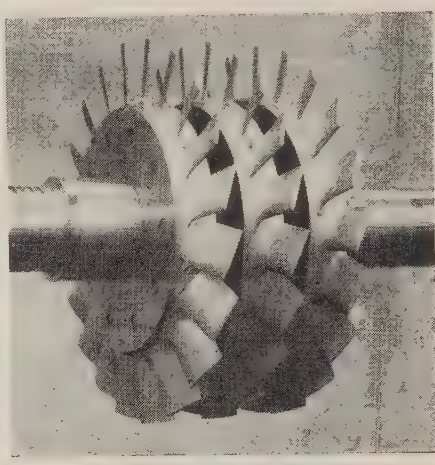


Fig. 17 Impeller of Modern Escher Wyss Axial-Flow Compressor

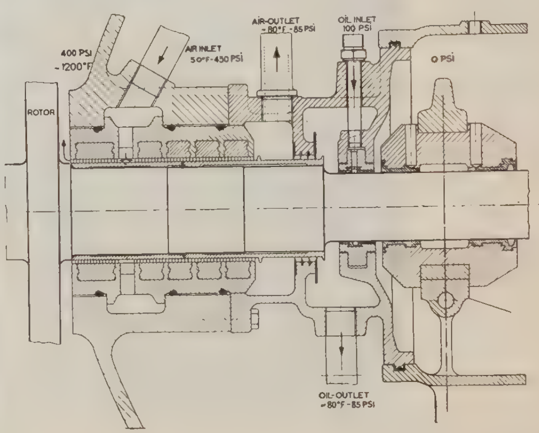


Fig. 18 Combined Labyrinth-Liquid Sealing for High-Temperature Turbines

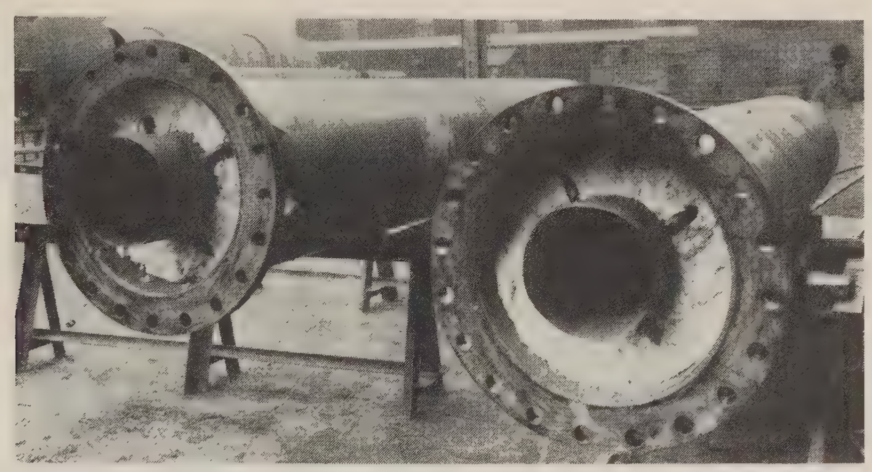


Fig. 19 Double-Shell Piping for High-Pressure Hot Air

vails. Outside these air-sealed labyrinth glands there is a ring through which oil under pressure is supplied to the shaft. This pressure oil prevents the escape of any air from the circuit. On either side of the pressure oil ring means are provided for leading the oil to a reservoir, which is under a suitable pressure above atmospheric, since it is connected to the interior of the circuit. The oil-sealing ring can also be combined with the bearing itself. By suitable connections of the sealing air pipes to the circuit it is possible to insure that at all loads, i.e., at all pressure levels, the sealing pressures will likewise rise and fall automatically, and that the direction of the current will remain the same; this without regulating valves of any kind having to be interconnected.

In practice the foregoing is very important for insuring the safety of the plant. The measures described have proved fully satisfactory in trial operation under the most exacting conditions. The sealing air, introduced to the end sections of the hot turbine shafts, is simultaneously utilized in an advantageous manner for cooling these parts, so that the bearing sections remain quite cold.

Hot-Air Piping. In order to reduce in so far as possible the quantity of metal capable of withstanding high temperatures, the hot-air piping has been made with double walls according to the same principle as for the turbines, Fig. 19. The design comprises a thin-walled internal tube of a heat-resisting material serving only for conducting the stream of gas. By means of openings this tube is relieved from the pressure in the heat-insulating space which surrounds it. The heat-insulating space in its turn is enclosed by a thicker-walled pipe of standard material which can easily take up the pressure of the working medium, since it is protected by the insulating material and therefore it is not under high temperature. The necessary measures are, of course, taken to prevent insulating material from gaining access to the tube. With this design much high-quality and expensive steel can be saved.

Air Heater. In the aerodynamic circuit the air heater plays a similar part to that of the steam boiler in a steam-turbine plant. The heat of the fuel is imparted indirectly to the working medium by heat-transmission surfaces, the combustion gases being kept away entirely from the machines.

Since no feedwater is employed, the air heater can, in principle and in contradistinction to a steam boiler, be installed in the open air without any building, all danger of freezing being nonexistent.

The design of the air heater is dependent upon the fuel that has to be dealt with. Coal-fired air heaters resemble in their design up-to-date steam boilers, as may be seen from the examples, Fig. 20 and Fig. 21. According to the present stage of development, the space requirements of coal-fired air heaters are not greater than those of up-to-date high-pressure boilers.

Fig. 20 shows a section through a coal-fired air heater with granulating chamber. This project for a 12,000-kw plant is conservative as regards the combustion chamber, as well as the stresses and temperatures for the tube walls, since care in this connection appears necessary for the first installation. It may, however, be definitely expected that subsequent developments will lead to the heating surfaces and the dimensions being considerably reduced. The inlet temperature of the tube nest in the convection section amounts to only about 1850 F. By returning the flue gases in the combustion chamber, its temperature is regulated and reduced. The tubes have diameters of approximately $1^1/_2$ to $^3/_4$ in. and have wall thicknesses of 0.15 to 0.1 in.

Fig. 21 shows a project for a plant of the same output with liquid-ash extraction and increased temperature in the combustion chamber for utilizing the radiation. The walls of the combustion chamber are lined with short tubes of small diameter. The air passes through them at high velocities so that the wall temperatures nevertheless remain sufficiently low, to permit of their being subjected to the radiation without further protection. In view of the fact that the waste gases from the air heater have relatively high temperatures as a result of the highly preheated circuit air, their waste heat is employed for preheating the combustion air. The use of preheated combustion air leads to an increase in the furnace temperature, particularly in cases where pulverized coal is used. For example, in plants burning pulverized coal such preheating of the secondary air is desirable since it permits a reduction in the size of the combustion chamber.

It is not purposed to deal further herein with the details of coal-fired air heaters, since our studies and investigations in this connection have not been completed in all respects. Experimental equipment for air heaters with pulverized-coal firing is being subjected to experimental work in Zurich. The behavior of ash and slag on hot tubes is being particularly studied. The results so far obtained justify the opinion that pulverized-coal firing will not present any fundamental difficulties for the AK-air heater.

Oil- or gas-fired heaters can be made more compact. In the case of the latter the possibility exists, especially where space is restricted as in marine installations, of supercharging the com-

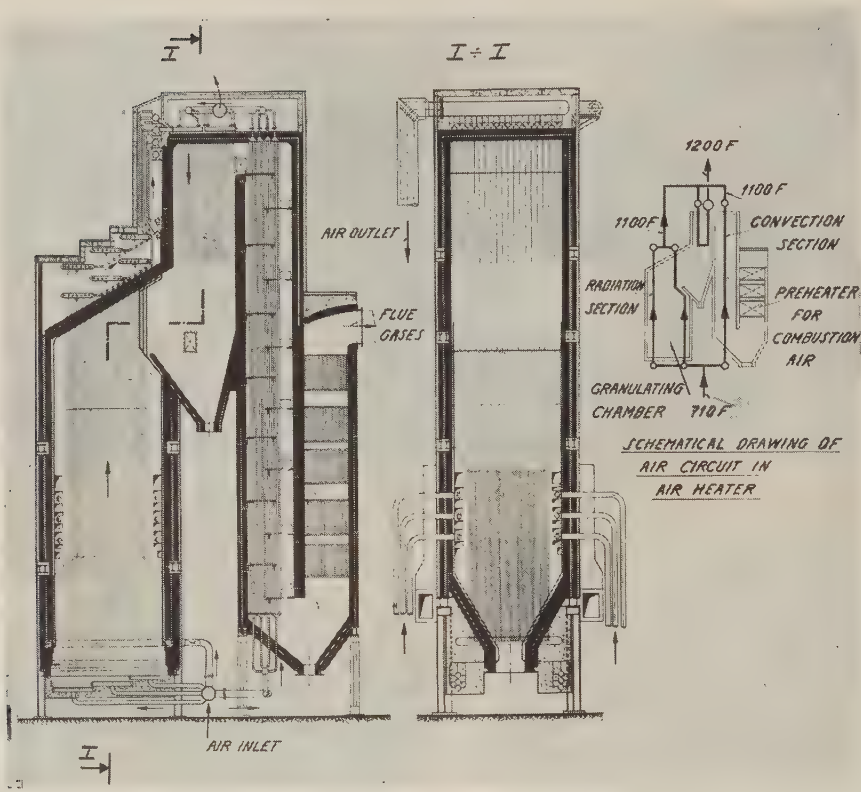


Fig. 20 Air-Heater for Coal-Firing With Granulating Chamber

bustion chamber by which means the dimensions and weights can again be considerably reduced (down to one half the surface) compared to ordinary firing.

Fig. 22 shows an example of an oil-fired air heater for a 6000-kw plant with 600 psia and double heating. The tubes are arranged around a supercharged combustion chamber. The tube walls for heating the air in the first stage and in the second stage, 600 psia and 200 psia, respectively, are united in a common heater. In the case of this project the combustion chamber is lined with refractory material, and the heat is given up to the tubes mainly by convection. By raising the pressure of the combustion gas to about 45 to 70 psia, and with velocities of combustion gases from 100 to 160 fps, and air velocities inside the tubes from 65 to 130 fps, the heating surface can be kept small. For such projects it amounts to only 0.5 sq ft per kw net output of the plant.

The total weight of iron in such an air heater is less than 9 lb per kw. Of this figure the proportion of alloy steel is about 50 per cent. The tubes, which may expand in operation by about $2^1/_2$ in., can move freely in the upward direction. In order to keep the hot-air piping as short as possible the heated air for the high-pressure turbine and low-pressure turbine is discharged below. The shell of the heater is of ordinary steel and made airtight. It can be dismantled in a number of sections so that the tubes are easily accessible from the side and can be removed without difficulty. The shell serves at the same time as a support for the tubes.

Since the waste gases are still of high temperature (approximately 1000 F), they are expanded in an exhaust turbine and simultaneously cooled. The exhaust turbine drives the compressor for the combustion air. In this way a preheater for the combustion air can be entirely eliminated. The supercharging set need not be particularly efficient because most of the losses are recovered in the firing. As there is no regenerator there is little danger of detrimental soiling. The design of an air heater employing blast-furnace gas or natural gas is, for the greater part, the same as for oil-firing. The heating surfaces are likewise quite similar.

The design of the air heater according to Fig. 22 corresponds in principle to the one which has proved its merits in the experimental plant from the beginning, only with the difference that in the latter case firing under pressure was not adopted, for the sake of simplicity.

The working medium which has passed through the heat exchanger is supplied to the air heater always in a highly preheated state (600 to 750 F). However, the supply of heat does not bring about any change in condition (evaporation) as is the case in the steam boiler. Evaporating elements, large collectors, and the drums of steam boilers can therefore be dispensed with. Since the air is under pressure, the employment of tubes as heating elements is found to be the most satisfactory means. According to the fuel adopted and the temperature in the furnace, the heating surface is subdivided into a radiation section and a convection section.

In view of the fact that a gaseous medium has to be heated by means of combustion gas, higher tube-wall temperatures have at first to be considered than in the case of steam, where, as a result of the high heat-transmission coefficients on the water and steam side, the tube-wall temperatures are not much higher than the steam temperatures. The working pressures at which the

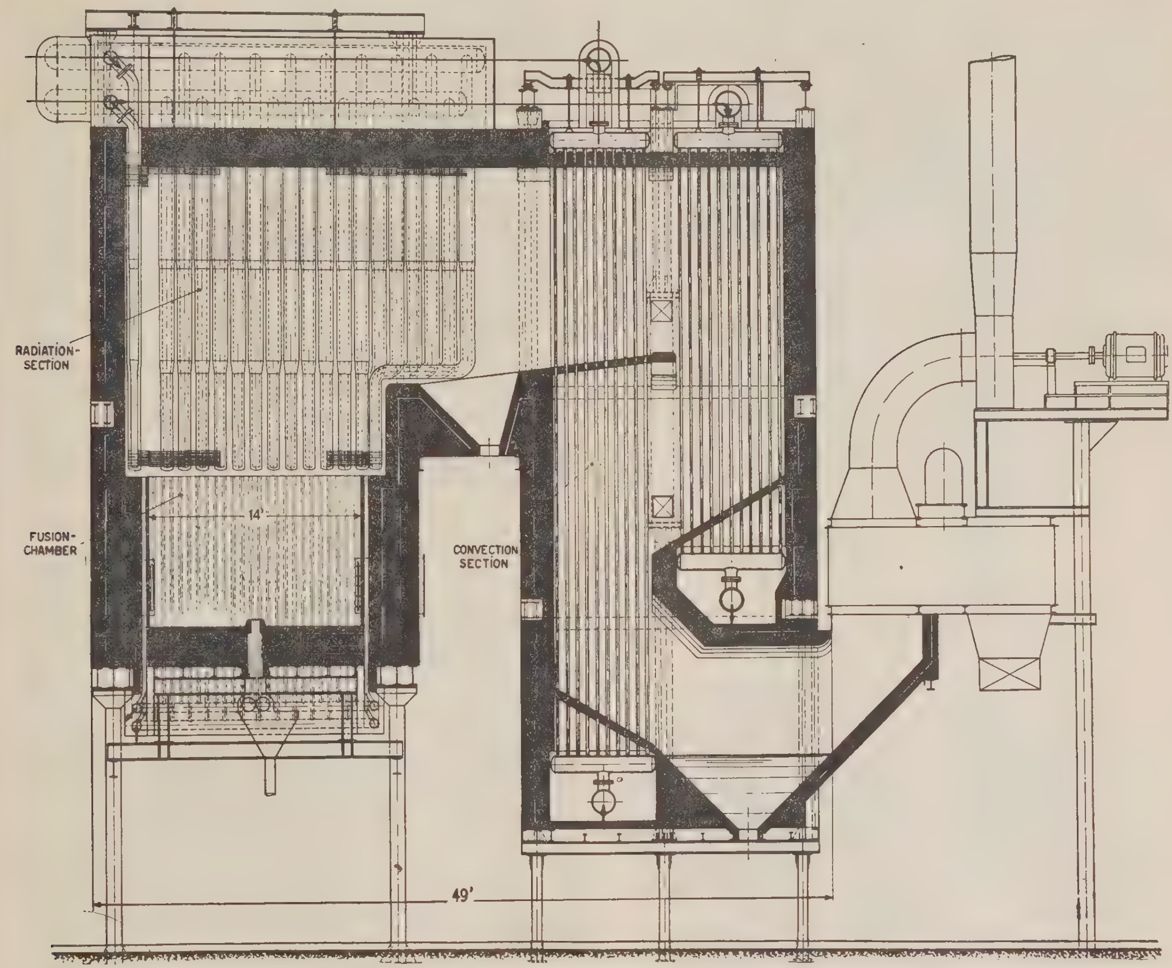


FIG. 21 AIR-HEATER WITH FUSING CHAMBER FOR PULVERIZED COAL

closed cycle operates have the same effect. The heat-transmission coefficient in the interior of the tubes can be so great that the tube-wall temperature is displaced to a considerable extent toward the cold side and comes within ranges to which alloyed steels nowadays obtainable on the market can be subjected without hesitation. The maxi mum temperature of the hottest tube wall can be forced down to 70 to 100 deg F of the final temperature of the air. The heat-transmission coefficient for the interior of the tube is

$$\alpha = c\,(w \cdot p)^{0.75}$$

where

c = constant factor
w = velocity
p = pressure

and accordingly proportional to the product $w \cdot p$.

Thus for the same velocities the heat-transmission coefficient, compared to conditions at atmospheric pressure, increases many times, for example, at 400 psi to 12 times, at 850 psi to 22 times. For practical cases heat-transmission coefficients of 100 to 200 Btu per sq ft per deg F per hr for the interior of the tubes can be reckoned with. Fig. 23 shows, by way of example, how for the same-percentage pressure drop $\epsilon = \Delta\ p/p$ per unit length, the tube-wall temperatures automatically fall as a consequence of increased pressure in the interior without velocity increase, so that the air-heater tubes can be subjected to the duty in the combustion chamber without special cooling measures being necessary.

A simple means for compensating for excessive combustion-chamber temperatures is a return circuit of the flue gases to the combustion chamber, whereby practically any desired temperature can be adhered to. This has been effected also in the oil-fired air heater of the test unit of 2000 kw.

It is often assumed that a very high velocity and a considerable detrimental pressure drop are necessary in the heater tubes for carrying away the quantity of heat. However, as a consequence of the pressure action and with suitable layouts this is by no means the case. For the air heater of a 400-psi plant the pressure drop is only about 10 to 15 psia; at 850 psi for two-stage heaters it totals 20 to 30 psi.

If the flow velocities in the interior of the tubes could be increased at will, any desired reduction of the tube-wall temperature down to the temperature of the air current inside the tubes could theoretically be envisaged. But this can be attained only with a loss of over-all efficiency.

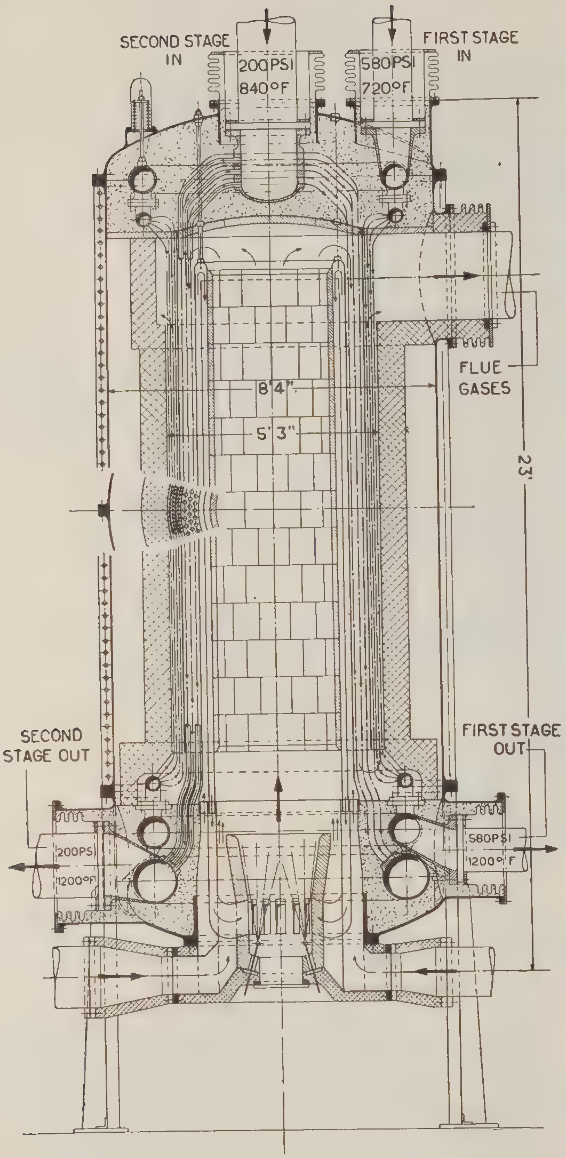


Fig. 22 Oil-Fired Air Heater for 6000-Kw Set Supercharged

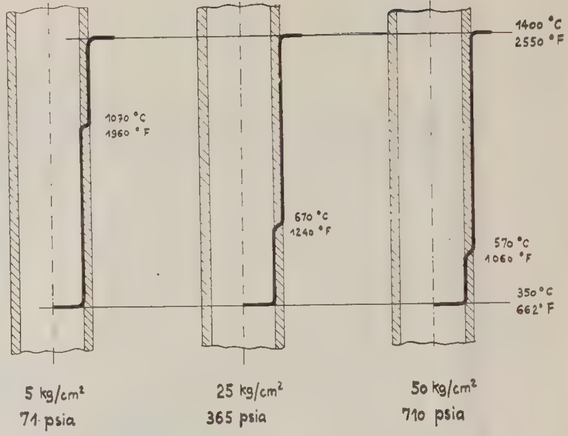


Fig. 23 Influence of Inside Pressure of Air-Heater Tubes on Wall Temperature

An increase in the velocity involves the need for minimum pressure losses. Thus the velocity in the various heater sections will be increased only to such an extent as is necessary for attaining the admissible tube-wall temperature. This knowledge leads to the adoption of tubes having different diameters which are suited to the various ranges of temperature.

Careful calculations of all these problems have led to a series of fundamental circulation layouts for the working medium to be heated and for heating combustion gases. The wall temperature can be further reduced by suitable layouts employing countercurrents, parallel currents and transverse currents in the various heater sections.

Fig. 20 illustrates one of the many solutions that can be adopted in this connection. The countercurrent to be heated is subdivided at the inlet into two parallel currents, one of which, mainly subjected to radiation, flows as a direct current and the other, the temperature of which is raised by contact with heated parts, flows as a countercurrent to the flue gases. This brings about a reduction of temperature in the hottest sections. At the same time the subdivision into two or more parallel currents permits a considerable reduction in the total pressure losses. The two partial currents pass into a common final heater and collector pipe.

If it is desired to avoid excessively high temperatures in the radiation section, the pipes can be arranged in such a manner that final heating takes place in the section heated by contact, where the tubes are protected from the effects of radiation.

Only a part of the air heater must be made of special high-quality materials. In this connection the fact should not be lost sight of that for the working pressures concerned the wall stresses are astonishingly small (2500 to 4000 psi), even with thin tubes. Those figures still lie far below the creep limits of good alloy steels for the corresponding temperature ranges.

Heat Exchanger (*Recuperator*). Unless a heat exchanger is employed it is hardly possible to increase the thermal efficiency of any kind of gas turbine plant above about 20 per cent. The quantity of heat that has to be given up by the current of gas issuing from the turbine to the current of gas after the compressor at full regeneration is of the same magnitude as the quantity of heat which in the air heater is introduced from an external source.

The heat-transmission coefficients on both sides of the heat-exchanger surfaces are raised considerably in the case of the supercharged closed cycle. In combination with the small specific volume this offers possibilities of reducing considerably both dimensions and weights. A further point is that one can in principle adopt very small cross sections for the tubes, or other fine exchanger elements, because any danger of soiling is entirely eliminated.

The conditions for constructing a good heat exchanger of small dimensions are doubtless more favorable for a closed circuit than for gas turbines with which the flue gases pass through the heat exchangers. Apart from the high blading efficiency of the machines, the easy realization of a very high heat exchange is one of the chief reasons for the high over-all efficiency of closed-circuit installations, even when employing moderate temperatures.

In practice heat-transfer coefficients of 30 to 50 Btu per sq ft per deg F per hr, depending on the admissible pressure loss, may be taken as a basis. These figures are multiples of the heat-transfer coefficient characteristic of the heat exchangers for com-

bustion-gas turbines. For economical reasons one is compelled not to go too far with the recuperation in the case of the last-mentioned plants, since space and weight for the construction would become too great. According to available figures, such new two-stage installations require about 3 times or more heating surface. With the closed cycle we find surfaces of 1.5 to 3 sq ft per kw sufficient, depending upon the pressure, and furthermore for an over-all efficiency of the plant exceeding 33 per cent; the degree of recuperation being about 90 per cent. It is possible to bring the weight of the heating surfaces down to even less than 2 lb per kw. The following illustrations likewise indicate clearly that the heat exchangers of closed-circuit installations are relatively small.

In principle, any kind of heat-exchanging surface (flat or tubular), and any means for conducting the current can be adopted. However, since the plant operates under pressure, the use of simple steel tubes which can be manufactured at relatively low cost represents the given means for effecting this transmission.

Fig. 24 illustrates the design of a tubular exchanger operating according to the countercurrent principle and transmitting a maximum amount of heat with a minimum loss of pressure. The high-pressure air flows through the interior of the tubes and the external shell has to withstand only the lower back pressure. The apparatus can be arranged as desired, according to the available space or subdivided into two or more parts. The use of normal tubes of small diameter permits standardization of all heat-exchanger elements for the various outputs and thus to a large degree manufacture in series.

We employ tubes of only 0.15 to 0.25 in. diam. They are separated by special spacers which offer little resistance to the current of air. A large number of these thin tubes is assembled in a tube nest. In their turn the latter are connected by a small number of collector pipes. Separate removal of each tube nest is easily possible. In the case of stationary plants this can be effected, for example, by drawing the various tube sections out of the heat exchanger in an axial direction. For marine installations the shell of the heat exchanger is split and can be easily lifted, thus giving access to the tube nests from above. The tightness of each nest can be checked separately. However, since a perfectly clean medium flows on either side of the tube walls, there is no reason to fear interruptions in the operation.

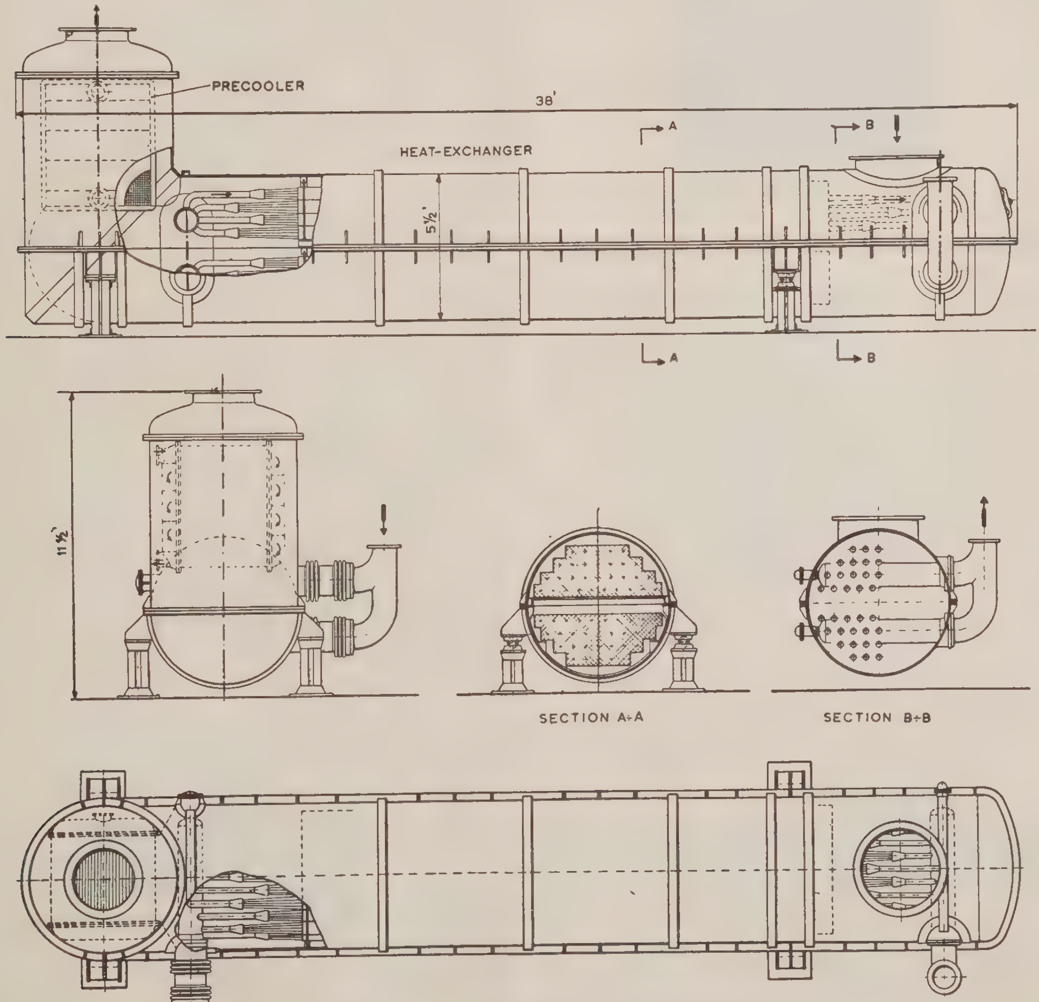


Fig. 24 Heat Exchanger or Recuperator for Closed-Cycle Unit, Built Up of Small-Diameter Tubes

Furthermore, the highest temperature in the heat exchanger amounts only to about 850 F, so that it is unnecessary to employ special-quality metals for the tubes. Fig. 25 shows a view in the part of the apparatus where the spacers are arranged; as may be noted, the external bright part where the low-pressure air flows is hardly obstructed by these distance pieces.

The temperature of the low-pressure air on issuing from the heat exchanger is about 200 to 250 F, after which it passes to a precooler, through which water circulates, for cooling in so far as possible down to the inlet temperature of the compressor, the object being to reduce the compression work. Since, as a result of the raised pressure the heat-transmission coefficients on the air side are also favorable in the precooler and in the intermediate coolers of the compressor, the surfaces and dimensions of these units are not large, in contradistinction to water-cooled air coolers when operating with small working pressures. Incidentally, the precoolers and intermediate coolers are of

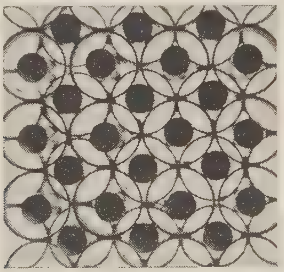


Fig. 25 View of Space Arrangement in Regenerator

standard ribbed-tube design with water circulation through the tubes.

In many cases it is advantageous to combine the low-pressure turbine arranged in the circuit before the heat exchanger in the manner illustrated in Fig. 26. In this way additional pressure losses are avoided and the stream of air passes directly to the tube nests. As may be seen in Fig. 26, the heat exchanger, whether arranged in this or some similar manner, is an exceptionally simple apparatus that operates very reliably.

Governing. For raising the output, air is introduced to the circuit from a high-pressure accumulator of cold air, while for reducing the output, air is withdrawn from the circuit and passed to a low-pressure accumulator. Automatic governing for an installation with rigid couplings between the machines takes place fundamentally as follows: When load is thrown off the consequent rise in speed influences the centrifugal governor (pendulum) which causes the discharge side of the combined inlet-outlet valve to open, Fig. 27, so that air from the high-pressure branch of the circuit issues into the low-pressure accumulator LP. On the other hand, when load is thrown on the plant the resulting drop in speed causes the inlet side of the valve to open, as a consequence of which air from the high-pressure accumulator HP is admitted to the circuit at the same point.

For small reductions in load, and consequently only slight increases in speed, the main valve operates only within a small range without opening either the inlet or outlet. On the other hand, the by-pass valve opens and by-passes air from the high-pressure side of the circuit to the low-pressure side without developing output, so that the useful output of the plant is reduced.

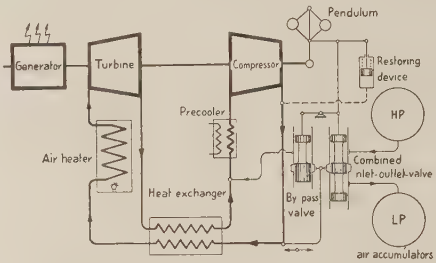


Fig. 27 Scheme of Regulating Device for One-Shaft Machine Set

(Combination of inlet and outlet valve with by-pass valve.)

By regulating small and frequently recurring load fluctuations with the by-pass valve the air consumption from accumulator HP, for purposes of regulation, is reduced and charging work thus saved.

Regulation by supplying or withdrawing working medium to or from the high-pressure side of the circuit has the advantage of immediate efficacy, since the pressure ratio $p_H : p_L$ is immediately raised on air being admitted, thus causing the plant to give up additional output. Inversely, when air is withdrawn from the high-pressure side, the pressure ratio immediately drops, so that for example, when suddenly withdrawing less than 20 per cent of the air content of the circuit, it has dropped to such an extent that the transition from full load to no load has already taken place. On the other hand this "momentary effect" would be unsuitable for the supply or withdrawal of working medium on the low-pressure side.

When admitting or withdrawing air on the high-pressure side of the circuit the consumption of air for such regulating purposes is, in the case of quickly recurring small periodical load fluctuations, no longer proportional to the number of actual load fluctuations and instead increases relatively less, since insufficient time remains between the separate fluctuations for re-establishing the stationary pressure ratio, and the control consequently takes place chiefly under the influence of the "momentary effect," Fig. 28.

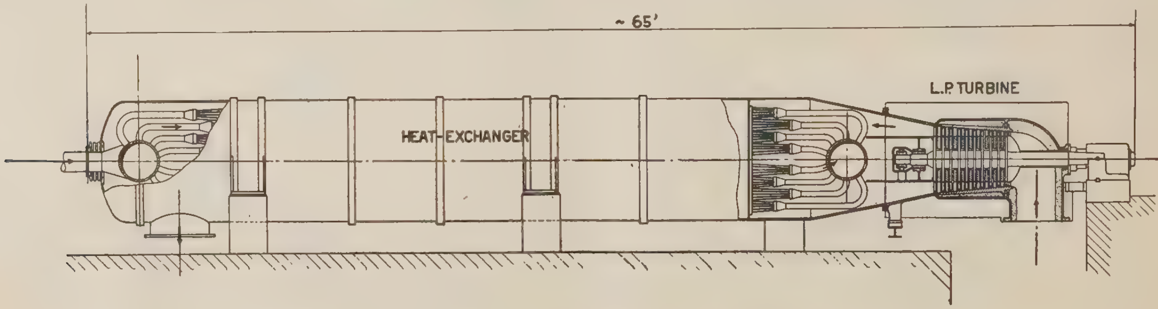


Fig. 26 Combination of Heat Exchanger and Turbine to Conserve Space and Pressure Loss

The influence of the momentary effect on the governing, which for known machine characteristics can also be calculated theoretically, has been checked by experiments. The pressure course, Fig. 29, plotted for a load-reducing action, shows that during the first moment the equilibrium in output is brought about by reducing the pressure ratio $p_H : p_L$; and that the pressure level only gradually drops to the final condition.

The regulation of the furnace, not indicated on the schematic drawing, Fig. 27, need not take place very quickly, thanks to the accumulation of heat in the heaters and apparatus. Changes in the power output of the turbine due to smaller deviations of the final heating temperature of the working air from its stationary value are compensated by automatic and temporary raising or lowering of the pressure level by a small amount.

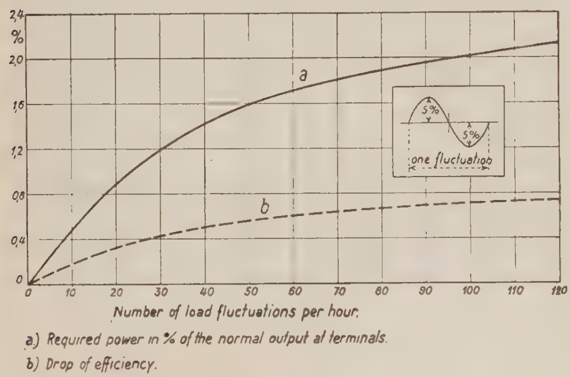


Fig. 28 Power Needed in Per Cent of Net Output for Production of Accumulated Air at Load Fluctuations of ±5 Per Cent

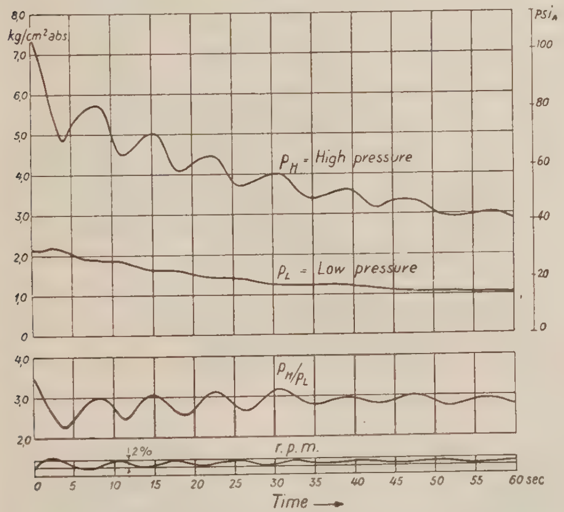


Fig. 29 Pressure Variation at Sudden Load Reductions Reveals "Momentary Effect"

The method of regulation herein described has the advantage that the regulating means can be accommodated in a block outside the circuit proper, Fig. 30, that they are traversed only by cold air, and that under normal working conditions the air of the circuit does not pass through them, so that no additional throttling losses result.

For installations in which the turbine developing useful output is separated mechanically from the compressor set, which is particularly the case where useful output has to be given up at different speeds (for example, compressor drive, ship propulsion), the compressor set operates in the normal operating condition, thus remaining stable and without any special regulation. When the equilibrium is disturbed, i.e., during the supply and withdrawal of working medium for changing the useful output, deviations from a given speed range, either above or below, for the free-running compressor set, are prevented by partly by-passing the turbine developing useful output or the compressor-driving turbine by a special valve actuated from a speed-limiting governor of the compressor set, Fig. 31.

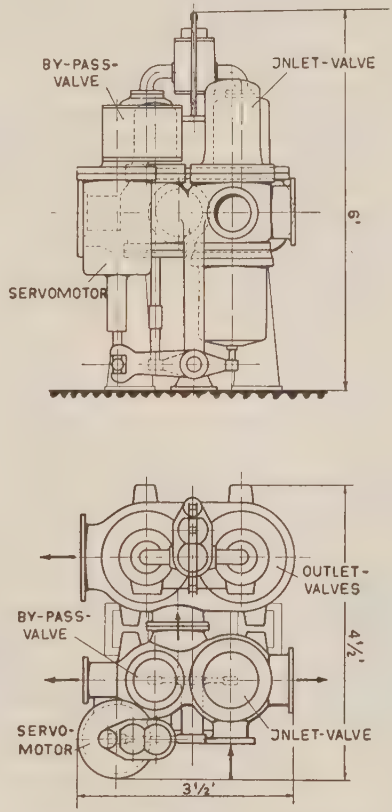


Fig. 30 Regulating Block for 12,000-Kw Set

The test plant has worked for long periods entirely separated from the municipal network and supplying the whole works of Escher Wyss. On these occasions the regulating governing, already in its simplest form, proved to be very satisfactory.

Projects for Power Stations and Marine Plants

The disposition of the various machines and apparatus of an AK-plant within the available space does not involve special requirements. Air is not subject to the force of gravity like steam condensate, so that differences in level, such as are required for insuring passage of the condensate and feedwater, need not be provided for. Thus attention can be paid to a combination of the machines and apparatus in a small closed set with short connecting pipes in order to reduce losses of pressure and temperature to a minimum, of the utmost importance in such plants.

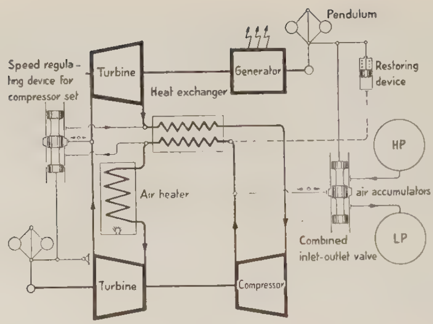


Fig. 31 Schematic Diagram of Regulating Device for Two-Shaft Machine Set
(Marine installations, for example.)

The limited number of auxiliary machines and other accessories greatly simplifies the operation of the plant, whereas in the case of high-pressure steam power stations for high thermal efficiencies, these auxiliaries have become rather cumbersome. The chief factors which bring about a considerable improvement, compared to a steam plant, are the elimination of feedwater, the necessary pumps and apparatus for its preparation, the small cooling-water consumption which amounts to only a fraction ($^1/_5$ to $^1/_{10}$) of that required in steam installations, the elimination of the condensing process with its condensate pumps and air pumps, and the absence of all fittings for very high pressures.

Closed-cycle installations may be operated entirely without water, in that intermediate cooling of the air in the compressor can be effected by employing ambient air as cooling means in place of water. For the large available temperature drops of the intermediate cooling, this arrangement with which cooling water is entirely eliminated, can be realized in an economical manner in contradistinction to steam plants where the condensing process at constant low temperature makes it necessary to endeavor to attain minimum temperature differences when giving up waste heat to the ambient air.

If one takes into account:

1 The relatively small expense involved for special materials capable of withstanding the high temperatures (as a consequence of the small dimensions resulting from supercharged operation);

2 The simple layout of the plant (absence of extensive foundations);

3 The reduced number of accessories;

then it is evident that in spite of the higher thermal efficiency, compared with those of steam plants of the same output, the total costs of the two installations will not greatly differ from one another.

Fig. 31 illustrates, by way of example, a number of AK-installations such as are ready for construction. They are all characterized by high efficiencies, well above 30 per cent. These installations employ pure air as working medium. Fig. 32 illustrates a 12,000-kw plant with pulverized-coal firing for industrial

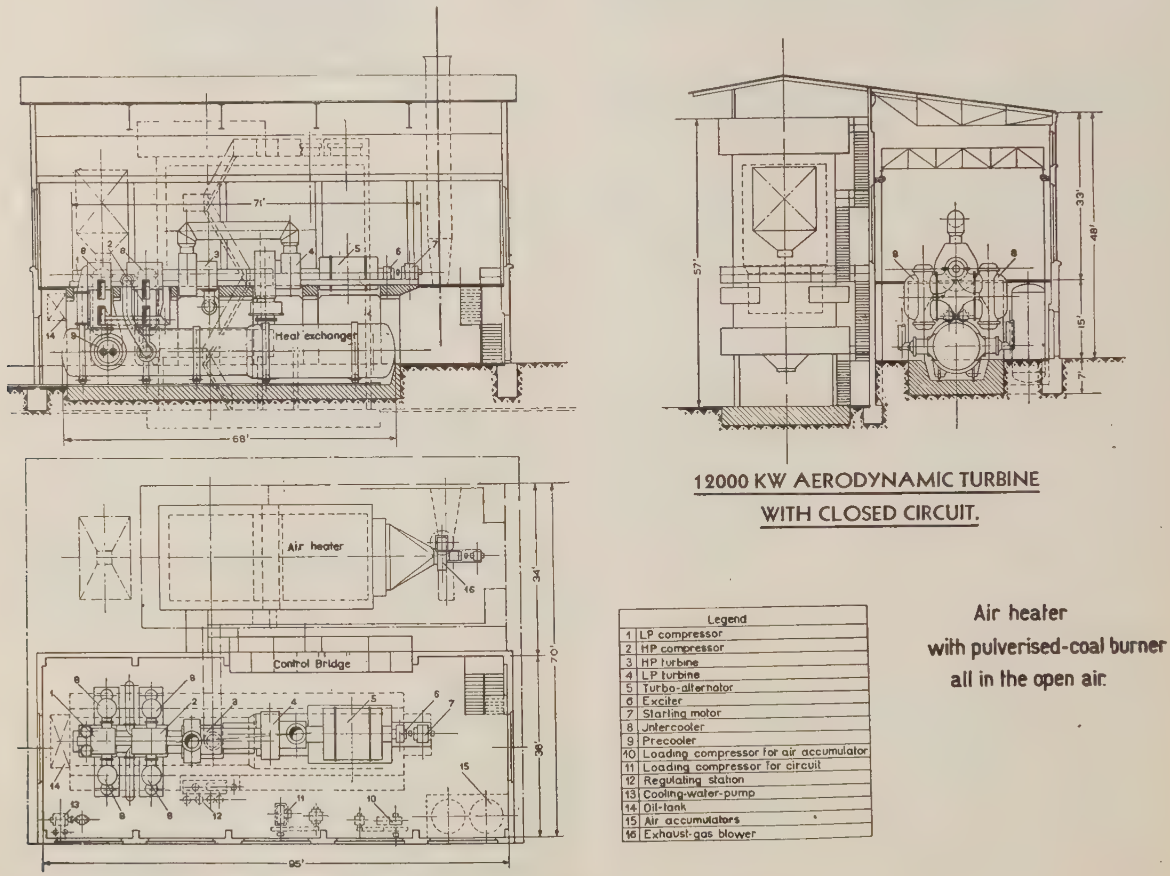


Fig. 32 12,000-Kw AK-Set for Single-Stage Power Plant With Coal-Firing

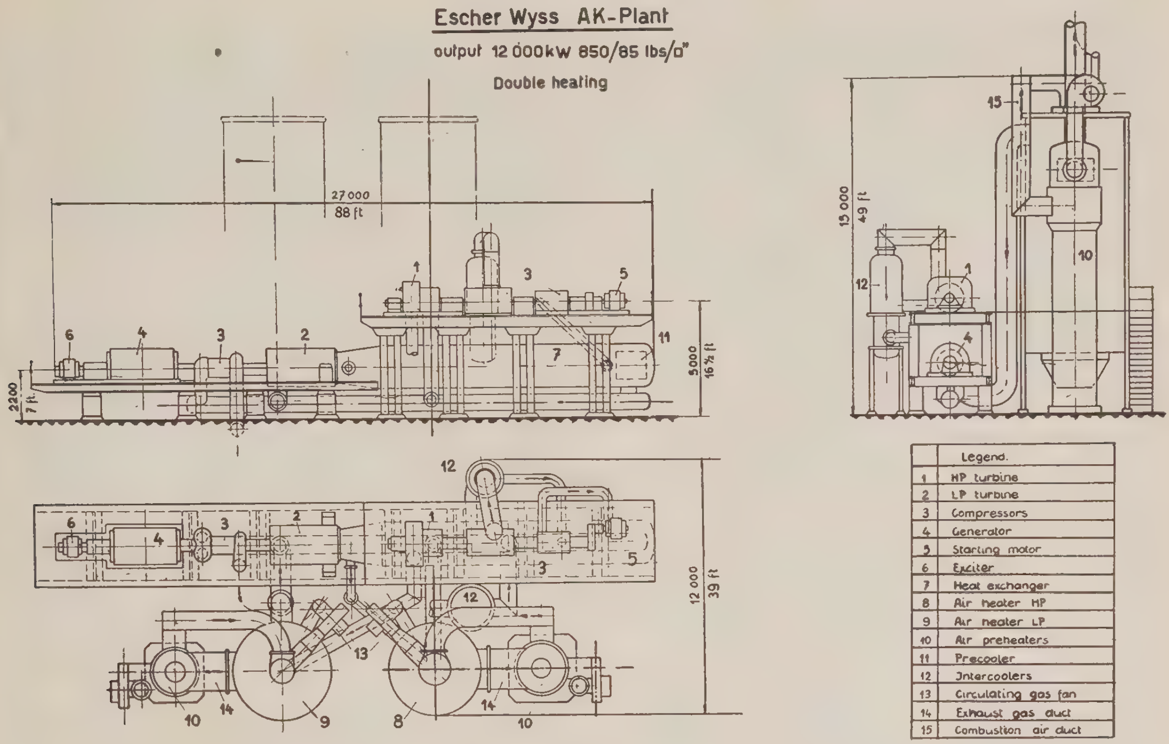


Fig. 33 12,000-Kw AK-Set With Oil- or Gas-Firing, and Intermediate Heating

purposes. The air heater is installed in the open next to the powerhouse, thus considerably reducing the cost of the building. This is a plant with single-stage expansion. The working pressure at full load amounts to 400 F at the turbine inlet, the maximum temperature to 1200 F, and the back pressure to 115 psia. The machines are arranged all in one row. The high-pressure turbine drives the compressor serving the circuit while the low-pressure turbine drives the generator. The heat exchanger is installed underneath the machine set but can of course also be arranged elsewhere, for example, in the open, depending upon the available space. The weight of the circulating air is about 250 lb per sec. All the chief auxiliary drives that are required may be seen in the illustration. Their small number in comparison to the auxiliaries of up-to-date steam power plants is characteristic. The over-all thermal efficiency of such an installation amounts to at least 32 to 33 per cent at full load, still attaining 31 per cent at half load and 28 per cent at quarter load. Naturally these figures vary somewhat according to the quality of the coal.

Fig. 33 illustrates a project for a plant of the same output but arranged for oil- or gas-firing and double heating, the initial pressure being 850 psi and the back pressure 85 psi. The machine plant is subdivided into two independent sets. The high-pressure turbine 1 is fed from the first air heater 8 and drives a part of the compressor rotors. This compressor set is arranged above the heat exchanger 7. The low-pressure turbine 2 drives the generator 4 and the low-pressure part of the compressor 3. The low-pressure turbine and heat exchanger are built together, Fig. 26. The low-pressure turbine receives the reheated air at a pressure of 300 psi from the second air heater 9.

In these stationary installations where the space requirement is not of decisive importance, ordinary furnace-chamber pressures can be adopted for oil- and gas-fired air heaters. This means that the required heating surface becomes larger than for firing under pressure, but on the other hand the charging set is eliminated. In place of the latter an air preheater for the combustion air is adopted. In this case the air preheaters are built as tubular units. The air heaters are arranged immediately next to the machine set proper which leads to favorably short interconnecting pipe work. The fact that no excavation is required for the set and that it rests on a light foundation framework is worthy of note. All parts are easily accessible.

The starting motor 6 has an output of approximately 300 to 400 kw. On the basis of the present state of development, a thermal efficiency, including auxiliaries of 34 to 37 per cent, is attainable at full load and 30 to 33 per cent at one-fifth load. The efficiency is, of course, dependent upon the size of the heat-exchanger and air-heater surfaces as well as upon the cooling-water temperature. For an inlet temperature of 60 F and 160 F exit approximately 40 gal per sec are required. A steam plant of the same output needs about 5 times more. The circulating-air weight is approximately 150 lb per sec.

Fig. 34 illustrates the machine designed for a plant of 25,000 kw output. It remains the same for oil-, gas-, or pulverized-coal firing. Details of the corresponding machines are illustrated in Figs. 12, 13, and 14. The machinery plant can of course be arranged in two sets parallel to one another instead of in a single row, if the available space makes this preferable. When comparing with Fig. 33 it is apparent that the larger output calls for hardly any additional length, since it is primarily the diameter of the machines and auxiliaries which is increased.

An arrangement of the sets in parallel is preferable for marine installations. Fig. 35 shows a marine plant with oil firing for 8000 shp. This plant can be used for turboelectric drive or

for propulsion in conjunction with a variable-pitch propeller. It operates with 600 psi, 1200 F at the turbine inlet, with double heating and supercharged firing of the air heater. The combustion circuit is entirely separate from the working circuit proper, so that they do not influence one another during regulating actions. The reduced strain for the tubes in the case of supercharged firing permits of the wall thicknesses being reduced. The back pressure amounts to 60 psi. The air heater is arranged in front of the machine set. The heat exchanger lies amidships. On the one side is the compressor set for the circuit. It is driven by the high-pressure turbine. On the same side is the charging set for the combustion chamber which receives its drive from an exhaust turbine. The low-pressure turbine is the prime mover proper.

With this arrangement all parts are easily accessible and can be dismantled without difficulty since they all lie in the same plane. When the propeller speed is changed under varying loads, only the speed of the low-pressure turbine is correspondingly altered. The independent regulation of the two machine sets takes place according to the schematic drawing, Fig. 31.

The thermal efficiency of such a marine plant including auxiliaries amounts, at full load and 1200 to 1300 F, to about 32 to 34 per cent, and still attains at one-fifth load an efficiency of 27 to 30 per cent. The weight of the installation in relation to output is approximately 40 lb per shp. The weights, space requirements, and efficiencies are primarily dependent upon the surfaces that have to be adopted for the heat exchangers. The space requirements of a plant, according to Fig. 35, are approximately 2.4 cu ft per hp.

In accordance with present progress in metallurgy it is intended to operate AK-plants at temperatures of 1200 to 1300 F. Every future increase that can be made in the temperatures will, for each 10 deg F, lead to a saving in fuel of 0.75 per cent, i.e., for 40 deg F increase in temperature an improvement in the over-all thermal efficiency of about 1 per cent may be expected.

When making a comparison with open-cycle gas turbines, the pipes passing through the deck which are necessary for supplying the combustion air and for discharging the combustion gases should be borne in mind. Whereas a closed-cycle turbine requires practically the same quantity of fresh air and discharges about the same volumes of waste gas through the funnels as in the case of Diesel engines or steam plants, these volumes are many times greater for an open-cycle combustion-gas turbine because of the large surplus air volume of its working cycle for the purpose of reducing the temperature. Needless to say, the closed-cycle turbine is very suitable for turboelectric drive.

One cannot discuss new gas-turbine developments for marine propulsion without mentioning the future development of the propelling means.

The most suitable solution for reverse operation which, at the same time gives ideal operating conditions for the whole plant, is the adoption of variable-pitch propellers, thus obviating the necessity of installing a separate turbine for propelling the vessel astern. From wide experience in the building of Kaplan-type water turbines, in 1934 we developed such a new marine propeller based upon the same principles. No failures or defects have been reported from the 35 marine propellers already delivered.

Other Fields of Application

Remote Heating. In contrast to the condensing plant of steam turbines, where heating of the cooling water may amount to only a few degrees owing to the necessity of maintaining a good vacuum, the required elimination of heat from a closed circuit during the compression can involve a considerable increase in the temperature of the cooling water without any drawbacks resulting. In this way the quantity of cooling water is reduced from 10 to 20 per cent compared to steam plants. Furthermore, heating of the cooling water can be raised from 160 to 180 F, without alteration of the temperatures within the circuit. Thus the waste heat is given up at high temperature and can be utilized for heating purposes. It is particularly worthy of note that the entire quantity of waste heat can be made use of for heating purposes, without any modifications having to be made, as compared to operation without utilization of the waste heat. The final temperature of the cooling water can be raised to any desired level, for instance, by restricting the recuperation.

AK-Plant in Connection With Blast Furnace. In blast furnaces, coke ovens, oil fields, and refineries as well as in various branches of the chemical industry, large quantities of waste gas result from the processes employed.

Utilization of such surplus gases in open-cycle combustion turbines calls for compression of the gases to a pressure amounting to a few atmospheres, so that they can be burned in the combustion chamber which is under pressure. For this purpose the gases must be precooled and in the majority of cases also cleaned, i.e., all processes which involve considerable additional apparatus. In the case of a closed-cycle turbine the hot gases can be utilized just as they are, without preparation and compression of the fuel for the air heater, i.e., a possibility which increases the thermal over-all efficiency.

A closed-cycle turbine operated with blast-furnace gas can also be employed for direct drive of blast-furnace blowers. The

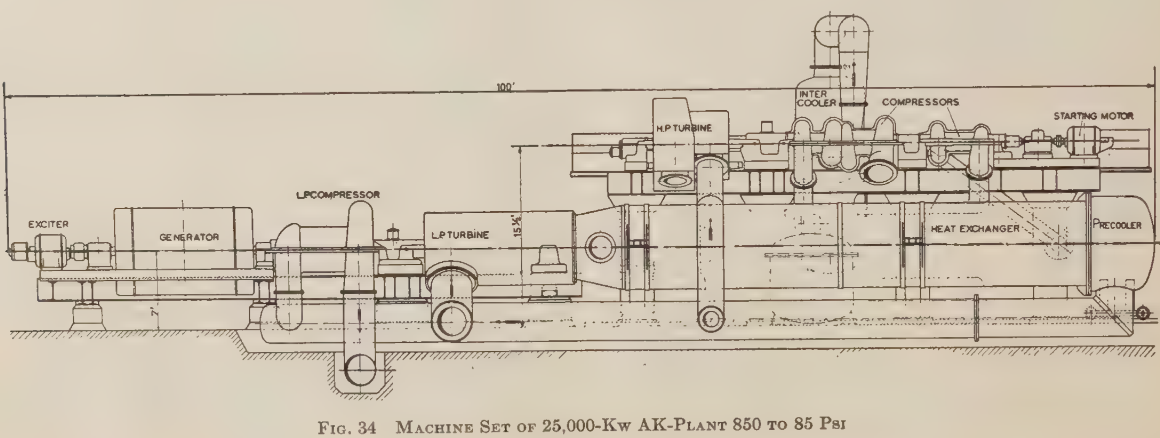


Fig. 34 Machine Set of 25,000-Kw AK-Plant 850 to 85 Psi

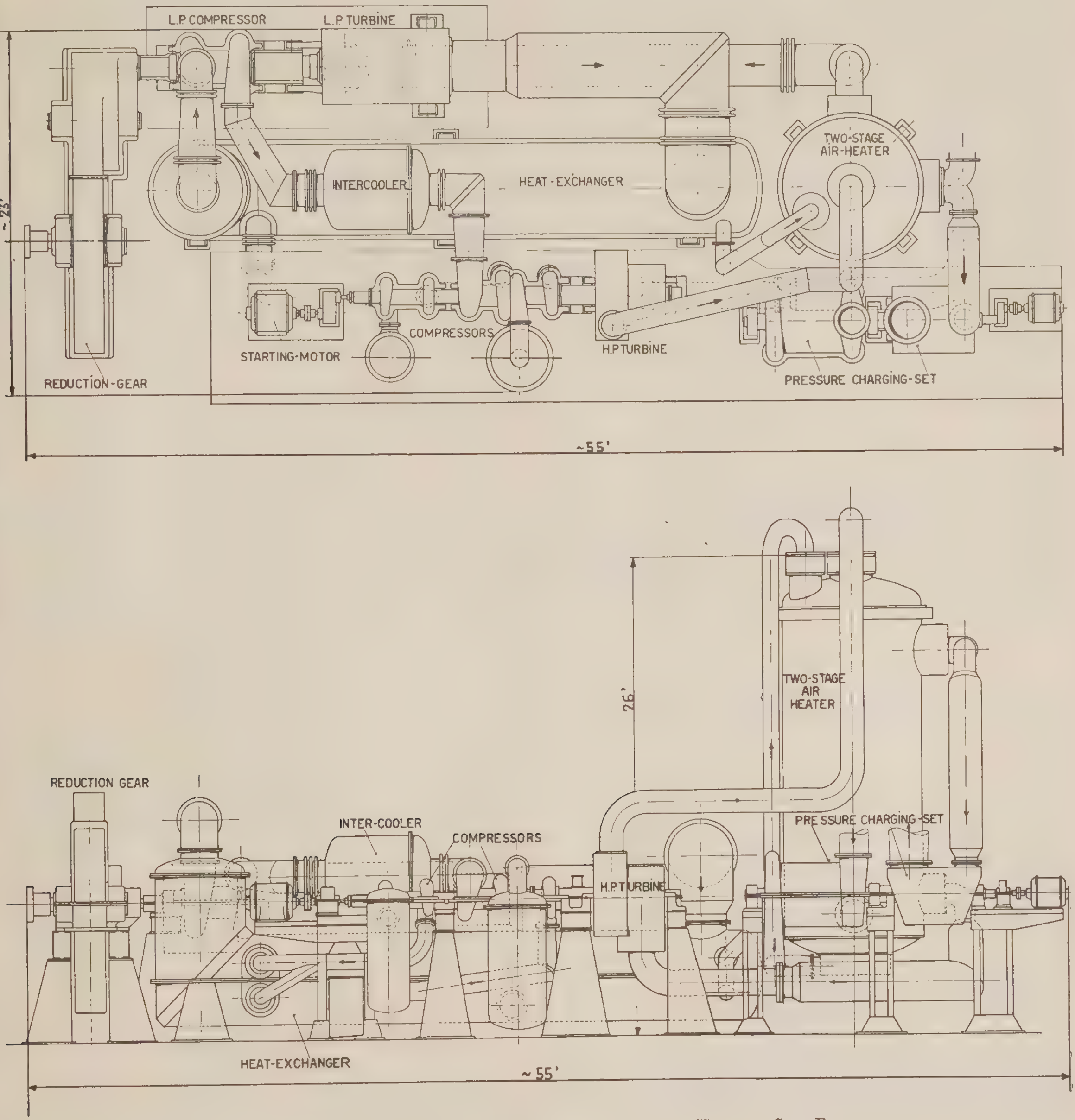


Fig. 35 Layout of a High-Efficiency 8000-Shp Closed-Cycle Unit for Ship Propulsion

arrangement of the turbine stages in conjunction with the compressor required for the circuit and the blast-furnace blowers, which must operate under widely differing speeds and loads, can be effected on similar lines to those already described in the preceding section concerning marine turbines. In this case also, the provision of two sets operating independently of one another may prove advantageous, since in this way a possibility is offered of attaining favorable thermal efficiencies over a large operating range, with regulation of the circuit density at different loads.

Another point to which brief reference may be made herein is that favorable combinations of blast furnaces, blast heaters, and air heaters for the power process can be realized for AK-plants. Particulars concerning such projects are to be found in a previous article (5).

Gas Generator and Fuel Producer. When coal is employed as fuel it may, according to its nature and quality, prove convenient to adopt gas generators in place of furnaces for producing the combustion gases for the air heater. In this connection, cleaning, cooling down, or compression are not necessary, such as are called for when an open cycle is adopted. The produced gas can be directly fired in an air heater.

Remarks Concerning Use of Light Gases in Closed Circuits

The following remarks concerning the advantages of employing other gases are the result of theoretical studies that have been

carried out by Professor Ackeret. He has compiled his calculations in the following form especially for the purposes of this paper:

For congruent entropy diagrams the efficiency of the closed-circuit process is independent of the nature of the gas. When assuming congruency, it is taken for granted that the relative pressure losses $\frac{\Delta T}{T}$ and temperature losses $\frac{\Delta p}{p}$ are the same on all heat-transmission surfaces. However, in spite of this, the adoption of other gases can prove advantageous because the output of the plant can be considerably increased at the same efficiency and with only slightly changed dimensions. To prove this more decisively let us assume that gases of the same atomic number, for example, only biatomic or only monoatomic gases be employed, the molecular weight m of which can be continually changed. For example, by a mixture of helium and argon, m could be continually increased from 4 to 40. For gases of the same atomic number the magnitudes

$$k = C_p/C_v \text{ and } \sigma = \frac{\eta \cdot g \cdot C_p}{\lambda}$$

are as is known independent from m.

We may observe the heat transmission at a given point of a tube having the diameter D which is traversed at said point by gases with velocity u, density ρ, pressure p, temperature T and viscosity η. Between the wall and the gas a difference in temperature $\Delta T = \theta$ shall prevail. The following equation then applies for the shear stress

$$\tau = C_f \frac{\rho}{2} u^2$$

and for the heat transmission per unit of surface according to Reynolds and Prandtl

$$g = \frac{C_f}{2P} \rho g C_p \cdot u \cdot \theta$$

in which for Reynolds numbers of medium size $\mathsf{R} = \frac{\rho u D}{\eta}$ the following may according to Blasius be inserted

$$C_f = \frac{0.0791}{\sqrt[4]{\mathsf{R}}}$$

The expression

$$P = 1 + \frac{1.7^4}{\mathsf{R}^{1/8}} (\sigma - 1)$$

is in small measure dependent only on R and is about 1. According to Eucken, σ is connected with k and may be expressed with considerable accuracy by

$$\sigma = \frac{4k}{gk - 5}$$

For a short tube section the following can immediately be ascertained

$$dp = -4C_f \frac{\rho}{2} u^2 \frac{dx}{D}$$

Congruence of the entropy diagrams, i.e., the same efficiencies, is realized when $\frac{dp}{p}$ and $\frac{dT}{T}$ are of equal magnitude for the different gases. Hereby, however, dx must be somewhat changed. In order to utilize the material equally well in all cases, let us further assume that at full load the pressures are also equally large. For gases the following equation generally applies

$$R = \frac{R}{m} \qquad \rho = \frac{p}{gRT} = \frac{pm}{gRT}$$

and in addition for gases of the same atomic number

$$C_p = \frac{p}{m}$$

Thus if we require for the same $\frac{dp}{p}$, it follows that

$$\frac{dp}{p} = -2C_f \frac{\rho}{p} u^2 \frac{dx}{D} = -2C_f \frac{u^2}{gRT} \frac{dx}{D} = -2C_f k M^2 \frac{dx}{D}$$

in which M is the Mach number $= M = \frac{u}{a}$ (a = velocity of sound). The same $\frac{dT}{T}$ leads to

$$\frac{dT}{T} = \frac{2c_f}{PT} \theta \frac{dx}{D} = \text{const}$$

in other words, $\frac{c_f \cdot dx}{P}$ or, since we can look upon P as being practically constant, $c_f \cdot dx$ = const for all m; but from the foregoing it follows that M = const.

Thus we have as a condition $\frac{u}{a}$ = const. But the velocity of sound

$$a = \sqrt{gkRT} = \sqrt{gk \frac{R}{m} \cdot T}$$

Consequently the flow velocity must increase with $\sqrt{\frac{1}{m}}$ if the same conditions of pressure and temperature are to be obtained.

For the element of length the following is obtained

$$\frac{dx}{D} = \frac{\frac{dp}{p}}{2c_f} \cdot \frac{1}{kM^2}$$

$\frac{dx}{D}$ is thus porportional to

$$\frac{1}{c_f} = \frac{\sqrt[4]{\frac{\rho u D}{\eta}}}{0.0791}$$

$$\frac{dx}{D} \sim \sqrt[4]{\frac{\rho u}{\eta}}$$

For the same temperatures the viscosity of gases having the same atomic number, with the one exception of hydrogen, differs only slightly (for the He-A mixture the viscosity varies between $m = 4$ and $m = 40$ only by about 10 per cent). Thus one can insert with sufficient accuracy $\frac{dx}{D}$ proportional to $\sqrt[4]{\rho \cdot u}$

From the foregoing we have

$$\frac{dx}{D} \sim \sqrt[4]{m \frac{1}{\sqrt{m}}} = \sqrt[8]{m}$$

i.e., only a rather small variation in the sense that for lighter gases the tubes become somewhat shorter. One sees immediately

where the advantage of light gases is to be found. The useful output of the machine is, namely, for the same tube diameter, proportional to $\rho u c_p$. It is also proportional to

$$m \frac{1}{\sqrt{m}} \cdot \frac{1}{m} = \frac{1}{\sqrt{m}}$$

Thus for lighter gases it will be considerably greater. Let us compare, for example, two gases, the molecular weights of which are as 1:9, then the velocity of flow adopted for the lighter gas must be 3 times greater; the output for the same efficiency becomes 3 times greater and the tube lengths for the heat exchanger are reduced by about 25 per cent. The weight of the charge of gas in the machine will increase somewhat more than m, becoming for this example less than $^1/_9$.

In physics the variation in the output can also be made clear in the following manner: According to Avogadro, the unit of volume for the same pressure and same temperature contains the same number of molecules, i.e., also just as many molecule degrees of freedom. Each molecule transports the same energy. However, since we are delivering 3 times the volume, it follows that 3 times the output will also be converted.

It is, however, known that in general light gases call for a larger number of stages (in the compressors and turbines). This is a drawback of light gases. In the case of the compressor, for example, the following equation obtains for the number of stages

$$n_k = \frac{2}{k-1} \frac{\ln \delta}{\psi M^2}, \quad \ln = \text{natural logarithm}$$

in which δ represents the adiabatic temperature ratio of the compressor, ψ the pressure coefficient. Thus in our case the law applies for the compressor (and also for the turbine) that n is proportional to $\frac{1}{M^2}$ in which M = peripheral velocity/sound velocity. Since $M \sim \sqrt{m}$ for the same peripheral speed, the number of stages would follow $1/m$, becoming, in the case of the example, 9 times greater. For the turbine there is little hope of deviating therefrom, but this does not apply to the compressor. Since the peripheral velocity of the axial impellers of the compressor does not reach the limit of their tensile strength, the peripheral velocity could, without impairing the efficiency by the Mach effect, be increased and the number of stages thus reduced.

As the weight of the machinery is relatively small in the case of a highly supercharged AK-plant, the increase in the number of stages could be accepted relatively easily, especially if it proves possible to employ radial impellers which can be operated up to the highest circumferential velocities without detrimental Mach effect.

For the heater tube the same heat-transmission coefficients as for water tubing are possible.

Comparative figures are given in Table 2 to facilitate understanding of the conditions. The comparisons relate to various gases and mixtures of gas with air.

As may be noted from the foregoing there are good prospects that further development work, especially in the case of installations for special purposes, will lead to higher efficiencies and reductions in the size of the plant. These remarks of Professor Ackeret confirm the author's opinion that the present development of the closed-cycle process by no means represents the ultimate attainable.

TABLE 2 COMPARISON OF DIFFERENT GASES FOR THE AK-PROCESS[a]

Gas	Air	He + CO_2	He + CO_2	He	H_2	
Mean molecular weight	29	8	6	4	2	
Specific heat, Btu/(lb)(deg F)	0.26	0.755	0.90	1.25	3.5	
Ratio of viscosity (T = const)	1	1	1	1	0.5	
Ratio of sound velocity	1	2.1	2.4	3	3.9	
Adiabatic pressure ratio for temperature ratio	4	2.92	2.71	2.52	3.65	
Volume, per cent CO_2 (1.45)	...	10	5	..	...	
Number of stages (ratio)	1	2.8	3.5	4.8	13.5	Constant circumferential velocity
Circumferential velocity	1	1.75	1.9	2.2	3.7	Constant number of stages
Diameter (ratio)	1	0.76	0.73	0.68	0.52	
Revolutions per min	1	2.30	2.6	3.3	7.1	
Heat exchanger						
Coefficient of heat transmission	1	1.86	2.12	2.56	4.35	
Number of tubes	1	0.66	0.62	0.56	0.27	(Ratio)
Length of tubes	1	0.82	0.76	0.70	0.85	
Surface area of tubes (weight)	1	0.54	0.47	0.30	0.23	
Heater						
Coefficient of heat transmission	1	1.6	1.7	2.0	4.0	(On one side)
Number of tubes	1	0.78	0.75	0.68	0.30	
Length of tubes	1	0.82	0.79	0.74	0.85	
Surface area of tubes (weight)	1	0.64	0.59	0.50	0.25	(Ratio)

[a] Machines assumed for equal output, maximum pressure, temperature and triangles of velocity; (ϵ) = const, T = const, p = const.

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A comparison of the ideal process with AK-plant, steam turbine, and combustion-gas turbine. Size of machines and heat exchangers of AK-plant and of combustion-turbine plant. The regulation, nozzle or throttle regulation for steam turbines, temperature regulation for combustion turbines, pressure-level regulation for AK-plant. A comparison of attainable thermal efficiencies and fuel costs. Various uses of the AK-plant.

5 "The Aerodynamic Turbine for Steel Works," by C. Keller and R. Ruegg (in German), *Revue Polytechnique Suisse*, vol. 122, 1943, no. 1, p. 1.

Various layouts for producing the blast in steel works in conjunction with AK-installations. Combined blast and air heater. Calculations for various cases with particulars about the heat consumption. Comparison of the space requirements of blast installations with AK-drive and other drives.

6 "Particulars as to the Present State of Development of the Aerodynamic Thermal Prime Mover," by J. Ackeret and C. Keller

(in German), *Revue Polytechnique Suisse*, vol. 123, 1944, no. 4, p. 37.

A preliminary report as to the results obtained up to the end of 1943, with the 2000-kw AK-experimental plant of Escher Wyss. Regulating tests and measurements of the thermal efficiency.

7 "The Aerodynamic Turbine 'Escher Wyss AK-Plant,'" (in German); "The Theoretical and Practical Foundations of the AK-Process," by J. Ackeret; "Realization of the New Thermal Power Plant," by C. Keller, and "Chief Results of the Official Performance Tests," by H. Quiby, *Neue Zürcher Zeitung*, no. 140, January 24, 1945.

A short historic review concerning the development of thermal prime movers and their theoretical bases. Description of the AK-process. The realization of the process in the 2000-kw experimental plant of Escher Wyss. General description of this plant and some projects for large installations. Preliminary information concerning the results obtained from the official tests carried out by Professor Quiby in December, 1944. Published on the occasion of the demonstration of the plant in January, 1945, to officials, the press, and other interested parties.

8 "Compte-Rendu des Essais de la Turbine Aérodynamique Escher Wyss AK," by H. Quiby, (in French), *Revue Polytechnique Suisse*, vol. 125, 1945, p. 269, and no. 24, p. 279.

A detailed description of the test results obtained by Professor Quiby in December, 1944, with the 2000-kw Escher Wyss AK-plant. Description of the plant and of its regulating equipment. Measuring results in the form of detailed tables, i.e., for four different loads on the machine. Course of the thermal efficiency in relation to the load. Behavior of the governing gear in practical operation. Diagrams relative to the speed and course of pressure in the plant under sudden load fluctuations, and when throwing load off.

9 "The Aerodynamic Turbine Escher Wyss 'AK-Plant,'" by C. Keller (in German), *Elektrizitaetsverwertung—Electrical Service*, vol. 20, 1945–1946, nos. 1–3, April–June, 1945, p. 81.

Reprint of a lecture given by Dr. C. Keller before a meeting called by the Swiss Hydraulic Association in March, 1945, in Zurich for discussing questions relating to hydraulic power and coal.

Description of the AK-process and of the experimental plant. Comparison with other machines. Projects for large installations, efficiency and Sankey diagrams according to the official measurements made on the experimental plant.

10 "The Aerodynamic Thermal Prime Mover, 'Escher Wyss AK-Plant,'" by F. Salzmann (in German), *Schweizerische Technische Zeitschrift*, 1945, no. 32, p. 413, and no. 33, p. 425.

The thermodynamic phenomena in the circuit and the basis for calculations. Pressure-volume diagrams. Thermal efficiencies of the ideal machine with consideration of losses, with and without intermediate heating. The Escher Wyss experimental plant and particulars about the test results. The uses of the plant.

11 "Some Remarks Concerning the Basis and Construction of the Thermal Power Plant Employing a Closed Air Circuit, 'Escher Wyss AK-Plant,'" by C. Keller (in German), *Schweizer Archiv. Fuer Angewandte Wissenschaft und Technik*, vol. 11, 1945, p. 225.

The theoretical basis and test results. Attainable efficiencies. Some general projects for large installations with single and double heating.

12 "The Theory and Performance of Axial-Flow Fans," by C. Keller and L. S. Marks, McGraw-Hill Book Company, Inc., New York, N. Y., 1937.

Theory and performance for use of fan designers.

13 "Performance Tests of 2000-Kw Closed-Cycle Unit. First Summary of Official Tests in the United States," by S. A. Tucker *Power*, August, 1945, p. 80.

Discussion

J. H. Anderson.[4] The author has made an excellent presentation of an outstanding achievement in engineering. Although time and further development will be required to determine whether the AK-system is to become an economic success, the author and his associates deserve only the highest praise for the ingenuity and perseverance required to turn this development into an engineering accomplishment of such obvious importance.

While this power plant could be compared to any prime mover, it seems most logical to compare it to a modern steam plant, for the simple reasons that both are externally fired and both can be built in large capacities. By the same logic, the open-cycle gas turbine most nearly compares with the Diesel power plant in its application and place in the economic structure.

It is, as the author has already pointed out, almost impossible to make an economic comparison between the AK-plant and a steam plant at the present stage of progress, but at the same time anyone will admit that this must be done eventually. With this in mind, it may be worth while to set down a few simple comparisons between the two types of plant.

Starting at the power unit, we have an air turbine in the AK-plant as compared with the steam turbine in the steam plant. Basically, the air turbine should be more efficient and cheaper for two reasons, i.e., there is no problem of handling liquids with attendant corrosion, and the volume ratio of expansion is far lower in the air turbine. Both of these considerations help to make a simpler and cheaper design possible, which should at least balance the problem of handling higher temperatures in the air turbine.

The air cooler and intercoolers in the AK-plant correspond to the steam condenser in the steam plant. Here the steam condenser is obviously much more expensive because more heat must be discharged at a much lower temperature difference. Also, the air-removal equipment and condensate pump are additional accessories required over and above those needed for the air coolers.

The compressor in the AK-plant corresponds to the boiler feed pump in the steam plant. Here the difference would appear to be in favor of the steam plant, although this must be subjected to thorough analysis.

The boiler, economizer, feedwater heaters, and superheater in the steam plant correspond to the air heater and recuperator in the AK-system. This is probably the largest question mark in a comparison of the two systems. In the air heater we have conditions roughly equivalent to those in the steam superheater. For this reason we have a higher average temperature difference to work with in the boiler. On the other hand, we have a lower heat input in the air heater. The recuperator is undoubtedly larger and more costly than the corresponding feedwater heaters. All in all, it is probable that the heaters in the AK-plant cost more than the corresponding heaters in the steam plant. Comments on this point by boiler manufacturers should be of great interest.

In the matter of auxiliaries, there would seem to be no question that the AK-plant is simpler and has less of them than the steam plant. This would be the case simply because in one case we have a liquid-gas plant, and in the other we have a pure gas plant.

Regardless of the present state of development, there can be no question but that the AK-plant is basically simpler to control than is the steam plant. Here again we have the simple reason that a gas is used throughout the system, instead of both a liquid and a gas.

It is difficult to make predictions on maintenance costs. However, it is probably safe to say that more machinery trouble is encountered from corrosion, wear, and solid deposits in any system where both liquid and gas are present than in a system where gas alone is present, and this is especially true in any system where variable temperatures are present. The problem of deposits on steam-turbine blades is a serious one which cannot be overlooked. While it is granted that the AK-system must operate with some higher metal temperatures than the steam system, it is probable that this will not cost as much in maintenance as other troubles in the steam plant.

It is perhaps foolish to pass judgment on the basis of the simple comparison here given, but even such a comparison should be enough to show that the AK-system deserves most

[4] Research Engineer, Ingersoll Rand Company, Phillipsburg, N. J. Mem. A.S.M.E.

careful consideration as an immediate competitor of the steam power plant.

The author has quite properly pointed out that the shaft seals are a critical component of this plant. This is obvious when one considers that there are probably 10 seals under high pressure. The ingenious method of using an oil seal at the lowest pressure in the system should make the seal losses roughly equivalent to those of an open-gas-turbine system. It would be interesting to learn what percentage of the losses was ascribed to the seals, and what clearances had to be maintained in the labyrinth packing to obtain this value of the losses.

The author has mentioned that the efficiency of an axial compressor dropped from 86 to 83 per cent during only 12 hr of operation in their shops. This does not seem to correspond fully with the experience of some others. For this reason it would be of value to have a further explanation as to how this drop in efficiency could be accounted for.

F. T. Hague.[5] The author's and Mr. Gygi's[6] visit to this country is both welcome and significant, in that it promises to focus increased attention on the study of gas-turbine power plants. The presentation of the paper, disclosing test results on a 2000-kw closed-type-cycle gas-turbine power plant, is evidence that the gas turbine for primary power generation is rapidly developing toward maturity.

It is becoming increasingly apparent that the complete gas-turbine power plant, which was a newcomer in this country 3 years ago, is making giant strides in being reduced to practice in several important lines of endeavor. Military releases have identified it as the fighting aircraft power plant of the future. Commercial interests view the prototype tests of propeller-drive gas turbines as a further boon to commercial aviation. In the locomotive field, powerful, compact gas-turbine power plants are currently being built. A high-efficiency open-cycle type of gas-turbine power plant has been demonstrated within the year, giving promise of application in fields of industrial power generation, and ship propulsion. The gas-turbine plant of which Escher Wyss has built a prototype, is the first reduction to practice of a design from which coal-burning central-station-type units of from 10 to 50,000-kw capacity may ultimately become a reality.

The author and his colleagues deserve high commendation for their engineering foresight in taking steps to reduce this type of gas-turbine power plant to practice several years before it was being given serious consideration elsewhere. It is only fitting to acknowledge that this foresight has given these engineers several years' time in which to study the theoretical and practical problems connected with this development. Their viewpoint on its ultimate accomplishment may well be more realistic than those who have been considering it for a shorter length of time. The fact that 98 per cent of Switzerland's power generation is water power presumably limited their combustion research with pulverized coal.

The proposed closed-cycle gas-turbine power plant is simple for engineers to understand who are already familiar with the open-cycle type of system. This closed-cycle system differs from open-cycle systems in one major respect; the physical size of the rotating and heat-exchange machinery is decreased as the system pressure is elevated. It is this characteristic of the closed cycle which makes it possible to build turbines and compressors for a 50,000-kw unit of smaller physical size than for a 5000-kw open-cycle unit. The closed-cycle type of system has its efficiency level affected in exactly the same manner as the open cycle with respect to any changes in maximum temperatures, compression ratio, intercooling, reheating or regeneration Like all good things, it is basically simple. This in no way detracts from the brilliance of the initial conception.

There are metallurgical limitations in the building of any high-temperature gas turbine. These are definitely less serious when the physical size of the parts is reduced. It may not be too much of an estimate to say that the metallurgical problems of a 25,000-kw closed-cycle plant will be comparable to those of a 5000-kw open-cycle plant. This is a favorable characteristic.

The practicability of the closed-cycle system may ultimately be determined by the problems associated with getting the heat from the fuel into the system. The author has elected to introduce the heat through an externally fired air heater, thus offering the possibility of using the cheapest grades of liquid fuels or pulverized coal. Such an air heater requires extensive use of high-priced stainless steel for temperatures above 1000 F. The precautions which must be taken to protect the tubes of such a gas-to-air heater from overheating, pose a difficult set of problems to the air-heater builder in designing an air heater comparable in both size and cost to a steam generator of equal plant capacity. Engineers interested in the development of a gas-turbine power plant will wait with interest to see how our air-heater manufacturers can meet these requirements at an economic cost when burning pulverized coal. Much, if not all, of the ultimate part that will be played by this cycle in primary power generation may well rest in the satisfactory solution of this air-heater problem when burning pulverized coal.

It is not to be expected that this and possibly other problems associated with this new form of gas-turbine power plant can be solved overnight. The problems involved in developing the steam cycle to its present level of performance and reliability required a long time for solution. We should, with equal consideration, concede that problems in connection with this new system which, at first glance, look to be formidable, may after further study actually yield to research and development. The engineering profession is learning to be optimistic regarding the possibilities of development of the gas turbine, and that optimism should be extended to studies of this new type of system.

M. L. Ireland, Jr.[7] The closed-cycle process at present appears to be the most practicable means to obtain single plant outputs in excess of about 6000 hp, and the author's company has shown unusual foresight in concentrating its efforts on this design.

For central-station applications this cycle offers the prospect of thermal efficiencies exceeding those now attainable with the most modern steam plants and this may well warrant the high development costs which must be incurred.

The prospect for closed-cycle marine gas-turbine power plants in excess of about 6000 hp appear to be less favorable for strictly commercial applications. Some idea of the initial cost and the operating and maintenance problems of such a plant can be gained from a comparison of the major components with the corresponding items of a steam-turbine plant for superheater conditions of 700 lb 850 F, as follows:

1 From the author's Fig. 35 we find that two air turbines are required which, though smaller in size, are probably more costly than the cross-compound steam-turbine unit because of the higher-temperature materials required.

2 Four air-circuit compressors and three intercoolers are required which correspond in function to the condensing and feed-water heating and pumping units of the steam plant. The cost of

[5] Assistant to Vice-President, Westinghouse Electric Corporation. Mem. A.S.M.E.

[6] Managing Director, Escher Wyss Engineering, Zurich, Switzerland.

[7] Engineering Technical Division, Newport News Shipbuilding and Dry Dock Co., Newport News, Va. Mem. A.S.M.E.

the compressors will probably more than offset the reduction in heat-exchange equipment, even when the increased circulating-water requirements of the steam plant are taken into account.

3 Even when pressure charging is employed, the air heater and regenerator when taken together appear to about equal the total heating surface, including air heaters of about 2 sq ft per hp, which is required for a steam plant of this size. Judging from the percentage of high-temperature materials required for the air heater, the cost will compare quite unfavorably.

4 The pressure-charging set for the air heater is understood to be necessary to reduce the dimensions of this unit, but this is felt to introduce undesirable complications for a marine unit which already contains so many novel features. In particular, it is believed that the pressure-fired boiler will be more sensitive to pressure and flow changes in the air circuit with consequent increased risk of damage to tubes subject to high thermal loading on the fuel-gas side.

5 There remain the high- and low-pressure accumulators, the control unit and the starting motor, which may be compared with the throttle and combustion-control elements of the steam plant.

When itemized in this way it is difficult to believe that the initial cost of the closed-cycle marine unit will compare at all favorably with a steam-turbine plant for, say, 700 psi 850 F at the superheater. The operating requirements appear to be quite similar, as in both cases primary control is applied to the weight flow of the working medium and the fuel control follows with a certain inevitable time lag. Considering the relative number of major rotating elements and the fact that combustion gases must pass over heat-exchanger tubes in both cases, there does not appear to be any evident advantage as regards maintenance.

The advantage in thermal efficiency over the conditions outlined for a steam plant is large, being approximately 15 per cent, but there remains a question, in this writer's mind at least, whether this will not be absorbed by increased capital charges and maintenance expense.

A substantial saving in weight is indicated which will be of importance for vessels carrying heavy cargoes. However, the fore-and-aft length of the closed-cycle unit in Figs. 35 and 36 is about 5 ft longer than the machinery compartment for a standard C-3 steam-turbine-driven cargo ship with 8500 shp. A considerable portion of the wing spaces abreast of the main plant will be required for generators, pumps, compressors and other ship's machinery and would be of little value for cargo space in any case. Therefore, in a cubic-capacity trade, the closed-cycle plant would probably be at a slight disadvantage.

In general it would appear that these factors can be considered only when a specific vessel and trade are under consideration and therefore should not be cited as intrinsic advantages or disadvantages of the closed-cycle plant as compared to a steam-turbine unit.

It is encouraging that the development of the controllable-pitch propeller has also been undertaken by the author's company and it is hoped that he will shed some light on one feature of this equipment which has given much cause for concern. During the process of reversing the propeller pitch it is necessary to move the blades back through the angle of zero lift, while the ship is still making headway. For a brief interval, while the blades are not fully reversed, they would appear to be receiving energy to drive the shaft which would result in an acceleration of the shaft speed. If the pitch control should become jammed in this position it would appear that dangerous overspeeding of the unit might occur. This situation would appear to be more critical for a gas-turbine plant than with Diesel engines which are believed to be the type of power plant to which the controllable-pitch propeller has so far been mainly applied.

Joseph Kaye.[8] This paper presenting a statement of the status of the closed-cycle gas-turbine power plant, brings to mind some interesting questions which it does not answer directly. (*a*) Does the closed-cycle gas-turbine plant possess any marked improvement in efficiency in comparison with other power plants of a similar nature, such as a steam power plant? (*b*) Is the size of the closed-cycle gas-turbine plant smaller than that of a comparable steam plant?

The closed-cycle gas turbine might also be compared with the more widely known combustion-gas turbine in regard to fuel consumption and size. However, such a comparison would be inconclusive at best, since different fuels are used at present for the two plants, and since the primary functions of the two plants are not similar. It should be noted that the closed-cycle gas turbine and the steam plant utilize the same fuels, perform the same primary functions, and possess parallel component parts.

For purposes of comparison, the following two power plants may be selected:

(*a*) The proposed closed-cycle gas turbine, shown in Fig. 10 of the paper, involves four compressors with three intercoolers, two turbines with two heaters, and a regenerator with an assumed effectiveness of 90 per cent. In addition a precooler is necessary at the entrance to the first compressor. The inlet temperature of each turbine is 1200 F, and from the data given in Fig. 10 the turbine efficiency is about 93 per cent and the compressor efficiency is about 90 per cent.

(*b*) The second power unit is the customary condensing steam power plant with one reheater present. For the purpose of this comparison the temperatures and pressures at each turbine inlet are taken as those in Fig. 10 of the paper. The condenser is assumed to operate at 1 in. Hg, and the feedwater pump to have an efficiency of 90 per cent. Regenerative feedwater heaters are omitted for simplicity; their effect on efficiency can be easily estimated from known data. The pressure drops in the boiler, superheater, and reheater are equal to the corresponding pressure drops in the high-pressure side of the regenerator, first heater, and second heater, respectively, as given in Fig. 10.

The efficiency of the proposed closed-cycle gas turbine in (*a*) is about 39 per cent allowing for external losses.

The efficiency of the steam power plant in (*b*) is about 38 per cent for turbine efficiencies of 93 per cent, and is about 35 per cent for turbine efficiencies of 85 per cent, allowing in each case for external losses. These efficiencies could be increased by 10 per cent through the introduction of several regenerative feedwater heaters. The efficiency could also be increased somewhat by the readjustment of the pressure ratios for the two turbines to yield the maximum steam cycle efficiency for the same temperatures.

On the basis of efficiency, the proposed closed-cycle gas turbine in (*a*) represents no marked improvement over the comparable steam plant in (*b*).

The power units in (*a*) and (*b*) may be compared on the basis of size. Consider first the turbines and compressors of (*a*) and the turbines and feed pump of (*b*). The turbines of (*a*) must deliver several times the power of the turbines in (*b*) for the same net power of the complete unit, since the power consumption of the feed pump in the steam unit in (*b*) is negligible compared to the power required to drive the compressors of the gas-turbine unit in (*a*). In other words, for a net power of 2000 hp, the turbines in the steam plant in (*b*) will deliver slightly more than 2000 hp, whereas the turbines of the unit in (*a*) must deliver about 4000 hp, and the compressors will absorb about 2000 hp. Even if the specific weight of the turbines in (*a*) is less than that for the turbines in the steam unit in (*b*), it is apparent that the steam

[8] Assistant Professor of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Jun. A.S.M.E.

plant has a decided advantage with respect to size of its prime movers.

Consider next the size of the heat-transfer units of the closed-cycle gas turbine in (*a*) and of the steam plant in (*b*). Both units have a heater in which heat is transferred to the working fluid at high temperatures. Both units have a cooler for rejecting heat from the working fluid; in the case of unit (*a*), this cooler comprises the precooler and intercoolers, while for unit (*b*), it is the condenser. Moreover, the gas turbine in (*a*) has an additional piece of heat-transfer equipment, the regenerator.

For equal heat flows and similar constructional details, the size of the heaters and of the coolers will be determined mainly by the major resistances to heat transfer. Under these conditions the size of the heater of the gas turbine in (*a*) will be larger than that of the steam plant in (*b*) for several reasons: The thermal resistance of the high-pressure gas of the heater in (*a*) will be larger than the resistance of the high-pressure vaporizing fluid in the steam plant in (*b*). Then, the use of high-pressure combustion to reduce the thermal resistance on the flue-gas side of the heater is applicable to the steam power unit in (*b*) as well as to the gas turbine in (*a*). For equal heat flows and similar constructional details the combined size of the precooler and intercoolers of the gas turbine in (*a*) will be larger than the size of the condenser of the steam unit in (*b*). These heat-transfer units use water as the cooling agent, and they will have about the same thermal resistance on the cooling-water side. However, the thermal resistance of the low-pressure gas in (*a*) will be considerably greater than the resistance of the condensing steam in (*b*).

In conclusion, if the large size of the regenerator of the closed-cycle gas turbine in (*a*) is considered, it is apparent that the over-all size of the equipment for the closed-cycle gas turbine in (*a*) will be greater than the size of the steam plant in (*b*) without a compensating gain in efficiency.

ALF LYSHOLM.[9] In the writer's opinion the high thermal efficiency is to a great extent due to the perfect aerodynamic design, especially of turbine and compressors, as well as to the high-efficiency 90 per cent regenerator, which has counterbalanced the inherent unavoidable losses of the closed cycle. These losses are as follows:

1 Losses in the externally fired air heater.

2 Aftercooler losses, including the loss caused by a higher inlet-air temperature of the first compressor, as compared to the open cycle. In cases where the cooling-water temperature is lower than the air temperature a precooler could be used with advantage for the open cycle.

3 "Feed"-pump losses.

The author states that the inlet volume for the open cycle is many times higher than that of the steam-turbine cycle. That is correct for an open simple gas-turbine cycle, but not for the reheat intercool cycle using the same or somewhat lower maximum pressure than that of the closed cycle.

In Table 3 of this discussion a comparison is given of the inlet volume of different gas-turbine cycles, as well as steam cycles, taking the author's data for the 8000-shp unit as a basis.

The table shows that the inlet volume of the open cycle, using the same maximum pressure as the AK-cycle, is only about 20 per cent higher than the combustion air volume of a steam-turbine plant operating at maximum load. As higher velocities in the ducts may be used for a gas turbine, the ducts will be of about the same dimensions as those of the steam plant.

Compared with the closed cycle the efficiency of the multiple-reheat-intercool cycle will be considerably higher, amounting to 10 to 15 per cent for the example just given. On account of the low air rate this combination is ideal for very large gas turbines.

As regards coal-firing, it should be noted that the open cycle also may be fitted with an external air heater if that should be required. In this case the additional heat losses due to combustion air amount to only a few per cent, as hot air from the gas-turbine exhaust may be used as combustion air in a "tail-end" boiler.

TABLE 3 COMPARISON OF VARIOUS GAS-TURBINE CYCLES[a]

Cycle	Air pressure, psi	Pressure ratio	Air rate, lb per hphr	Compressor-air-inlet volume, cfm	Combustion-air-inlet volume, cfm	Thermal efficiency, per cent
AK-double heat.....	600/60	10	33	15000	14000	33
Reheat intercool, 4-3-0.85..............	600/14.7	41	18	31000	.....	38
Reheat intercool, 3-2-0.85..............	250/14.7	17	24	41000	.....	37
Steam maximum load	14.7	..	..	.....	26000	17
Steam, normal load..	14.7	..	..	.....	19000	23

[a] The data in this table have been computed with a turbine efficiency of 88 per cent, and a compressor efficiency of 83 per cent.

FREDERICK NETTEL.[10] The author mentions that the double-isotherm cycle (in which we recognize the Ericsson cycle) is in its theoretical form equivalent to the Carnot cycle, and seems to claim on this basis some sort of superiority for it. It must be kept in mind that the approach to the Carnot cycle is realized by assuming not only ideal compression and expansion, but also 100 per cent recuperation.

Under the same assumption, however, even the simple Brayton cycle approaches Carnot efficiency as the pressure ratio decreases. For thermodynamical purposes Fig. 3 can be interpreted as agglomeration of three Brayton cycles.

Any basic superiority of the double-isotherm cycle appears doubtful from this viewpoint, apart from quite fundamental additional considerations which enter the question as soon as even slight deviations from 100 per cent efficiencies are contemplated for compression, expansion and recuperation.

On the basis of fundamental thermodynamic research (the results of which are not yet released) it must be said that it is even doubtful whether any of the two cycles is particularly suited to approach Carnot efficiency with a minimum of heat-transfer surfaces, and in particular it must be doubted whether the intentional complete omission of any adiabatic compression is always helpful. Let us consider only the last stage of an isothermal compression. There we carry away compression heat by cooling, only to replace it immediately thereafter by heating. This is all right as long as the heating is obtained, so to speak, free of charge by 100 per cent recuperation, but if we have only 90 or 80, or 70 per cent recuperation it is sheer waste of heat. This is only one of the reasons why the Ericsson cycle is not the ideal to

[9] Consulting Engineer, Stockholm, Sweden.

[10] Consulting Engineer, Manhasset, Long Island, N. Y.

strive for in practical plants. Approaching a cycle which is best under ideal conditions does not give the plant that is best under conditions prevailing in practice, namely, with finite efficiencies of the components.

The interrelations between the method of intercooling, reheating, and recuperation, if applied simultaneously, are much more complex than it would appear from the presentation of the AK cycle, and that is the reason why the writer arrives at appreciably different results in setting up optimum conditions for open as well as closed cycles.

The author speaks of the supercharged closed cycle as "an important new feature" and mentions as its "chief characteristic" the changing of the density of the working fluid in accordance with the load to be furnished. Without wishing in the least to detract from the fact that Ackeret and Keller have introduced this principle for air "turbines," we must face history by mentioning that the closed cycle and density regulation were actually anticipated for air "engines" decades ago. However, this is not the place to go into details, and that does not change the fact that the application of this principle in a properly modified form has outstanding promise for high-power air turbines. In commenting on the merits of the AK cycle, its demerits and limitations need also to be given due consideration.

The utilization of coal as fuel is imperative if the gas or air turbine is ever to play an important role as a prime mover. The author points to the radical decrease in the size of heat exchangers and the favorable heat transfer to highly compressed air. He touches on the necessity to preheat the combustion air in the air heater furnace.[11] Since the stack gases from the furnace and the combustion air have to exchange heat near atmospheric pressure, this air heater does not profit in any way from the supercharging of the power cycle, and the necessary heat-transfer surface will be of the same magnitude as that of air heaters used in steam boilers. We know that those air heaters have surfaces which often are multiples of the boiler heating surfaces proper, and we know too that they determine the furnace efficiency, not only in steam power plants, but equally in air-turbine power plants of the AK-type. However, this is not necessarily the case in other closed or semi-closed plants of a modified layout and hook-up.

With reference to the section of the paper on operating cycles of the AK process: Two intermediate coolers and a pressure ratio of about 3 to 4 are mentioned as the most favorable for the simplest air-turbine plants. The author does not give any basis for this statement. Actually the best pressure ratio varies widely with recuperator effectiveness, top temperature at turbine inlet, turbine and compressor efficiencies, etc.

In practice an "oversize" heat exchanger can be made to cover up nearly any deviation from optimum layout, and this is what appears to have been done in the 2000-kw experimental AK plant, where a recuperator of close to 90 per cent effectiveness is being used. While a recuperator of 75 per cent effectiveness is already pretty large, one of almost 90 per cent is definitely oversize.

Now the author may reply that his recuperator is as such much smaller than those required in open gas-turbine plants due to the much better heat transfer resulting from supercharging. However, this would be beside the point if it can be shown that an equal or better efficiency can be reached in a similar plant of modified layout with smaller transfer surfaces; and that can be shown.

Actually, in a similar manner as indicated by Soderberg-Smith for open cycles, the optimum pressure ratios increase with increasing refinements (intercooling and reheating) but fall with rising effectiveness of the recuperator, and there is no short cut to fixing a ratio at 3 to 4 as "most favorable."

All Sankey diagrams show high furnace efficiencies obtained by air preheaters for the combustion air, but the text gives no indication of their size and the degree of air preheat.

In view of the foregoing, it is difficult to understand Fig. 9 fully. In the "Index of Figures" (referring to Fig. 9) a minor error (due probably to translation) has crept in. Only a small Δt_A = 18 deg F in combination with a large ϵ_B = 10 per cent can lead to the same curve 2 as a larger Δt_B = 36 deg F in combination with a smaller ϵ_A = 5 per cent. The same holds good for curve 3.

With reference to air heaters: Under this section the influence of preheated combustion air is stated "to lead to an increase in the furnace temperature," and that in plants burning pulverized coal "such preheating of the secondary air is desirable," since it permits a reduction in the size of the combustion chamber.

Every reason for favoring air preheating given is sound, but the most important fact, namely, that the AK plant would badly disappoint in efficiency without such a very large air preheater, may escape notice as the author proposes "not to deal further herein with the details of coal-fired air heaters."

There can be little doubt that supercharging of the air heater on the gas side has great merits for marine plants. However, the problem of regulation of the supercharging set and the problem of very low loads, for example in naval plants, is one of the most complicated that awaits solution. Probably the author did not go into these details as outside the scope of a general paper. It is in any case necessary for us to realize that supercharging of air heaters in closed or semiclosed air-turbine plants deserves closest attention in the future.

Do the figures 1.5 to 3 sq ft of heat-transfer surface per kw include the air preheaters for the combustion air?

Governing: The size of the air accumulators for a 50,000-kw plant would be interesting to know if we consider the possible tripping of the generator circuit breaker twice or three times within, say, $^1/_2$ hr.

The writer will conclude with some remarks regarding the efficiency curves in Fig. 7. According to the author, without altering the temperatures the efficiency of the plant should be almost equally high at part loads as at full load.

The curves, while comparing very well indeed with those of other prime movers, will necessarily show a steady drop with the load. The reason is simple enough; if at full load the tube material in the air heater is thermally fully utilized, a certain safe limiting wall temperature is reached. At lower loads the air density drops and with it the heat transfer from wall to air, and if we try now to maintain the top heating temperature the wall temperature would rise beyond a safe limit. Since that is inadmissible, a lowering of the top temperature becomes imperative, resulting in a somewhat lower efficiency.

While curve (*a*) is shown as a horizontal, this would seem attainable only if at full load the tube material would not be thermally fully utilized, thus admitting higher wall temperatures at partial loads. Whether or not a plant should be designed in this way will depend primarily on the load diversity factor of the plant during operation.

J. K. Salisbury.[12] After one has mastered the prerequisite elements of engineering, the problem of design of power-generating equipment becomes primarily one of economics. It is well recognized that any power plant consists primarily of a means for supplying heat to a system, and an accompanying means for rejecting less heat than is supplied. The difference between the

[11] Without, however, mentioning the conditions under which this must be done.

[12] Turbine Generator Engineering Division, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

heat supplied and the heat rejected is equal to the power generated, according to the fundamental laws of thermodynamics. In many cases the supplying of heat to a system and the rejection of heat from a system requires the use of indirect heat-transfer surfaces. Both the weight and the quality, as well as the degree of fabrication of the materials which go into these heat-transfer surfaces determine to a large extent the cost of the power-generating system.

Typical exceptions to the generalization that power plants require heat-transfer surfaces are the open-cycle gas turbine and the Diesel engine. In these plants heat-transfer surface is not required because the heat generated within the working fluid is transferred by direct contact, thus eliminating an expensive type of surface, which must be suitable for operation at high temperatures, for high efficiency. Both of these power plants reject heat to the atmosphere through the medium of the exhaust gases. In the case of the Diesel engine, an appreciable portion is also rejected to the cooling water. Obviously, both of these plants thus eliminate a large item of expense.

Consider the general proposition of the transfer of heat through an indirect heating surface in the type of power plant which requires it. According to the well-known equation of heat transfer, the heat transferred Q, is given by the expression

$$Q = UA\Delta T$$

The last two factors in this equation represent material for interesting speculation. By increasing the value of A we can decrease the value of ΔT, with constant heat input Q. When we increase the surface area A, we increase the cost of the power plant; at the same time we reduce the value of the temperature difference ΔT; according to the second law of thermodynamics this improves the efficiency of the power plant. It is readily apparent that we may exchange cost of the power plant for efficiency of the power plant; when the cost increases the efficiency improves, when the cost decreases the efficiency becomes poorer. This principle is particularly apparent to any student of gas-turbine cycles, although it is also applicable to steam power plants.

The first factor in the heat-transfer equation, the heat-transfer coefficient U, is ordinarily not considered to be susceptible of much improvement. However, the author has, in the design of the plant described in the paper, indicated a method whereby U may be increased without appreciable direct cost. By increase of the pressure level in the system the heat-transfer coefficient is increased almost in direct proportion, thus reducing the value of A required for a given rating and efficiency. In other words, the use of the pressure cycle in which heat-transfer surfaces in regenerators are reduced by the use of high pressure on both sides of the surface, is considered to be of importance in the gas-turbine field. This principle permits realization of high efficiency without the usual accompanying high cost of the regenerator.

The engineering of the plant described in the paper indicates an acute awareness of the factors which are effective in yielding a high-efficiency power plant. The engineering of this plant cannot be questioned because in the first place the power plant has successfully performed, and in the second place the plant has produced a high efficiency. The author and the Escher Wyss Company are to be sincerely congratulated on their achievement. The advantages of this plant, in order of importance, seem to the writer to be as follows:

1 Control of output by variation of density, thus maintaining constant internal efficiency. (It is obvious that there is no inherent advantage in efficiency at a given load due to the use of the pressure cycle.)

2 The probable ability of the plant to burn coal.

3 The reduction in heating-surface cost per kilowatt.

4 The use of clean air in the circuit, avoiding decrease of compressor and turbine efficiency with use, and justifying the use of a larger regenerator.

5 The possibility of building units in larger ratings.

Although the engineering of this plant cannot be questioned, the economic situation does appear to be open to question, especially in the United States. It is recognized that any gas-turbine plant is handicapped by the requirement of a compressor rated at about $^2/_3$ of the total gross power output of the turbine. Thus the sum of the ratings of the compressor and turbine in nearly any gas-turbine plant is equal to about 4 or 5 times the net output. Intrinsically therefore the gas-turbine power plant uses machinery which, on the basis of its rating, should cost several times that of the turbine in a steam plant of the same rating. In the open-cycle gas-turbine plant this higher cost is partially offset by the elimination of expensive heat-transfer surface such as the boiler.

The closed-cycle plant suffers the same disadvantage, with respect to the cost of the rotating machinery, as the open-cycle plant and, in addition, requires the use of expensive indirect heat-transfer surface in the air heater. In the writer's opinion it is highly questionable whether this air heater, or one of similar capabilities, could be built in the United States for less than $75 per kw of plant rating. This price, when added to the higher price of the rotating machinery and regenerating equipment, would seem to make the total cost of a power plant of this type considerably higher than the usual price for steam power plants in the United States. The apparently higher fixed charges on such a plant would mitigate against its economic application in either industrial or central station plants, particularly the former, where the fixed charges are usually taken to be a higher percentage than in the central-station industry.

Even with the higher fixed charges, which in our judgment would obtain, such a plant might be acceptable, especially where the fuel cost is high, if it were impossible to build a plant with this efficiency in any other manner. Our studies, however, indicate that with equal ease a 2000-kw steam plant can be designed having an efficiency equal to that of the Escher Wyss closed-cycle plant. Such a steam plant would be designed for a somewhat higher pressure than the experimental plant operated in Switzerland, but for a lower temperature, to compensate for the higher pressure; the material requirements would be no more stringent than in the author's plant. The inlet steam conditions would be 600 psig 1200 F, and the exhaust pressure 0.66 in. Hg.

This exhaust pressure, according to our estimate, could be obtained with the temperature of the cooling water used in the Escher Wyss test plant. Using a turbine efficiency which includes some margin, a good feedwater-heating cycle, 5 per cent auxiliary power, and 92 per cent dry-boiler efficiency (equivalent to about 86 per cent on the usual basis), the thermal efficiency of the steam plant was found to be 30.4 per cent. While on first thought the superheating of steam to 1200 F seems to be difficult of accomplishment, it must be realized that in this country steam superheaters have already been built with an outlet temperature of 1400 F, for special applications. Furthermore, the quantity of heat which must be put into the superheated steam at high temperature, for a unit of energy output at the generator terminals, is considerably less, as will be shown.

The degree of severity of the metallurgical problems in the respective plants is described quite clearly by a statement of the relative quantity of heat put into the working fluid between given temperature limits, per unit of net energy output. One interesting way in which this may be done is to express the energy in-

put in units of kilowatt-hours, in order to make this energy directly comparable with the output of the generator. Since modern metallurgy does not recognize temperatures below 900 F as a problem, these temperatures will not be considered. In Table 4 of this discussion is included a mercury-steam plant, in which a mercury-vapor cycle is superposed on the 600-psig 1200 F steam cycle. While the writer's company is not prepared to build such a plant today, it is of academic interest in that it indicates the considerably smaller quantity of heat supplied at high temperatures in this cycle, despite the very high thermal efficiency of the plant.

TABLE 4 COMPARISON OF QUANTITY OF HEAT SUPPLIED AT VARIOUS TEMPERATURES PER UNIT ENERGY OUTPUT

(Kwhr of heat supplied per kwhr at generator terminals)

Temperature range, deg F	Escher-Wyss aerodynamic turbine plant	600 psig − 1200 F steam plant	500 psig − 1200 F mercury superposed on 600 psig −1200 F steam plant
900–1000	0.578	0.131	0.119
1000–1100	0.584	0.132	0.119
1100–1200	0.590	0.131	1.464
1200–1268	0.405	0	0
Total, above 900	2.16	0.394	1.70
Plant thermal efficiency, per cent	30.5	30.4	37.8

This table is of great interest because it indicates that in a steam plant, designed for the stated steam conditions, only about $^{1}/_{5}$ as much heat per net kilowatt of output is required to be transferred at temperatures above 900 F. Even in the mercury-steam plant, which has a thermal efficiency considerably better than that of the closed-cycle plant, the heat transferred at temperatures above 900 F is considerably less, although admittedly a large proportion of the heat is transferred at 1200 F. This heat transfer, however, occurs from gas to boiling liquid, so that the metal temperatures approximate the boiling-liquid temperatures.

The writer would like to ask the author several detailed questions. In view of the fact that the average steam plant has a total heating surface in the boiler and the condenser of about 2 to 2.5 sq ft per kw of plant rating, what is the corresponding figure for the Escher Wyss plant, including regenerator, air heater, combustion-air heater, intercoolers, and precooler? Since, according to previously published information, the combined pressure drop on the high-pressure side of both the regenerator and air heater of the test plant was only 3.5 psi, how is this reconciled with the 10 to 15 psi pressure drop mentioned by the author for the air heater only? Published figures indicate that the recirculating-fan power in the test plant was only 2 kw, or 0.1 per cent of the net plant output. American experience in this connection has indicated that recirculating-fan powers are usually very much higher than this percentage. Will the author undertake an explanation of how such low fan power was achieved? Calculations by the writer indicate that the mechanical efficiency of the rotating machinery, based on the net generator output, is approximately 86 per cent. Even when all of the mechanical losses in the turbines are charged against the sum of the powers of both the turbines and compressors, the mechanical efficiency is only 97.4 per cent, a rather low value. Will the author state whether this low value is due to the oil seal used on the shaft ends or whether there is some other reason for the high losses?

Author's Closure

General. It is not an easy matter for the author to answer the many questions posed in the long discussions of this paper in as exhaustive a manner as he would like to do. The principal reason for this is the fact that this new system of gas turbines is actually still at the beginning of its practical development. Therefore, not all of the elements have found their final form. The field of application is very large and, always considering the principle of the high-density circuit, many solutions are possible. It is impossible to give definite and general answers, for example, for the best layout or for prices. They may vary widely with given conditions, such as cooling-water temperature or supply, air temperature, efficiencies asked for, necessary weight, etc.

A crew of engineers has worked for more than 10 years and not under favorable conditions, on this project to bring it to its present state, namely, actual installations in industrial plants. The deeper we went into details in our studies, the more we found, not only from the technical and physical but also from the economical point of view, that the closed-cycle power unit offers many operating advantages as well as higher efficiency, as compared with steam turbines and open-cycle combustion gas turbines in many applications, especially for ship propulsion and stationary plants.

It is an honor and a pleasure for us to see what great interest has been created by our visit to America, and in the discussion with many famous scientists and engineers. These discussions developed many new and helpful ideas for us and we derived some very useful data, especially from the metallurgists. All this encourages us to an optimistic outlook regarding broader realization of our objectives. We are much impressed and thankful for the open-minded criticism.

Before answering the several items of discussion in detail, the author would like to make a few general remarks which hold for many questions and allusions common to different discussers. It will be recalled that this conference was meant to give a general survey of what has been accomplished by the author's company during the war and to explain the physical base of the cycle. Naturally, we have constantly kept in mind the matter of economics for future plants. We know much more about costs now than we knew a year ago, as a consequence of relief in prices for different materials involved. Even at the present time, a closed-cycle plant of, say, 12,000 or 25,000 kw output with oil and gas firing does not cost much more than a modern steam plant, if one takes into account all the saving in space for the building, the foundations, the lack of feedwater preparation, etc.

In contradistinction to steam-turbine practice, the whole range of output from, say, 6000 to 50,000 kw can be covered by a few types of machines and apparatus only. Many elements, such as blades, tube bundles of heat-exchangers, headers, or air heaters, then auxiliaries as leakage compressors, regulating gear, etc., can more easily be simplified, classified, and normalized. This will help considerably in achieving low-cost fabrication methods, and therefore in lowering further the cost of a plant.

An oil-fired plant of about 17,000 hp net output is under construction now at the Escher Wyss Works in Zurich. It is hoped that, in the near future, much more can be said about prices, operating experience, and maintenance, based on facts. The overall efficiency, including all auxiliaries, of about 37 per cent at full load must also be considered when comparing prices and economy with other systems.

The author would like to point out that the construction of the test plant of 2000 kw at our works, which is dealt with extensively in the paper was started more than 10 years ago. This was really meant for development and research work and not as representative of later plants. All direct comparisons with other caloric plants concerning results achieved are therefore misleading. It may be of interest that in the same space which we used for the test plant, at the present state of technical development, we can install a 15,000 to 20,0000-hp plant. This shows that many simplifications and savings in construction based on the long experience and studies have been achieved.

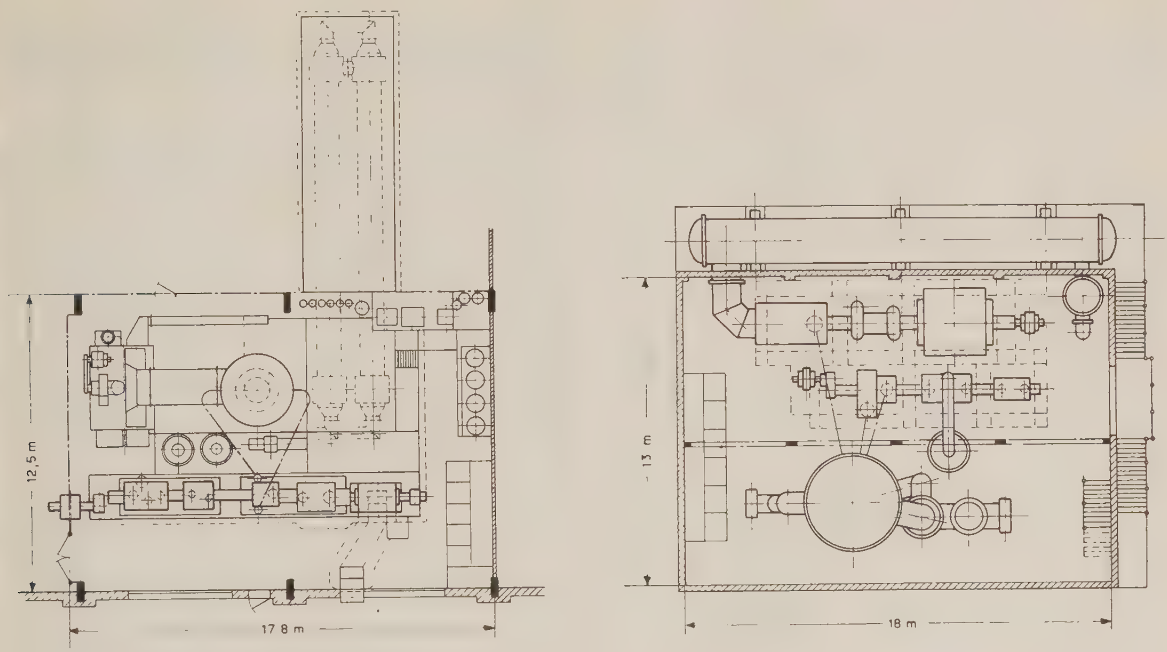

FIG. 36 SHOWS AREA COVERED BY TEST PLANT (LEFT) AND THE FIRST INDUSTRIAL APPLICATION (RIGHT) NOW UNDER WAY
(Test plant, 2000 kw; plant under construction, 12,500 kw.)

Fig. 36 of this closure shows the area covered by the test plant with which many American engineers are familiar, as well as the first industrial application now under way.[13] The saving is due mainly to the large output, and the choice of the higher pressure of 50 atm abs with a pressure ratio of about 10 between the low- and high-pressure sides because two-stage expansion is used.

All necessary details of this plant will be divulged to the engineering profession when it is finished. Therefore, the author begs the reader's indulgence for not going into detail here. However, it may be said that we are not only optimistic about the technical progress achieved but also concerning prices, as a result of the reasons given herein. In this respect, it must not be forgotten that the amount of special and expensive materials for the machines is small because the dimensions are small, as compared with other gas turbines.

On our last visit to America and England, it was gratifying to learn that the necessary tubes for the air heater can be produced on a normal commercial basis. They are similar to the type widely used for modern cracking plants under more severe conditions than occur in the AK heater. The total weight of heat-resisting material, compared with the total iron weight of the whole installation, amounts to only about 12 to 15 per cent. To give an idea; the cost of the air-heater tubes for a big plant amounts only to about 7 per cent of the total cost. These figures show that the supercharged cycle, because of its smaller machine dimensions and heat-exchanger surfaces, offers advantages generally not anticipated.

Reply to J. H. Anderson. The principal points concerning matters of cost have been dealt with in the foregoing general statement. In comparison with corresponding steam plants the better possibilities to normalize elements and to build standard plants will play a big role in the future and affect costs. As the air heater from the construction point of view is a simpler apparatus than a steam boiler, especially when supercharged-firing is used, there is no reason why this apparatus should cost more than a modern high-grade boiler.

The control and regulating system of the plant by varying the density only leads to valveless machines and apparatus in the cycle itself. The whole regulating block is a separate unit outside the cycle in cold atmosphere. This arrangement gives favorable operating conditions and much lower costs than the regulating device of modern high-pressure high-temperature steam turbines.

We have tried different shaft-seal constructions in the test plant. We now use the same liquid-seal method which has been widely used in our compressors for poisonous and dangerous chemical gases where shafts must be absolutely tight. As a consequence of the different alterations we have made in the test plant, the leakage loss has been relatively high. As the official report by Professor Quiby[14] on the trials shows, the compensation for air leakage amounts to about 30 kw at full load (2000 kw). The percentage loss in a new plant will be less.

The drop in efficiency of 3 per cent at a certain test compressor in a rather short time has been measured in different instances. It is believed that such losses depend naturally upon the individual form of guide vanes and runner blades. The difference in these friction losses in different airfoil-bladed machines may be due to the different Reynolds number involved. If a test is made with low velocities, the effect is naturally not so obvious as at high Reynolds number, where, as modern research on friction on plain and curved surfaces shows, the relative roughness plays a much bigger role.

Reply to F. T. Hague. It was very encouraging for us to have the opinion of an engineer who has dealt with many pioneer

[13] A brief description of the entire project is given under the title, "New Developments in Gas Turbine and Boiler-Plant Practice," *Power*, vol. 90, Aug., 1946; details of the Escher Wyss installations are given on pp. 88–90, of this article.

[14] Reprinted in *Oil Engine*, November, 1945.

efforts in the fields of steam and gas turbines and who has successfully carried new ideas on to realization. We can assure him that what seems to have occurred to him, when first studying our project, occurred to us also in the course of its development. Many details appeared to be formidable at first glance but our team of engineers has always found technical solutions. Construction of all elements has been based upon intensive research work in our laboratories. As we cannot accomplish all our objectives at the same time, we first have concentrated on the construction of oil- and gas-fired installations. For this purpose, we have arrived at simple arrangements for the air heater. The coal-fired air heater will be the next step to be achieved, based on actual studies and experiments. We are quite optimistic about this field too.

The principal problem in the case of pulverized-coal firing is the necessity for keeping slag and ash particles as far as possible away from the innermost rows of tubes near the combustion center. This can be effected by blowing in the recirculated flue gases, which serve for reducing the temperature of the furnace, along the whole inner tube wall between the tubes themselves. Tests which have been carried out in this connection show that this is already possible even in the case of small circulating volumes.

Since circulating gases at approximately 1000 F are relatively cold, the blown-in current has the additional action of bringing about quick cooling and granulation of the slag particles.

As Mr. Hague looks favorably and optimistically to the possibilities of the closed cycle, the author also would like to tell him that we think this system may also have a future application in connection with atomic power.

Reference has already been made to the advantages of employing light gases as the working medium for special purposes. For instance, helium or helium mixtures permit of heat-transmission values being attained which are of the same magnitude as for water. Thus, for otherwise similar conditions, the heating surfaces in a helium heater could again be considerably reduced compared to the air heater.

In so far as conditions can be foreseen, the closed helium circuit would prove suitable in a thermal power plant employing atomic energy. Since helium is neutral to the material of the pile and in addition has advantageous characteristics from the viewpoint of nuclear-physics in that, like graphite, it acts as a neutron moderator, the pile could be brought in direct contact with the gas. The pile might, in fact, possibly be arranged in a pressure vessel. The helium issuing from the compressor of the circuit could be supplied to the pile under a high pressure of 700 to 1400 psi, for increasing the heat-transmission figures and reducing the dimensions, and would afterward flow in direct contact as a cooling medium around the uranium rods, as also through bores in the graphite itself. In this way the pile would give up the heat which it produces by direct convection to the helium. It may be foreseen that under the assumptions mentioned (increased pressures) the large quantities of heat produced in the pile will, on account of the good heat transmission, be capable of being usefully taken up. In this manner no tubes would be required at all for the helium heater. Calculations show that if we assume a pile temperature of 1500 F as admissible, only a very small surface is needed to heat up the circulating helium to 1300 F.

For 1000 kw net output of the plant, about 15 to 20 sq ft in direct contact with the gas are sufficient without a greater total pressure loss than 3 to 4 per cent in the "pile heater." Therefore, a 50,000-kw plant would need only a heat-transfer surface of less than 1000 sq ft. Whether such dimensions are possible from the physical aspects is not yet known.

The adoption of a uranium pile in place of a combustion chamber would in the case of an open-circuit turbine be detrimentally affected by the fact that only small heat-transmission values are attainable with air. Furthermore, the nitrogen would absorb neutrons and the open air current continually deliver radioactive products to the atmosphere. Moreover, there would be the danger of the graphite in the pile being burned by the oxygen in the air.

The use of steam as working medium likewise has the drawback of a considerable absorption of neutrons and of a small heat transmission in the superheated condition, also requiring complicated installations.

It is also possible to combine a normal closed cycle working with air or another suitable gas with a "pile"-heater through an intermediate circuit bringing the heat from the pile to a heat exchanger (air heater with tubes). This intermediate circuit is closed too and would contain compressed helium in order to get high heat-transfer coefficients both in the pile and in the air heater. By this arrangement the working circuit with the machines is entirely separated and safe from radioactive particles. Calculations show that the necessary surface for the helium air heater amount only to about 200 plus 300 sq ft per 1000 kw when working with pressures of 60 to 80 ata.

We appreciate that the problems of utilizing the plant in conjunction with atomic energy are as yet unsolved and cannot yet be foreseen fully, particularly with regard to the construction. Nevertheless, the foregoing remarks show that the closed circuit has by no means reached the end of its development. On the contrary, this new principle opens up wide possibilities for the improved conversion of heat into mechanical energy.

Reply to M. L. Ireland, Jun. The method of estimating costs of future plants and the author's point of view upon that subject are discussed in the introduction to this closure. In connection with point 4, it may be said that all our recent studies show that pressurizing is very suitable for marine applications with oil-firing and also for gas-fired heaters in stationary plants. The control of the small charging set is not complicated.

The author agrees fully with the discusser's advice that only in specific cases can the different systems be reasonably compared.

We have made some studies for ship propulsion in the meantime and especially on the regulating problems in connection with variable-pitch propellers. Some of these problems were discussed by Dr. Salzmann, Chief Engineer of Research Laboratories of Escher Wyss, at the Detroit meeting of the A.S.M.E. in June, 1946. As a result of these studies, it may be said that we do not foresee any particular difficulties in regulating and stabilizing in connection with turboelectric drive or in combination with variable-pitch propellers.

For ships the closed air cycle offers an important advantage over the steam plant because of its ability to fit into given spaces as no gravity is acting on the working medium. All apparatus and machines can be arranged in a great variety of ways. The fact that the amount of air to be taken in for the closed-cycle air heater is the same as that of a Diesel engine or steam plant gives the same dimensions of inlet openings and outlet openings for the combustion gases. The combustion gas turbine needs many times more surface for these purposes.

Reply to Joseph Kaye. Our projects show that a closed-cycle power plant needs less total room than a corresponding steam plant of the same output. To give an idea; a 25,000-kw stationary plant can be built in a space of 90 × 52 × 40 ft for oil or gasfiring. In this space all the auxiliaries, all the accumulators, and all the heaters and regenerators of the machines are situated. The weight will be about 35 to 40 lb per kw. Under favorable conditions, we can guarantee an efficiency at the coupling of the output turbine of 37 per cent at full load, 35 per cent at half load, and even 27 per cent at 20 per cent load, all auxiliaries included; this with a working temperature of about 1200 F. Higher temperatures of 1300 F or more, we believe, can be real-

ized in the very near future, after having gathered experience with the first plant. The figures cited can be attained by modern steam plants only for large outputs and with rather complicated installations.

Reply to Alf Lysholm. It is true that we use high-efficiency regenerators in the closed-cycle system and that we need, like all other turbine systems, highly efficient compressors and turbines. But just to realize high figures constantly the absolutely pure medium offers ideal possibilities in the supercharged cycle. The Reynolds numbers are very high and heat transfer is good. The simple open-cycle combustion gas turbine uses a great excess of air to cool down the combustion gases to allowable temperatures for the turbines. This cannot be changed. The closed cycle needs about the same amount of intake air as a Diesel or steam plant.

Concerning the reheat-intercool cycle mentioned for open gas turbines but with high pressures, the author is not familiar with the possibility of its practical realization. The high pressure ratio of about 40, involved in this system, will require other constructions in machines. It is also thought that the different reheat stages would lead to constructional complications. More reheat stages or intercooler stages could also be introduced in the closed cycle as already pointed out in the paper. Yet the theoretical gain of efficiency would, in our opinion, be balanced by additional mechanical pressure and temperature losses.

Reply to Frederick Nettel. We try to follow the theoretical double isothermal cycle as closely as possible because we think this is a suitable way for gas processes to approach the maximum possible efficiency between given temperature limits with simple technical means which, from the engineering, constructional, and metallurgical point of view, are available today and in the near future. Mr. Nettel mentions another approach to the best thermal output, but as he does not give further explanation, the author cannot very well discuss the possibilities. It may be that there are other schemes for gas turbines but we shall stick to the one idea explained. All the deductions concerning best pressure ratios, reheat factor in heat exchangers, etc., should be regarded only in connection with the double isothermal cycle. For other cycles there are naturally, as Mr. Nettel points out, other most favorable relations. We arrived at these relations, for example, of 3–4 for the best pressure ratio with direct expansion and 10–12 for double expansion, not only from the theoretical calculations of the cycle but also taking in account all the losses of pressure and temperature involved in practical plants. Also, the quantity of circulating gases for a given output plays an important role.

The supercharging of the air-heater furnace not only lowers the surface of its tubes but also the surface of the combustion-air preheaters, because a part of the heat of the combustion gases is used in the expansion turbine which drives the supercharging set.

Concerning the size of the heat exchanger, it may be said that 75 per cent effectiveness is too low to get high efficiencies with normal circuit temperatures of about 1200 to 1300 F; 90 per cent effectiveness is really the figure upon which our actual construction is based. This corresponds to temperature differences of 45 to 65 deg F in this apparatus between the two air streams. It is an important item of the closed cycle that this high effectiveness can be realized with comparatively small surface and in small space; the heat-transfer in coefficients is many times higher than in an open-cycle installation, and we have to deal with an absolutely pure medium, so that we can really choose small diameters of tubes without any danger of scaling. As Mr. Nettel does not explain how he will attain better efficiencies and with what means, the author can only explain here the reasons which led us to our design.

All indications of air preheat are given in Table 2 of Quiby's official report.[15] As Mr. Nettel explains, an error crept in the explanation of Fig. 9 of the paper. It should be read as explained in his discussion.

Our studies on supercharged air heaters led us to the opinion that the charging set is not so complicated an arrangement as perhaps it seems at first glance. All the regulating is established automatically to a great extent.

The figure of 1.5 sq ft of heat-transfer surface per kilowatt relates to the air-heater surface plus preheater for a supercharged heater set. The air-preheater surface is dictated by the point of view of economy and is different in ships from that in stationary plants.

The size of the accumulators depends upon the chosen pressure of the cold-storage air, and upon the "time-table" of the plant. The load variation, Mr. Nettel admits, is communically severe. To give an idea we would need about 100,000 cu ft.

Reply to J. K. Salisbury. We were much interested to have Mr. Salisbury's opinion of our work. As we know, he is in every respect an expert in the gas-turbine field. We would like to thank him again for the compliments he paid to the engineering work of our team. The author was gratified to have had several conversations with him on this subject, and the principal advantages of the plants he mentions in his discussion are just those which convinced us since the start of our work to stick to it through all de-development difficulties. In connection with Mr. Salisbury's survey of the economics of the situation, for the reasons explained in the introduction to this closure, the author can only agree with him that it is an open question. Naturally, the author is not too familiar with conditions in the United States. For the markets he knows better, he is not at all doubtful about cost development.

The air heater with pressure firing is much less expensive than Mr. Salisbury estimates. The costs for the whole plant are even less than 75 dollars per kw.

The comparison of a 2000-kw modern steam plant which can be buil according to Mr. Salisbury's proposition, with the first 2000-kw closed-cycle test plant is not too favorable for the closed-cycle system. As explained in the introduction, one must not forget that the test plant was meant for other purposes than for industrial service and that it is a 10-year-old design. An up-to-date new design would naturally give much better values. Apart from that, at the present time we do not intend to build plants with such small outputs, because in this respect the author agrees fully with Mr. Salisbury: We think it would not pay. The advantages of the closed cycle are much more pronounced for plant ratings, say, above 5000 kw.

The last detailed questions of Mr. Salisbury can be answered as follows:

Surfaces. Our actual design is based on a total surface, including heater, regenerator, combustion-air heater preheater of about 4 to 5 sq ft per kw. These figures may change in the future with other designs. About 20 per cent relates to the air heater, 20 per cent to the preheater and coolers, and 60 per cent to the regenerator. Naturally, this changes the whole aspect.

Pressure Drop. The absolute pressure drop depends upon the pressure level used; 10 to 15 psi result with a maximum pressure of ~800 psi in modern design. The smaller value belongs to the air heater of the test plant.

Circulating Fan. In operation of the test heater, we only needed a very small amount of recirculated gas to cool and protect the tubes, and the furnace, as a consequence of the layout. The secondary air of combustion, led in a suitable way along the lower end of the temperature. In new air heaters the construction will be somewhat different and a bigger fan will be used.

Mechanical Losses. As Prof. Quiby pointed out in his report,[15] these losses were higher than usual because of the special arrange-

ment of the machine group and bearings. At least 1.5 per cent total efficiency could have been won with other design. The oil seals play only a minor role in the mechanical losses. But one must not forget that the mechanical losses, as derived from the tests also include the internal leakage losses of working air. Therefore, an exact figure for the mechanical losses cannot be calculated. Naturally, the rather high value is not inherent in the closed-cycle system and can be kept normal in other plants.

The Influence of Viscosity on Centrifugal-Pump Performance

By ARTHUR T. IPPEN,[1] CAMBRIDGE, MASS.

The wide use of centrifugal pumps in the oil industry demands that their performance characteristics for oil be predicted with reasonable assurance. A systematic study of performance changes with increasing viscosities had not been undertaken so far under controlled laboratory conditions. For the purpose of undertaking such a study theoretically and experimentally, the Hydraulic Laboratory of Lehigh University and the Cameron Pump Division of the Ingersoll-Rand Company co-operated in following through a comprehensive program of research at Lehigh University during 1944 and 1945. Over 200 performance tests for viscosities up to 10,000 SSU were completed on four variants of centrifugal pumps, employing a special test stand designed and built under war conditions.

The influence of viscosity changes on the head, discharge, and input-power characteristics of the pumps is demonstrated graphically and systematically for a wide range of viscosities and speeds. The usefulness of a special Reynolds number for pumps is demonstrated and test results are correlated on that basis. General conclusions are possible as to the influence of various features of design, since several pumps of different specific speeds were tested. A theoretical analysis establishes definitely the variables to be considered and places the discussion for the problem on a sound scientific basis, especially with regard to disk and ring losses.

Introduction

The influence of viscosity on the performance of centrifugal pumps has not received, up to the present, the systematic attention of the pump engineers which this problem deserves in view of the ever-increasing use of the centrifugal pump for the transport of viscous liquids. A first pioneer effort, and so far the most extensive one, was made by Professor Daugherty (1)[2] about 20 years ago. Various papers (2, 3) have appeared since then, but they were never based on more than a relatively small number of field tests carried out by various investigators with greatly differing pumps. These tests were used to derive correction curves for efficiency by extensive extrapolation of the test information on hand and must clearly be recognized as a temporary means of considering the viscosity influence.

In recognition of the need of a systematic approach to this problem under controlled conditions, the Hydraulic Laboratory of Lehigh University and the Ingersoll-Rand Company of Phillipsburg, New Jersey, entered into a co-operative agreement to explore the behavior of centrifugal pumps when pumping oils. A special test stand was designed and constructed under wartime restrictions and over two hundred performance runs were made during 1944–1945 on four variants of centrifugal pumps with viscosities ranging up to 10,000 SSU. The influence of viscosity changes on the head, capacity, and input-power characteristics was systematically explored for each variant of pump and for several speeds. The usefulness of the Reynolds number for the presentation of these characteristics was demonstrated convincingly and all results are correlated on that basis. The data presented in the diagrams are those obtained in the present investigation only. No attempt was made to include information from other sources, since it was almost always found lacking in completeness. This situation is analogous to that existing for many years in the field of pipe friction, until finally the pertinent variables were definitely established and a final solution found.

It is to be hoped that the experiments reported here will bring forth many detailed experimental results in complete form from the files of various investigators, so that the conclusions drawn from this work may be either confirmed or be modified to fit into a more general picture. However, it is felt that the paper represents the first attempt to analyze the test results on the basis of all pertinent variables so that individual influences can be isolated more easily. Thus their share in the over-all effect of the viscosity on the performance of centrifugal pumps is to be recognized in relatively true proportions.

It may be mentioned that the paper had to be severely limited in presenting the information on hand. The descriptive part on the experiments had to be cut more than may be desirable from the viewpoint of clarity in order to include the fundamental results of greatest value to the engineer concerned with this problem. The same limitations had to be imposed on the theoretical part which covers only those aspects which are most useful and characteristic in explaining the results of the experiments as stated in graphical form.

EXPERIMENTAL PART

General Outline of Test Program

The stated purpose of the experimental study was to find, if possible, a specific relationship between viscosity on one hand and efficiency, head, capacity, and power input on the other hand. The results for four different variants of pumps were to be correlated. The choice of pumps was naturally restricted by the power supply of the hydraulic laboratory, by the range of the torsion-dynamometer, and by the available storage space for the oils used in the tests.

(a) *Specifications of Pumps Tested.* Under the circumstances the four pumps listed in Table 1 were chosen as most suitable and as of widest possible use in the pumping of highly viscous oils. All vital dimensions and normal-performance data for water are listed in Tables 1 and 2. Pump No. 1 was a single-stage, single-suction pump which could be fitted with hydraulically identical impellers of the closed and open type. This pump was relatively small and of low specific speed. The pump No. 2 was a single-stage double-suction pump which could be run with impellers of different diameter. Thus the specific speed was varied to a value of almost 3000 with a small impeller, however, not without some sacrifice in efficiency. The experiments covered, consequently, a range of specific speeds between 1000 and 3000 which is the range of most efficient operation for radial-flow pumps.

[1] Associate Professor of Hydraulics, Department of Civil and Sanitary Engineering, Massachusetts Institute of Technology.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., Nov. 26–29, 1945, of The American Society of Mechanical Engineers.

(*b*) *Properties of Liquids Employed.* The program of testing contemplated originally the use of water and of two oils as experimental liquids. For the latter the range of viscosities was defined by the range of temperatures seasonably obtainable, i.e., from almost normal outside temperature upward to a maximum produced by dissipating the power input. No special heating or cooling equipment was provided. The variation in the volume circulated was sufficient to obtain the desired range of temperatures and viscosities. Two black oils, giving a medium slope in the viscosity-temperature chart, were selected and are referred to subsequently as heavy oil (HO) and light oil (LO). A third oil was added, however, after the LO-runs were taken, since it appeared desirable to bridge the gap between the viscosity of water and that of the light oil.

This so-called thin oil (TO) was produced by mixing the light oil with a certain amount of fuel oil. The light oil proved to be extremely stable and its viscosity remained constant for almost half a year of intermittent testing. The thin oil and the heavy oil were affected somewhat by the pump testing and frequent analyses were made to adjust results for changes in viscosity whenever they exceeded the limits of permissible errors. Fig. 1 presents the essential data on viscosities and specific gravities for these oils.

(*c*) *Schedule of Testing Program.* The experiments started by determining the performance of each impeller of the two pumps for water. All impellers were tested as summarized in Table 2 for three different speeds with the exception of the largest impeller, which was run only for the two lower speeds, since the torque for the high speed exceeded the dynamometer capacity. The water tests checked very closely the performance of the pumps reported from the Ingersoll-Rand laboratory and served in addition the purpose of establishing working procedures and of finding out about the general performance of the test stand. After some minor alterations were completed, the four sets of runs

TABLE 1 PUMP DIMENSIONS

Pump No.	Imp. No.	File No.	Type	D_s	D_d	No. of Vanes	D_1	D_2	D_{sh}	B_2	D_R	B_R	$2 \cdot S_R$ Av.
1	1 Closed	IL 11	Single Suction	4"	2"	7	2-1/2	7-1/4	15/16	1/2	3	7/8	.014
1	2 Open	IL 12	"	4"	2"	7	2-1/2	7-1/4	15/16	1/2	3	7/8	.014
2	1	IL 21	Double Suction	8"	6"	7	5-3/16	8-5/8	2-1/2	1-15/16	7	1-7/16	.016
	1B	IL 21	Double Clearance	8"	6"	7	5-3/16	8-5/8	2-1/2	1-15/16	7	1-7/16	.032
	1C	IL 21	Half Width Rings	8"	6"	7	5-3/16	8-5/8	2-1/2	1-15/16	7	$\frac{23}{32}$	.014
2	2	IL 22	Double Suction	8"	6"	7	5-3/16	11-5/8	2-1/2	1-7/16	7	1-7/16	.018

A

TABLE 2 PUMP PERFORMANCE DATA

Pump No.	Specific Speed for Water	Maximum Efficiency for Water	Liquids Tested	Approximate Speeds in RPM
IL 11	1163	75.0	W LO TO HO	2330 2875 3460
IL 12	1163	74.0	W LO TO HO	2350 2875 3460
IL 21	2622	80.0	W LO TO HO	1230 1890 2320
IL 22	1991	85.5	W LO TO HO	1240 1880
Graphic Notations			● ● ○ ○	

were repeated for the light oil, covering usually the possible range of viscosities on the same day for any one speed. It was preferred to change the pumps and impellers rather than the liquids, thus avoiding contamination of one oil by the other, which would have become appreciable during the course of the testing program. The light oil was then cut by the addition of fuel oil to a lower viscosity and the four sets of runs were again repeated for this so-called thin oil. Finally the heavy oil was turned into the lines and the process was repeated a fourth time.

Thus two pumps with two impellers, as described in Table 1, were tested for four different liquids and for three different speeds. Six performance runs were taken for each oil in the average, so that the total number of tests exceeded 220.

Experimental Equipment

The experimental equipment was largely assembled under the restrictions of the wartime economy and therefore contains whatever could be made to serve on the test stand, from the material resources of the two co-operating agencies. The result of the planning on this basis, however, was adequate in every respect for the proper execution of the testing program.

(*a*) The general arrangement of the circulating system is shown in Fig. 2. Six 750-gallon tanks formed the basic storage units for two different oils. A triangular arrangement of three tanks each in two stories was decided to be the most compact and practical one with respect to piping and housing. When the pipes were filled with one oil, two additional tanks contained the same oil on two levels with the upper tank in the rear always empty and ready to be used as a volumetric-measuring tank. The upper tanks could be drained very fast by gravity into the lower tanks. The entire system of pipes and tanks may easily be analyzed from the isometric views in Fig. 2. The meter calibration circuit and the pump testing circuit are shown separately and are outlined by heavy lines. All tanks were open, however, all pipe ends were submerged considerably below the oil level, so that entrained air never posed any problem.

All the circulating was done by the test pumps driven by means of a torsion dynamometer, as illustrated in Figs. 3 and 4. The driving arrangement of the dynamometer was a system of pulleys and a 65-horsepower induction motor, which was powered from a special transformer set with 3-phase, 60-cycle, 220-volt alternating current. Automatic voltage regulation was provided in the 4400-volt main feeder line of the University, so that serious speed variations were never encountered during tests.

(*b*) *Measuring Equipment.* Pump discharges had to be metered over a wide range and three liquid meters shown in Fig. 5 were installed, therefore, in separate lines of 8 in., 6 in. and 4 in. diameter. Orifice meters were used for the 8 in. and 6 in. lines and a 4 × 2 in. Venturi happened to be available for the 4 in. line. The meter approaches were at least twenty-two diameters in length and the valves in the approach lines were only used in wide-open or completely closed position. The operation of the meters therefore was satisfactory at all times. All discharge regulation was done by means of valves at the end of the circuit, so that the entire system was normally under positive pressure. The orifice meter of 4.92 in. orifice diameter in the 8-in. line was used almost exclusively for the large pump and the 4 × 2-in. Venturi meter was adequate for the small pump, so that the 6 × 3.60-in. orifice was rarely used. The orifice plates were made of $^3/_8$-in. brass plates with a square edge of $^1/_{16}$ in. and a downstream bevel of 45 deg. The pressure connections were located one pipe diameter upstream and one-half pipe diameter downstream.

The temperatures of the liquids were checked at three points by means of standard thermowells and dial-type indicating thermometers. The latter were mercury-actuated and furnished

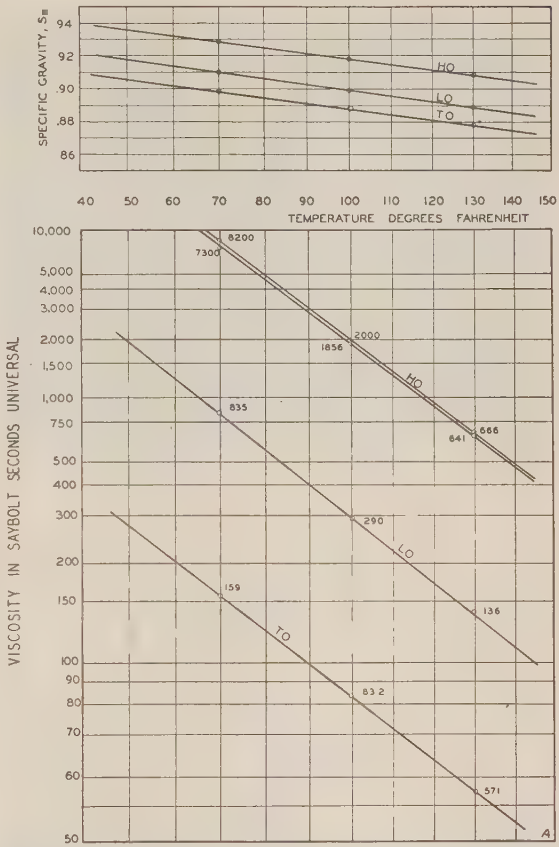

Fig. 1 Properties of Oils

with temperature-compensated capillary tubes. Readings to the nearest half degree Fahrenheit were deemed sufficient. The first well was located in the suction line of the pump about 4 ft upstream, the second was in the manifold immediately below the discharge line connection, and the third recorded the temperature of the liquid below the meters.

For visual checks, on the discharge and suction pressures, two Bourdon-type gages were included on the instrument board shown in Fig. 6; however, all discharge and suction heads used in the analysis were obtained from mercury-water manometers. The mercury manometer connected to the discharge side consisted of two differential gages connected in series, each of 6 ft length. A range of 50 psi could thus be covered without a balancing pressure. If the low pressure end of the manometer was placed under a 25 or 50-psi counterpressure, the range was extended to 100 psi. In the end the mercury gages arranged so as to require a minimum of corrections, saved considerable time since calibrations were unnecessary. The oil in the connecting lines was prevented from entering the water-mercury gages by means of large pots filled with oil in the upper half and water in the lower. The oil-water level in the pots was observed by means of glass gages and was checked and adjusted before every run. The same arrangement was used for the differential manometer connected to the liquid meters. Due to the relatively small differences in specific gravity of the oil and water, small changes of the oil-water level in the separating pots will not

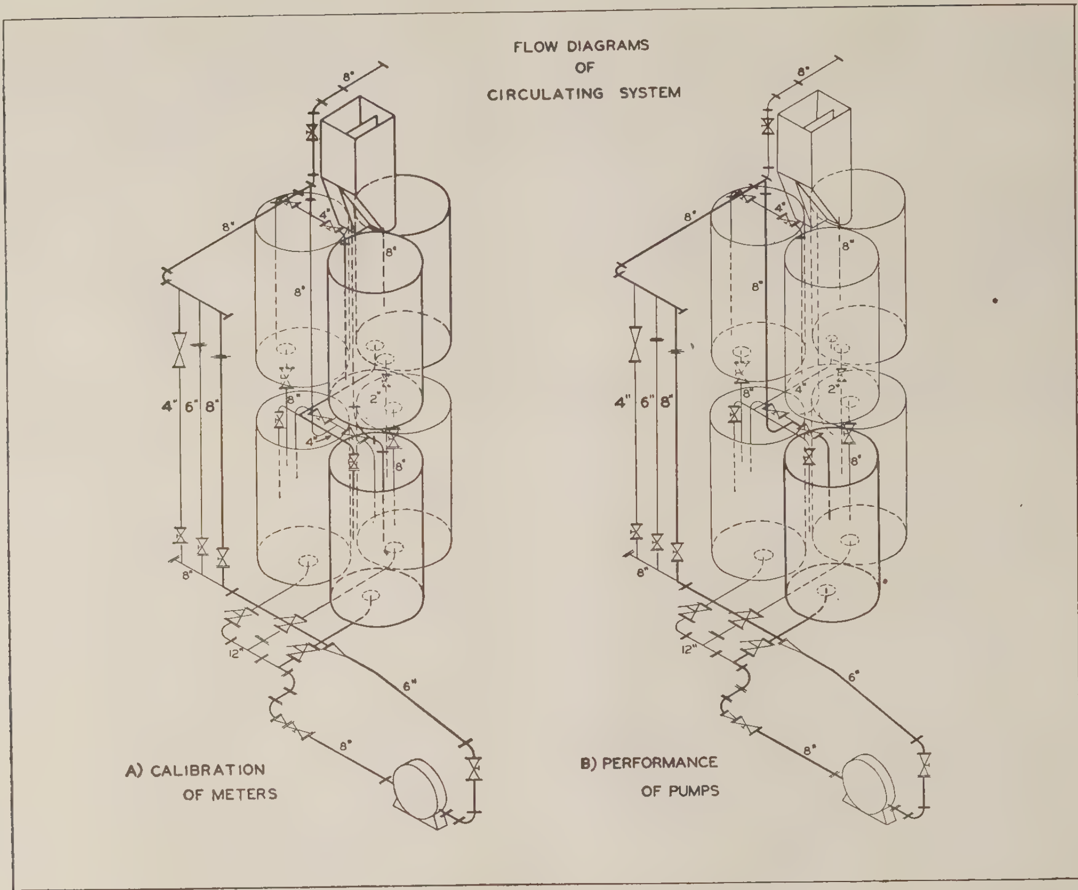

Fig. 2 Line Diagram of Circulating System

affect the readings. The speed was measured with a Chrono-Tachometer and with a revolution counter electrically operated for $^1/_{10}$-minute intervals. It was read for every test point and comparisons were made with a Hassler Tachometer with results well within the precision of either instrument. The tachometer and revolution counter are connected to a synchronous motor powered from a generator mounted on the pump shaft.

Experimental Procedures

(*a*) *Calibrations of Measuring Equipment.* All the instruments used were subjected to several calibrations during the experimental period. The mercury gages of course do not need special calibration, except that the Bourdon gage tester was checked against the mercury columns. All connecting lines were freed of air and particularly the discharge gage was checked before and after each run for air, by the use of a special water-air manometer, the latter indicating easily any errors of 0.01 ft of water column. Balance was always required within this limit and was always obtained.

The dynamometer bars were calibrated by careful static load tests and showed a constant torque per unit deflection over the entire range. Naturally the percentage of error of the reading for any run was dependent upon the torque range covered for any particular pump and speed.

The speed measurements were checked against simultaneous measurements taken with other tachometers and were found to be consistent within one fifth of one per cent, which was within the required limits.

The temperatures indicated by the dial thermometers were verified by the use of a mercury thermometer certified by the Bureau of Standards. The readings were also checked during runs by mercury thermometers in adjacent thermowells, which had been compared to the standard thermometers. Sticking of the dial hands was prevented by lightly tapping the glass cover.

The main problem in ascertaining the fundamental quantities for the performance of the pumps was encountered with the liquid meters. While the circulating system for calibrating purposes is indicated in Fig. 2, some of the phases of the calibrating runs merit mentioning. Noticeable temperature differences in the oil circulated had to be avoided. Therefore the oil was normally discharged from the line into the upper storage tank and from here by gravity back into the lower tank. The flow was then gradually switched over to the swing spout, thus taking the oil in the upper storage tank out of circulation. As soon as steady-flow conditions were established again through the meter, the flow was deflected into the measuring tank, while a constant level was maintained in the lower storage tank by admitting oil from the upper storage tank. This procedure naturally required considerable practice but worked out very satisfactorily. The results of the meter calibrations are given in Fig. 7 and are stated in a form which deviates somewhat from the orthodox way. The discharge coefficient C_D for the 4 × 2-in. Venturi meter is

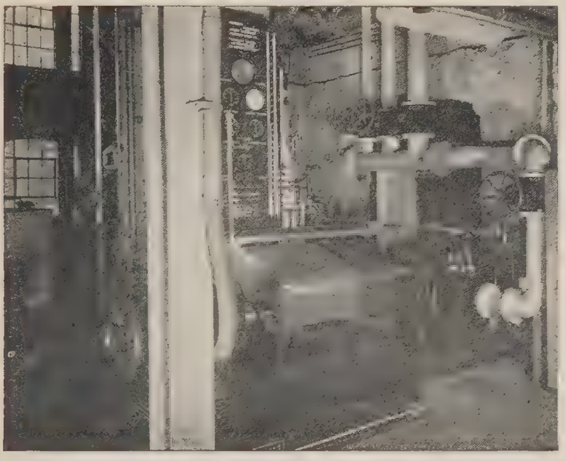

Fig. 3 General View of Test Setup

Fig. 4 Pulley System and Dynamometer

plotted against the Reynolds number for the 4-in-diameter pipe divided by the discharge coefficient C_D. Similarly the results for the 8 × 4.92-in. orifice meter are given in terms of a correction coefficient for viscosity K_m plotted against the pipe Reynolds number for the 8-in. pipe divided by the discharge coefficient. This latter quantity can be calculated directly from the differential-manometer readings and the correction coefficient K_m is thus obtained without trial and error. The discharge coefficient for water C_D, as introduced, is equal to 0.6185. The Venturi meter curve is compared to the curve given in the Fluid Meter Report of the A.S.M.E. and is seen to lie considerably below this curve. It is shown extending down to an experimental value of $C_D = 0.50$. The orifice curve is compared to Johanson's curve for a similar orifice in a 1-in-diameter glass tube and shows remarkable agreement. It may even be argued that the discrepancy is due to the slight difference in the orifice - pipe diameter ratio, which was 0.615 here, while Johanson's curve holds for 0.595.

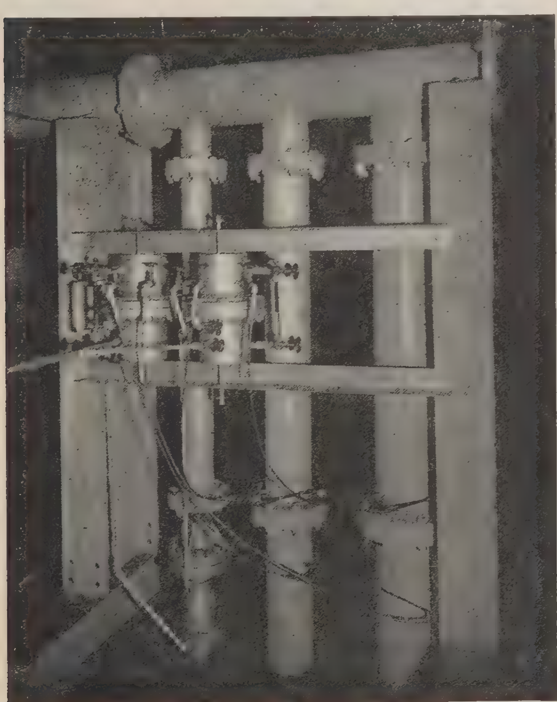

Fig. 5 Venturi and Orifice Meters

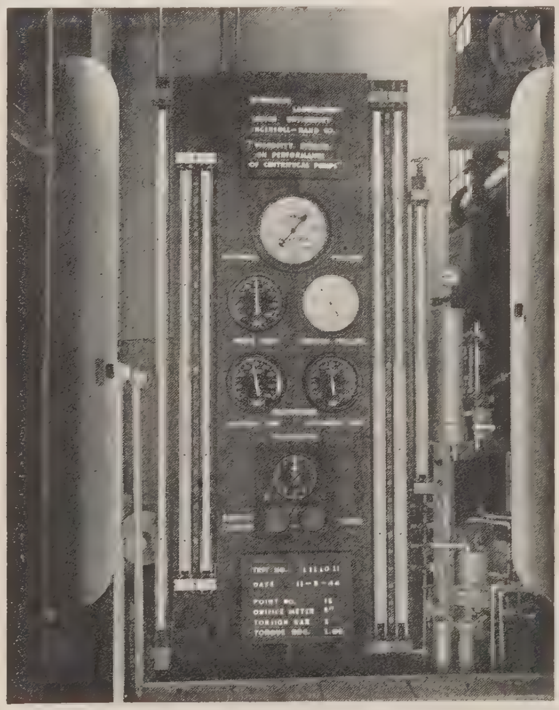

Fig. 6 Main Instrument Panel

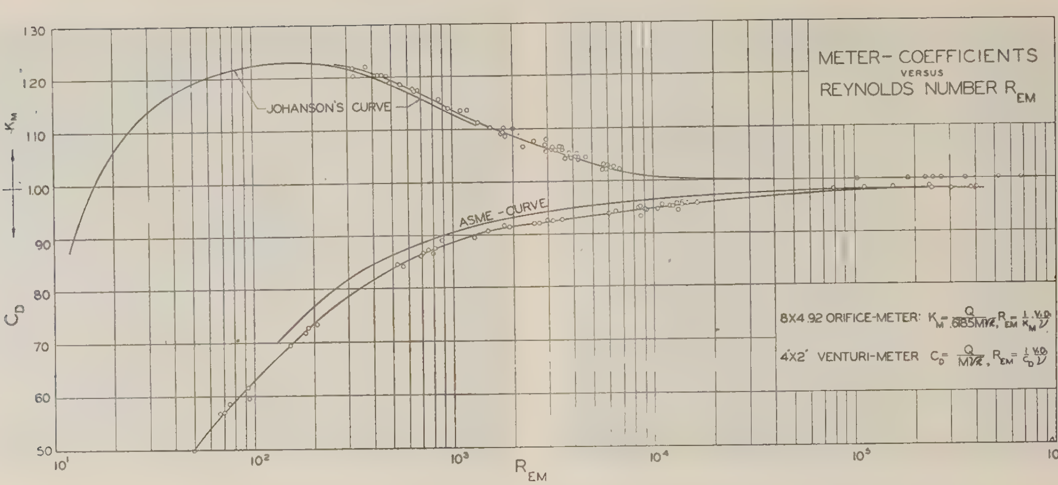

Fig. 7 Fluid-Meter Calibrations

(b) *Typical Performance Tests.* Most operational difficulties were discovered and ironed out by running first a full series of experiments with water. Only minor improvements, however, were necessary, and definite procedures for the oil runs were formulated. The dimensionless performance curves shown in Fig. 8 are those obtained for the medium-specific-speed impeller IL 22 and they are typical of the range covered by individual tests. The almost complete absence of viscous effects for the normal range of operation with water was indicated by all water runs, proving, if it be necessary indeed, that the Reynolds numbers normally encountered with water are in the so-called "rough-flow" zone.

All performance runs with oil were carried out by recording all pertinent instrument readings photographically by means of a 35-mm Kodak camera. Thus readings for individual points were taken speedily and simultaneously for permanent reference. Runs of ten to fifteen test points were normally completed within ten minutes, which fact considerably diminished the increase in temperature during any run. Upon termination of one run the pump was operated near normal discharge and was run continuously, until the dissipation of power had resulted in bringing the temperature to the desired higher value. The experimental procedure was then repeated. The temperature of the oil could be varied for any of the higher speeds from about 70 to 125 deg by continuous running within one day. Thus five to eight performance runs were obtained within a four- to eight-hour period. Shutoff conditions were usually tested by closing down fairly fast from a large discharge, thus retaining a fairly normal temperature of the oil churned in the pump. At best, however, the shutoff points are only approximate, especially for the higher viscosities.

The temperature changes in the oil could be controlled by adjusting the volume of oil in circulation. If the temperature rise was too fast the entire volume of oil could be utilized by circulating through upper and lower storage tanks. A differential temperature between upper and lower tank also permitted to arrest the rise in temperature during any run after some practice. Small increases in temperature were permitted, say 2 to 3 deg F, and their influence was determined by taking additional points immediately after each run over the same range. Corrections to a constant temperature and hence to a constant viscosity could be made therefore in the analysis. However, usually such corrections were too small to influence the maximum efficiency in location and magnitude.

Analysis of Test Results

(a) *Computation Procedures.* As pointed out before, all data necessary for the performance calculations are contained on a 35-mm film strip which could be projected, point by point, on a screen made of tracing cloth. The readings were taken down by an observer behind the screen. The gages showed throughout the experiments very steady readings; fluctuations were largely eliminated by the viscous action of the oil in the connecting lines and by symmetrical constrictions in the mercury-manometer blocks. Two exposures were taken for every point, in order to discover changes in readings and to insure that all the information could be read clearly. If poor test points were discovered later in plotting the results, they were invariably traced to faulty readings from the film and were easily corrected by checking. The value of preserving thus on film the original measurements cannot be overemphasized, especially when insufficiently trained personnel must be entrusted with a major portion of the computing work.

The entire calculation procedure involving a great many steps was carefully worked out, so that a standard routine could be followed throughout and the evaluation of data from the film to the final performance graph could largely be left to personnel without technical training. Viscosities and specific gravities, velocity-head corrections, and discharge coefficients were read from graphs plotted to simple linear scales.

Since the discharge manometer in many runs indicated very small differences due to the exclusive use of the 8 × 4.92-in. orifice meter, these readings were also read directly, which upon comparison with the readings obtained from the film showed very good agreement. It was found practical then to take down also the speed and the torque readings during the run so that some calculations could be started immediately after the runs were taken, before the films were developed.

Pressure readings were not corrected for pipe friction losses between pump flange and piezometer connections. This refinement would have introduced additional computing work to an extent unjustified by its merits, the pressure connections in each case being located only one pipe diameter from the flanges. It is natural also that the percentage error in the over-all efficiency becomes greater for higher viscosities, so that the foregoing correction stays usually within the permissible margin. It may be stated that the accuracy of the results is referred to the efficiency loss

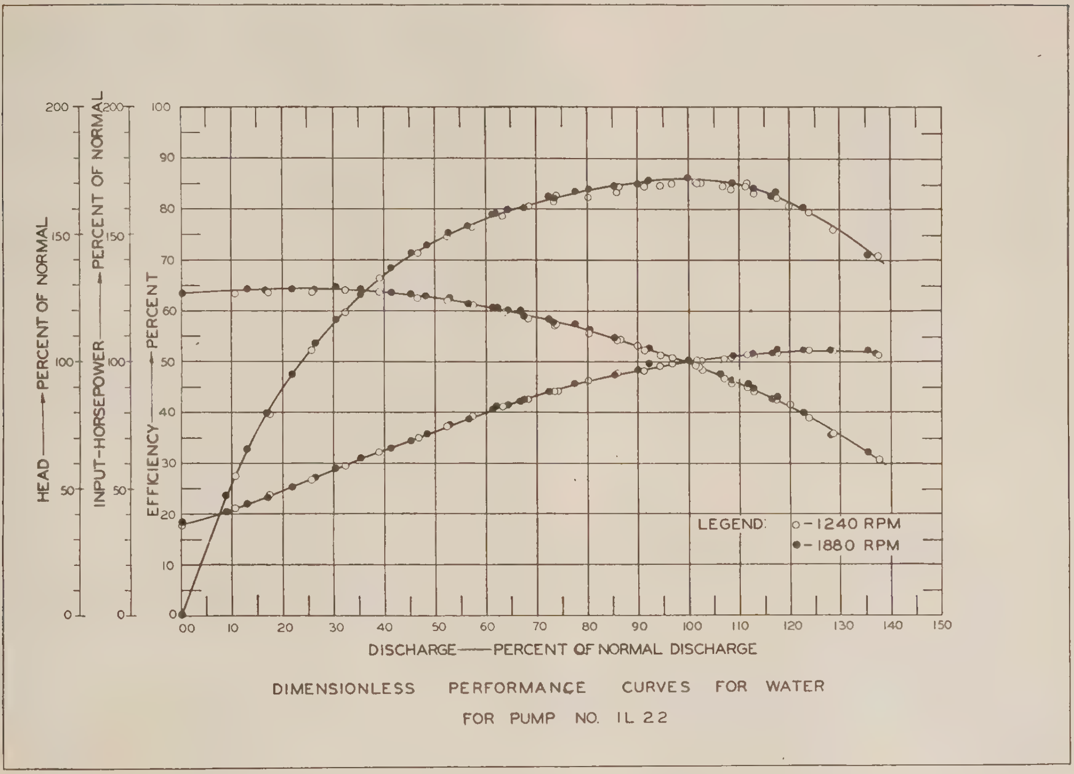

FIG. 8 EXAMPLE OF DIMENSIONLESS PERFORMANCE CHARACTERISTICS FOR WATER

$(100 - e)$ rather than to the efficiency (e) itself and that it is kept to ± 1 per cent in the average for $(100 - e)$.

(*b*) *Performance Characteristics.* Complete performance characteristics were plotted for all runs taken and discrepancies were therefore easily discovered. The influence of accidental errors is thereby minimized. Curves for successive runs were compared according to the order of their viscosities. These performance curves available for every run should yield upon further analysis additional experimental information concerning relative distribution of disk and ring losses, and on the operation near shutoff as well as at capacities larger and smaller than normal. The results presented in the following pages therefore touch only upon one phase of the problem under discussion, since maximum efficiency and normal capacities, head, and inputs are the only quantities considered at present in order to limit the scope of this paper.

ANALYTICAL PART

THEORY OF VISCOUS LOSSES IN CENTRIFUGAL PUMPS

(*a*) *Hydraulic Losses.* The so-called "hydraulic losses" in centrifugal pumps have been rather loosely defined in the past, since the thinking was predominantly influenced by the performance of pumps with thin fluids. With more viscous fluids the term "hydraulic losses" is here more definitely applied to the so-called "through-flow" losses, which are directly the result of skin friction and eddy systems along the primary path of the fluid passing through the pump. It is of course not possible to analyze these losses separately, since they depend on a considerable number of geometric variables, which may be briefly stated as follows:

1 Suction-pipe dimensions and shape of inlet which determine the state of flow at the entrance.

2 Design of pump inlet, eye diameter or its equivalent.

3 Shape, length, curvatures of impeller passages, contraction or expansion of cross sections.

4 Dimensions of volute, design of spiral, and diffuser.

It follows that in general the flow is nonuniform and that therefore pipe friction factors are not very useful here except in a very general way. In addition the curvatures encountered make any approach on the basis of boundary-layer theory impossible. A few general statements may be in order, however, in view of the conditions stated.

1 Every pump follows its own law of hydraulic resistance. Only the total losses can be determined experimentally, since they are interdependent.

2 Theories of individual losses will at best give only a very approximate quantitative explanation of the pump behavior. This may be extremely valuable, however, for the designer.

3 The relative weight of the losses will shift as a function of the specific speed and of Reynolds number. While skin-friction losses similar to pipe flow may predominate at low specific speeds, "body-resistance" losses will come to the fore with a change from radial to mixed and more or less axial flow for higher specific speed. In other words, since losses depend to a considerable extent on approach conditions, the character of the flow through the pump is more and more determined by the state of flow in the approach to the pump when higher specific speeds are reached. The relative length of the pump passages decreases with increasing specific speeds.

On the basis of these remarks, experimental results are indeed

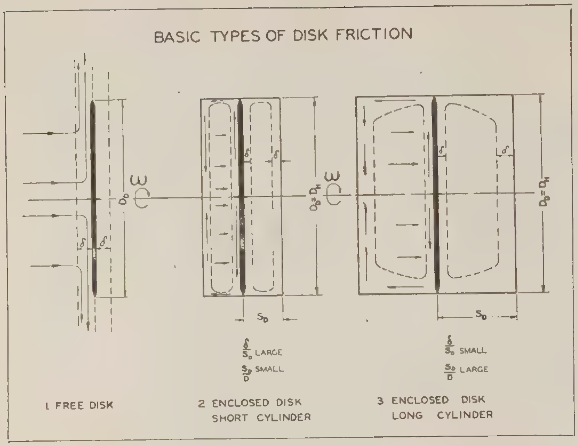

FIG. 9 BASIC TYPES OF DISK FRICTION

approached with hesitation. In retrospect it seems remarkable that the multitude of possible losses and geometric dissimilarities nevertheless produces statistical averages with a relatively narrow zone of deviations, as long as ring and disk friction losses remain comparable for the various pumps.

Energy losses for flows of a complex nature are customarily expressed in terms of velocity head of the mean flow. Choosing relative velocity at impeller exit as a suitable one, the difference between the head produced for water and the head for oil can be written as

$$H_l = (H_w - H_0) = C_R \cdot \frac{v_2^2}{2g} = C_R \cdot \frac{Q'^2_0}{a_2^2 \cdot 2g}$$

or dividing both sides by H_0

$$\frac{H_l}{H_0} = \left(\frac{H_w}{H_0} - 1\right) = C_R \cdot \left(\frac{Q'^2_0}{H_0}\right) \cdot \frac{1}{2g \cdot a_2^2} \quad \ldots\ldots\ldots [1]$$

wherein C_R is a function of Reynolds number, pump-design, and of roughness. Since $Q_0'^2/H_0$ was found practically constant for a wide range of Reynolds numbers, C_R can be obtained, if the water performance is known, from the plot of H_0/H_w against Reynolds number.

The question of a suitable Reynolds number was naturally the subject of considerable analytic experimentation. Eventually no particular advantage was found in any one as compared to another. However, the four fundamental forms stated below were calculated for all test points.

$$R_D = \frac{\omega \cdot r_2^2}{\nu} = \frac{u_2 \cdot d_2}{2 \cdot \nu} = \frac{\phi \sqrt{2gH_0} \cdot d_2}{2 \cdot \nu} = 2620 \frac{Nd_2^2}{\nu \cdot 10^5} \quad \ldots [2a]$$

$$R_H = \frac{\sqrt{H_0} \cdot d_2}{\nu} \quad \ldots\ldots\ldots [2b]$$

$$R_S = \frac{v_i \cdot d_{i_e}}{\nu} = \frac{283.7 \cdot Q_0}{d_{i_e} \cdot \nu \cdot 10^5} \quad \ldots\ldots\ldots [2c]$$

$$R_P = R_S \cdot \frac{d_{i_e}}{d_2} = \frac{283.7 \cdot Q_0}{d_2 \cdot \nu \cdot 10^5} = \frac{283.7 \cdot Q_0 \cdot b_2}{b_2 \cdot d_2 \cdot \nu \cdot 10^5} \quad \ldots\ldots\ldots [2d]$$

R_D and R_H differ essentially by the factor $\phi = u_2/\sqrt{2gH_0}$. Since ϕ very seldom differs greatly from unity, the reason for the small differences found in plotting the results against both is readily apparent. A small change in Reynolds number has little effect on the efficiency loss, while the differences due to pump design are more pronounced. The relatively best agreement between results for various pumps, especially for low Reynolds numbers, was obtained by using R_S, in which the equivalent or actual eye diameter of the impeller d_{i_e} was introduced as the characteristic length. For practical reasons R_D was chosen, since it is most readily calculated from known quantities. In addition disk and ring friction losses are most easily expressed as a function of R_D. It offers, therefore, the most systematic approach toward separation of the afore-mentioned losses from the so-called "through-flow" losses.

(*b*) *Disk Friction.* The problem of friction losses due to the rotation of a circular disk in a fluid has received the attention of investigators in theory and by experiments. A comprehensive view of this problem, however, has been lacking and thus the fact remained obscured that the theoretical and experimental findings of the various investigators show considerable agreement, if the various phases of the problem are systematically related. Fig. 9 illustrates the basis on which the subsequent treatment rests. Two papers (6, 7) attack the following problems.

1 The disk rotating in an infinite space, filled with fluid.

2 The disk rotating in a housing of approximately equal diameter and with small clearances between disk and housing.

Case 1. Von Kármán (6) dealt with problem 1 and established that the resistance mechanism depends entirely on the formation of a boundary layer adjacent to the surfaces of the disk, within which the tangential velocity of the fluid increases from zero to $\omega \cdot r$. The theoretical velocity distribution within this boundary-layer is given by Fig. 10 for the laminar case. This layer functions like the impeller of a pump. Since the pressure on the periphery of the disk is atmospheric, a radial flow q_D results into the surrounding fluid at rest with an angular momentum $u_2 \cdot r_2$ per unit of mass. On this basis von Kármán established the basic equations summarized below for laminar and turbulent flow. It is to be noted at this point that all kinetic energy imparted to the fluid by disk action is dissipated in the surrounding space. In general the torque to be applied is defined for one side of the disk by

$$T_D = C_D \cdot \gamma \cdot r^3 \cdot \frac{u_2^2}{2g} \quad \ldots\ldots\ldots [3]$$

(*a*) In the case of laminar flow, which is of particular interest here, the boundary-layer thickness is given by

$$\frac{\delta}{r} = \frac{2.58}{\sqrt{R_D}} \quad \ldots\ldots\ldots [4]$$

and the discharge of the disk by

$$q_D = 0.137 \cdot (2\pi \cdot r \cdot \delta) \cdot u$$

The so-called coefficient of friction becomes

$$C_D = \frac{1.84}{\sqrt{R_D}} \quad \ldots\ldots\ldots [5]$$

(*b*) For turbulent flow and a "smooth" disk, one for which all roughness irregularities remain submerged in the laminar sub-layer, the boundary-layer thickness is

$$\frac{\delta}{r} = \frac{0.462}{R_D^{1/5}} \quad \ldots\ldots\ldots [6]$$

on the basis of the seventh-root law for the velocity distribution.

The so-called friction coefficient becomes

$$C_D = \frac{0.0728}{R_D^{1/5}} \quad \ldots\ldots\ldots [7]$$

Equations [5] and [7] are plotted as lines A and B in Fig. 11. The application to "rough" disks has not been carried through, but it is obvious that the treatment is analogous to the friction problem for flat plates and will give an expression involving the ratio ϵ/δ, wherein ϵ stands for absolute roughness. The scattering of experimental results obtained by different experimenters for large Reynolds numbers R_D is probably due to variations in ϵ/δ. This case, however, is of little importance for the problem here.

Case 2. The second case defined before was treated experimentally and theoretically by Schultz-Grunow (7) and was essentially confirmed by the excellent experiments of Zumbusch as quoted in the same paper, the results of which are represented by line E in Fig. 11. The basic assumptions for this case should be clearly kept in mind for the later discussion of the differences in von Kármán's and Schultz-Grunow's results. The cylindrical wall surrounding the disk is assumed nearly equal in diameter to the disk and is short enough axially so that friction losses on the periphery may be disregarded. Thus the fluid is confined to the space between disk and housing and the angular momentum imparted by the disk is no longer lost to the surrounding fluid. Since the circular housing plates must exert a torque equal and opposite to the torque applied to the disk, it follows that the fluid confined to the space between them must revolve with one half of the angular velocity of the disk, so that equal frictional shears may be developed on disk and circular housing plate. This fact also is confirmed by experiments and is a common assumption.

Schultz-Grunow obtained for the one-sided disk for

(*a*) Laminar flow

$$\frac{\delta}{r} = 2.17 \frac{1}{\sqrt{R_D}} \qquad [8]$$

$$C_D = \frac{1.334}{\sqrt{R_D}} \qquad [9]$$

(*b*) Turbulent flow

$$C_D = \frac{0.0311}{R_D^{1/5}} \qquad [10]$$

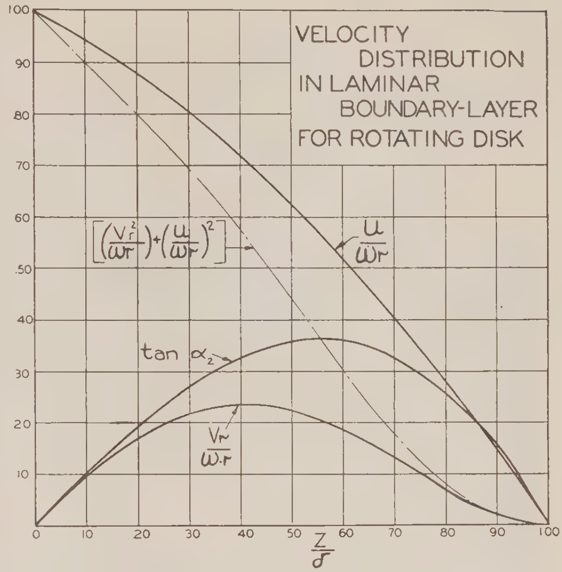

Fig. 10 Velocity Distribution in Laminar Boundary Layer Near Rotating Disk

These results are given as lines C and D_1 in Fig. 11.

Case 3. A more general approach to the disk-friction problem becomes now possible as a result of the work discussed as Cases 1 and 2. Using von Kármán's expressions, a general equation can be written for the frictional torque for a differential angular velocity ($\omega_D - \omega$). ω now denotes the angular velocity of the confined fluid, while ω_D represents the angular velocity of the disk. The results of this analysis may be reserved for a future paper, since space is lacking at this time. However, it is interesting to see the results of this analysis for the special case of $\omega/\omega_D = 1/2$. Introducing the latter ratio, the corresponding values of C_D are for

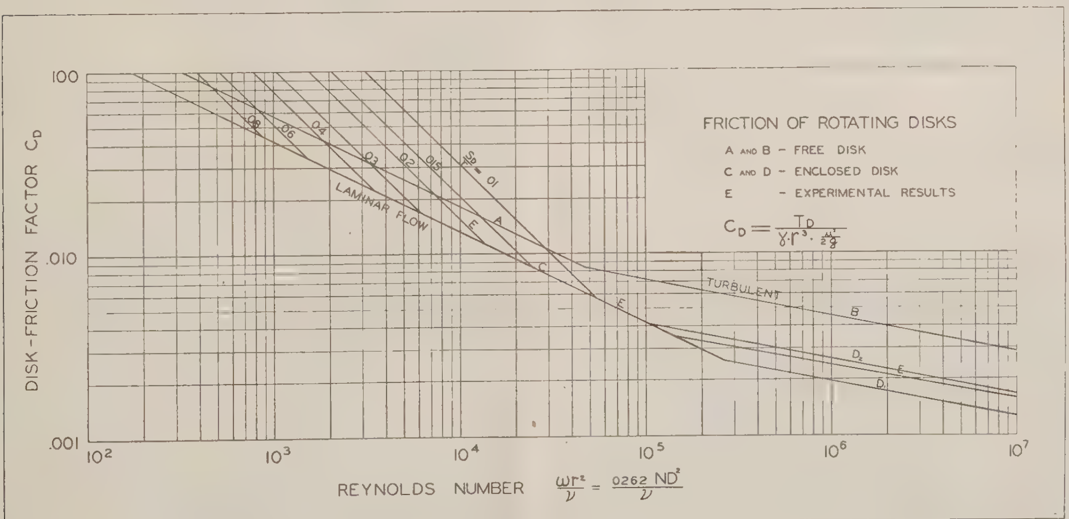

Fig. 11 General Reynolds Number Plot of Disk-Friction Coefficients

(*a*) Laminar flow

$$C_D = \frac{1.300}{\sqrt{R_D}} \quad\text{[11]}$$

(*b*) Turbulent flow

$$C_D = \frac{0.0418}{R_D^{1/6}} \quad\text{[12]}$$

with corresponding lines C and D_2 in Fig. 11. It is seen that von Kármán's equations applied properly check Schultz-Grunow's results extremely well for laminar flow. For turbulent flow the agreement is not as good, but Equation [12] checks the experimental evidence better than Equation [10], since values for the numerical constant obtained experimentally increase from 0.037 to 0.040 with higher Reynolds numbers. The experimental result permits another interesting conclusion, namely, the influence of friction on the cylindrical housing is increasing relative to the circular wall friction, since the boundary-layer thickness δ decreases with increasing Reynolds numbers, while the clearance between disk and housing, S_D, is constant. In other words, ω decreases in terms of ω_D.

The general conclusions for the influence of disk friction on pump performance especially for viscous fluids are therefore as follows. Theoretically the case discussed by von Kármán (see Case 1) represents the maximum disk friction which requires $\omega = 0$ or $\omega = \omega_D$. Neither condition exists normally in pumps. The conditions generally met in pumps are discussed as Case 3, with $\omega \gtrless {}^1/_2\,\omega_D$. The special case $\dfrac{\omega}{\omega_D} = {}^1/_2$ gives a theoretical minimum disk friction. A further more extensive discussion must be delayed at present.

Case 4. Special Case of Small Clearances S_D. A clearance is said to be small here, if radial velocities become zero and if the tangential-velocity gradient becomes constant across the axial clearance S_D. An elementary analysis gives

$$C_D = \frac{\pi}{R_D}\cdot\frac{r}{S_D} \quad\text{[13]}$$

Comparing this expression with Equations [11] and [4] as modified for the case of $\dfrac{\omega}{\omega_D} = {}^1/_2$ so that

$$\frac{\delta}{r} = \frac{3.65}{\sqrt{R_D}} \quad\text{[14]}$$

the following relation between S_D and δ is derived

$$\frac{\delta}{r} = 1.51\,\frac{S_D}{2r} \quad\text{[15]}$$

This equation represents the intersections of the C_D vs. R_D lines in Fig. 11, where the laws of resistance change from Equation [11] to Equation [13]. The influence of the boundary layer disappears therefore as soon as the total boundary-layer thickness (2δ) is of the order of magnitude of the clearance S_D. The transition is naturally a gradual one. The factor 1.50 instead of unity is explained by the velocity distribution in the boundary layer and by the fact that the law for small clearances takes full effect only after the gradient of the velocity has become uniform throughout the width S_D. Necessary minimum clearances for pumps lifting viscous liquids can be calculated from [14] and [15] to avoid excessive disk friction. Of course the length δ need not enter at all, once its physical significance is recognized and one may write directly the expression for the minimum clearance S_D in terms of a limiting Reynolds number R_{Dl}

$$R_{Dl} = 5.83\left(\frac{d_2}{S_D}\right)^2 \quad\text{[16]}$$

below which the disk friction would be unnecessarily large. While [15] and [16] have been obtained on the basis of $\omega/\omega_D = {}^1/_2$, they do not necessarily depend seriously on this exact value, since the total thickness of the boundary layers remains fairly constant for $\omega/\omega_D = {}^1/_2$. A decrease of δ near the rotating disk will be compensated by an increase of δ' near the circular housing plate and *vice versa*.

(*c*) *Ring Losses.* Considering losses due to the wearing rings it is clear that the discussion must be concerned with two different types of losses:

1 Leakage of fluid through the annular space due to the pressure difference between entrance and exit section.

2 Torque losses due to the tangential shear developed by the rotation of the pump ring within the stationary housing ring.

The attention of the pump designer has been focused mainly on the leakage losses, since thin liquids will readily pass in quantity even through small clearances and since the torque losses for such liquids remain small. It is clear that the general thinking must be revised as soon as the pumps are used to lift viscous fluids, which are 10 to 2000 times more viscous than water.

After a careful scrutiny of all information on hand (9, 10, 11, 12), considerable analytical work, which cannot be included here, established the following points:

(*a*) Leakage is not affected by rotation, as long as turbulent flow in the clearance space persists.

(*b*) Leakage will be affected by rotation whenever rotation changes the state of flow from laminar to turbulent in the ring-space.

(*c*) In pumping oils the leakage flow is usually laminar and is reduced very fast to an insignificant quantity. Temperature gradients along its path, however, prevent ready calculation of its magnitude. Volumetric efficiency is approaching unity.

(*d*) The torque losses due to the tangential shear developed are increasing considerably in magnitude and greatly affect the over-all efficiency.

(*e*) Reduction in leakage and increasing torque losses influence the running temperature of the pump and therefore the over-all efficiency.

Points (*c*) to (*e*) deserve further discussion. It is clear that reduction in leakage concentrates the dissipation of increasing torque losses within a smaller and smaller volume of liquid. Considerable heating must therefore be expected, which is fortunate indeed, since otherwise the viscous torque would very soon become excessive. As it is, the oil entering the ring space is absorbing the torque losses and is heated up with consequent reduction in viscosity, which in turn reduces the viscous shear and increases somewhat the leakage. A considerable temperature-gradient exists along the axial length of the ring. Naturally the problem is a rather difficult one, since the increase in temperature depends on the flow of heat from the ring surfaces, the exchange of heat from the rings to the oil leaking through, and on the temperature-viscosity function of the oil itself. However, some analytical solution is possible on the basis of adiabatic flow, which furnishes at least a rather useful qualitative explanation of the phenomena encountered in the analysis of the pump test results. The results of this solution show that differences in efficiency must be ob-

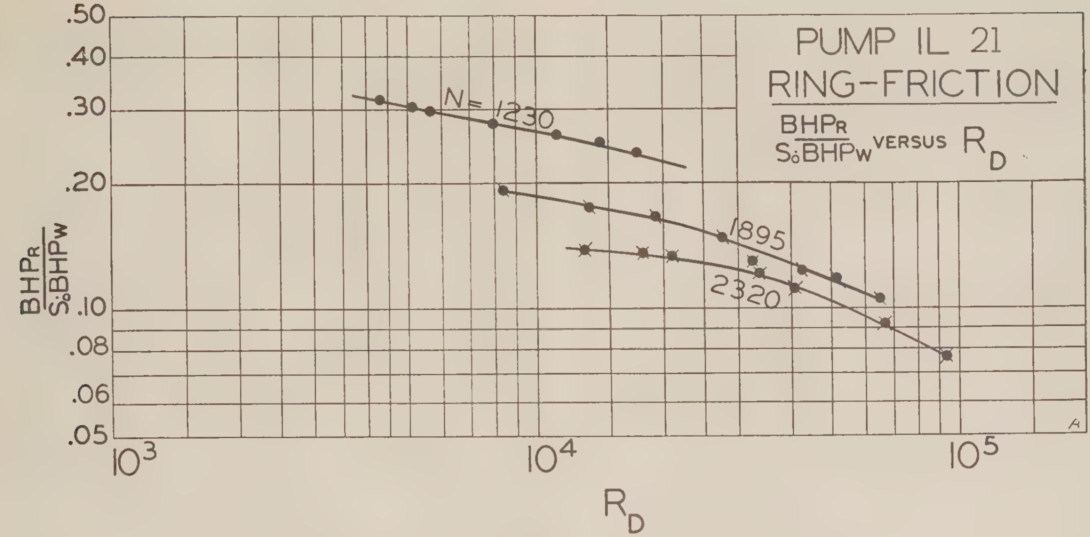

FIG. 12 EXAMPLE OF RING FRICTION AS A FUNCTION OF SPEED AND OF R_D

tained for the same Reynolds number R_D, if the speed of rotation is changed.

The horsepower loss BHP_R due to the tangential shear for one ring is given by the equation

$$BHP_R = \frac{3200 \cdot S_0}{\xi \cdot R_D} \cdot \frac{b_R}{S_R} \cdot \left(\frac{d_R}{d_2}\right)^3 \cdot \left(\frac{N}{1000}\right)^3 \cdot d_2^5 \ldots\ldots [17]$$

This equation is easily derived by assuming a constant viscosity except for the dimensionless factor ξ. The latter is a function of the temperature gradient along the ring length b_R, of the temperature-viscosity relation for the oil and of the ring dimensions. This factor ξ is therefore responsible for the breakdown of hydraulic similarity, in so far as the ring flow is concerned, since obviously a constant ξ would result in a continuous curve for BHP_R as a function of R_D. Using the analytical and numerical results obtained for ξ, the ratios $BHP_R/S_0 \cdot BHP_w$ have been plotted for three speeds against R_D for pump IL 21. The latter shows the effect of ring friction most clearly. It may be stated again that these analytical results should be judged as qualitative rather than quantitative, until more evidence becomes available. The plot, however, brings out clearly that the shift in power and efficiency-loss curves found in Figs. 14 to 16 for the various speeds is at least partly due to ring losses.

It can further be shown that increasing the ring clearance or shortening of the axial length of the rings does not have a corresponding effect on the power losses. This has also been proved by a few experiments, which, however, are not sufficiently complete to allow general conclusions at this time. The reasons for this behavior are probably that both measures produce increased leakage, thereby increased cooling and correspondingly higher tangential shear.

(*d*) *Miscellaneous Losses.* The only losses not discussed so far are stuffing-box losses and bearing losses. The latter are not affected by the type of liquid pumped and little need be said about them here. The stuffing-box losses are of the same type as the ring losses; their variation with the Reynolds number for any given pump will accentuate the tendency toward dissimilarity of hydraulic behavior, produced by the influence of the ring losses.

SUMMARY OF EXPERIMENTAL RESULTS

(*a*) *General Discussion and Methods of Presentation.* All experimental data were plotted in the customary way, i.e., performance curves showing input horsepower, head, and efficiency were plotted against discharge from complete shutoff to the maximum flow, which was always larger than the normal flow. All analysis work started from these basic curves, of which there are more than 220 covering a range of viscosities from that for water to 10,000 SSU. The analysis as carried out in the following is only concerned with the effect of viscosity on the so-called "normal performance points" of the pump, i.e., on the maximum efficiency. The influence of the viscosity on the shape of the performance curves may give considerable insight into the relative distribution of the various losses. Time at present did not permit, however, any further exploitation of these results, which must be reserved for a future date. In the following are given the methods which were followed to obtain systematically the variation of maximum efficiency, head, and input-horsepower with increasing viscosities:

1 The first approach considered the normal discharge for water as a constant also for the oil runs. Corresponding efficiencies and heads were obtained from the oil-performance curves and dimensionless correction curves were plotted as a function of Reynolds number. This was feasible only for lower viscosities and seemed to provide a good basis for comparison with water runs since all average velocities remain the same for water and oil runs. Thus the influence of viscosity is shown immediately and clearly. Increasing viscosity causes changes in the velocity distributions everywhere in the pump, which tend to become more and more nonuniform. Thus the kinetic-energy content of the flow must increase at the expense of the potential energy and, in addition, friction losses also grow rapidly for lower Reynolds numbers. The conversion of the kinetic energy to potential energy is naturally accompanied by higher losses. The size of the volute may not favor the same efficiency of conversion as for water. Furthermore, the velocity-head corrections are calculated on the basis of average velocities rather than on the basis of true kinetic-energy contents. A large decrease in head is therefore noted together with the lowering of the efficiency. Since

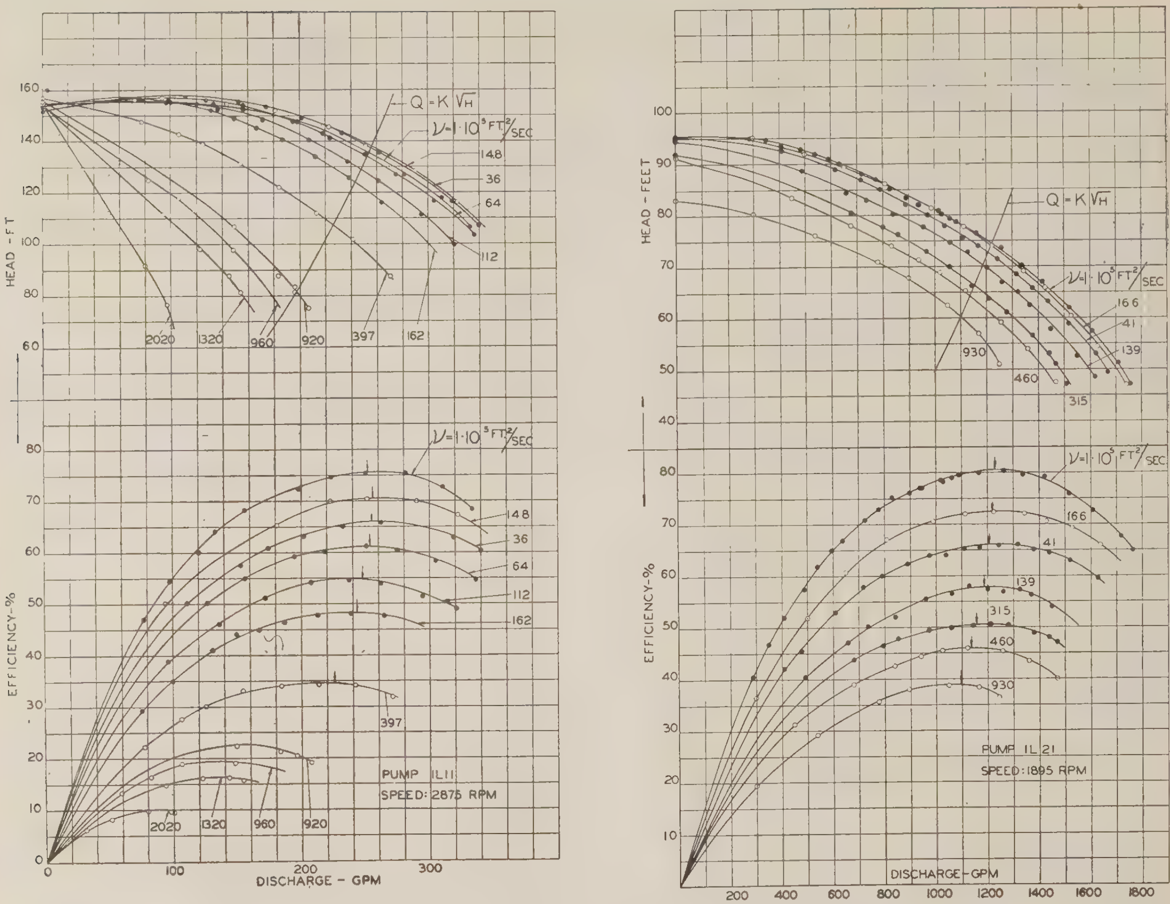

Fig. 13 Examples of Head-Capacity and Efficiency-Capacity Curves for Various Pumps and Viscosities

the efficiency on the basis of constant Q_0 departs markedly from the maximum or peak efficiency for higher viscosities, the method proved very soon impractical, as can be seen from Fig. 13, and was therefore given up.

2 At times it has been suggested to assume a constant head, regardless of viscosity. This gives normal capacities which are very soon much lower than the one corresponding to maximum efficiency as can easily be verified on Fig. 13. However, for viscosities up to 100 SSU either method may be used without introducing large errors.

3 It was found that if the ratio $(Q_0/\sqrt{H_0})$ was assumed to remain constant as calculated from the water performance, a parabola could be plotted, which would intersect the head-discharge curves at the point of maximum efficiency, as shown in Fig. 13. This proved to be especially helpful in view of the fact that for lower viscosities it is impossible to decide by inspection only on a point of maximum efficiency and corresponding normal discharge, head, and power. This method was used therefore in plotting the information presented in the following graphs, which represent the essential results of the study for easy reference. These graphs give the complete Reynolds number characteristics for the practical range of operation for all four pump variants.

(b) *Influence of Viscosity on Performance.*

1 Normal Head and Capacity. A few general remarks are in order to explain Figs. 14 and 15. All heads and horsepower inputs are given in terms of their corresponding water equivalents and are plotted against R_D. The efficiency loss $(100 - e)/100$ was plotted in preference to the efficiency, since logarithmic scales are used, so that accidental errors are shown in correct proportion. All curves show the water performance points on the right-hand side in black with special marks to identify the different speeds. The points for thin oil are white, followed by light oil points in black, and finally the points for heavy oil are white again. This identification by liquids proved desirable, since differences in performance were discovered as the liquid was changed, even though the Reynolds numbers remained of the same order of magnitude. The explanation for these differences is given later with the reasoning relative to the shift of curves with speed. The general trend of the curves is self-explanatory with decreasing heads and with increasing power input and efficiency losses as the Reynolds number decreases. The striking influence of the disk-friction losses becomes apparent by comparing Fig. 11 with Figs. 14 and 15. The steep rise in efficiency losses is directly traceable to the change from turbulent to laminar disk-friction losses. A special curve for reduction in capacity was found unnecessary

since the discharge Q_0 can be obtained from H_0 on the basis of $Q_0 = K\sqrt{H_0}$ wherein $K = Q_w/\sqrt{H_w}$. The latter relationship was found to hold true for Reynolds numbers R_D as low as 3000 to 4000. Below these values of R_D the capacity decreased more rapidly, the head-capacity curves became very steep and the points of maximum efficiency had to be picked by inspection. This, of course, results in greater scattering of the points, since at the same time the general accuracy becomes lower. A slight shift in the capacity may produce an extremely large change in the head. However, it is felt that the lower limit of $R_D = 4000$ will terminate almost any range that might come up in actual practice.

2 Input Power. The input power curves represent a correction factor by which the water horsepower input (BHP_w), as corrected for specific gravity, is to be multiplied to get the horsepower input for oil (BHP_0). These curves indicate a rather large increase in the latter for Reynolds numbers, for which the head and capacity corrections are almost insignificant. This would be further proof of the contention that the decrease in efficiency and the increase in power input for Reynolds numbers near the 100,000 mark is mainly due to the disk and ring friction. It should also be noted that pumps IL 11 and IL 22 show the largest increase here, while IL 21 with a small impeller but relatively large rings and IL 12 with an open impeller show smaller values. The rise in input horsepower for lower values of R_D is more or less linear in a log-log plot, corresponding to the change with Reynolds number of all hydraulic losses including disk and ring friction. For all pumps, differences in power input are found for constant Reynolds numbers as a function of speed. This may be explained in part by the effect of ring and stuffing-box friction as outlined in a previous section. The running temperature for higher speeds increases, this in turn causes a reduction of the shearing stresses, and thereby a reduction also in the percentage of ring losses in terms of total power input. The larger the power output for a given pump, the smaller will be the ring losses in per cent of input power for the same Reynolds number.

It is obvious then that these differences should be most conspicuous for pump IL 21 where the output was very small and that the difference between low and medium speed for pump IL 22 is much less, since the impeller of the latter was 35 per cent larger in diameter for the same inlet and ring dimensions. It is clear, furthermore, that the results for pump IL 11 will show the influence under discussion even less, since the rings here are very small in proportion to the impeller and therefore have but little effect on the total losses. The stuffing-box losses will accentuate the phenomenon, since IL 21 and IL 22 are double-suction pumps, while IL 11 is of the single-suction type. The disk-friction losses in contrast to the ring-friction losses will in general be larger for low-specific-speed pumps. They will therefore affect the performance of IL 11 most of all and will have a minimum effect on the behavior of IL 21. This, of course, was one of the reasons for selecting these pumps. It should also be mentioned that the curves for power input correction and efficiency loss for the low speed are probably not of practical value, since such speeds are seldom employed. They were included in these tests mainly to bring out the effect of speed and in order to increase the range of Reynolds number for each oil.

3 Maximum Over-All Efficiency. The tendencies of the head and input-power correction curves are essentially reflected in the plot of the ratios $(100 - e)/100$ against Reynolds number. These curves will show all the discontinuities encountered with each of the other types to a somewhat larger scale. The question may be asked, why the efficiency loss was not given as a fraction of the efficiency loss for water. The argument against the latter method would be that the efficiency loss itself represents already, to a certain extent, a dimensionless factor of resistance for a given type of pump, similar to the pipe friction factor (f) for a given pipe with relative roughness (ϵ/d)[5]. Just as it would be unwise to divide all pipe-friction factors by the special values of (f) obtained for various values of ϵ/d, so nothing could be gained by dividing all efficiency losses by the water efficiency. Since all efficiency losses remain constant with respect to Reynolds number in the range of water or air performance, while Reynolds number determines the behavior for more viscous liquids, the analogy to pipe-friction phenomena is close. The relative roughness of the pipe would find its counterpart in a specific speed modified to include a relative-roughness parameter. However, such an undertaking must be postponed until such time when more experimental material, systematically sifted and co-ordinated, is available. The efficiency-loss curves for low Reynolds numbers show a reversal of the trend at which the losses increase. It was pointed out before that below Reynolds numbers of $R_D = 3000$ to 4000 the capacity decreases at a much faster rate than above those values. A special capacity correction curve should have been added, which, however, was not believed necessary in view of its negligible practical significance and of the somewhat uncertain evidence which permits a wide interpretation, as can easily be seen in Fig. 13. If wanted, it can of course be calculated from the other curves.

There is, however, an interesting explanation for the lower rate of increase of the efficiency losses, which takes into account the heat exchange in the pumps. If no heat were developed by ring, stuffing-box, and disk friction, the efficiency would reach insignificant values very fast. As it is, the heat developed when heavy oils are pumped, will slow down this development considerably. As the capacity decreases the cooling of the pump body decreases, so that the efficiency-loss curve will approach a value of unity only for very small Reynolds numbers. The ultimate performance will be such that oil can be pumped only as it is warmed up by the dissipation of almost the entire power input into heat. The fact that this tendency became apparent in the curves also points to the limits of the practical use to which centrifugal pumps may be gainfully employed in pumping viscous liquids.

Because it was desired to show the application of a different Reynolds number, the results for pumps IL 11 and IL 22 have been plotted against $R_S = (283.7 \cdot Q_0/d_{ie} \cdot \nu \cdot 10^5)$. This, as is shown in Fig. 16, results in changing the order of magnitude of the Reynolds number by a factor of ten and since Q_0 will decrease somewhat with Reynolds number, the curves will be stretched over a wider range. But essentially nothing new is gained from this plot and the difficulty of having an unknown quantity Q_0 in the expression for R_S makes its usefulness quite doubtful.

GENERAL CONCLUSIONS

General Reynolds Number Characteristics

The results of the experimental and analytical work described so far show clearly that resistance of a centrifugal pump can be represented by distinct curves plotted against Reynolds number $R_D = 2620(Nd_2^2/\nu \cdot 10^5)$. Three curves are normally required to describe the behavior as a function of Reynolds number R_D:

1 Ratio of normal head for oil H_0 to the normal head for water: H_0/H_w

2 Ratio of normal power input for oil BHP_0 to the normal power input for water corrected for specific gravity: $BHP_0/(s_0 \cdot BHP_w)$.

3 The efficiency e or better the efficiency loss $(100 - e)$.

The capacity correction for the practical range of operation Q_0/Q_w is equal to the square root of the head correction: $\sqrt{H_0/H_w}$.

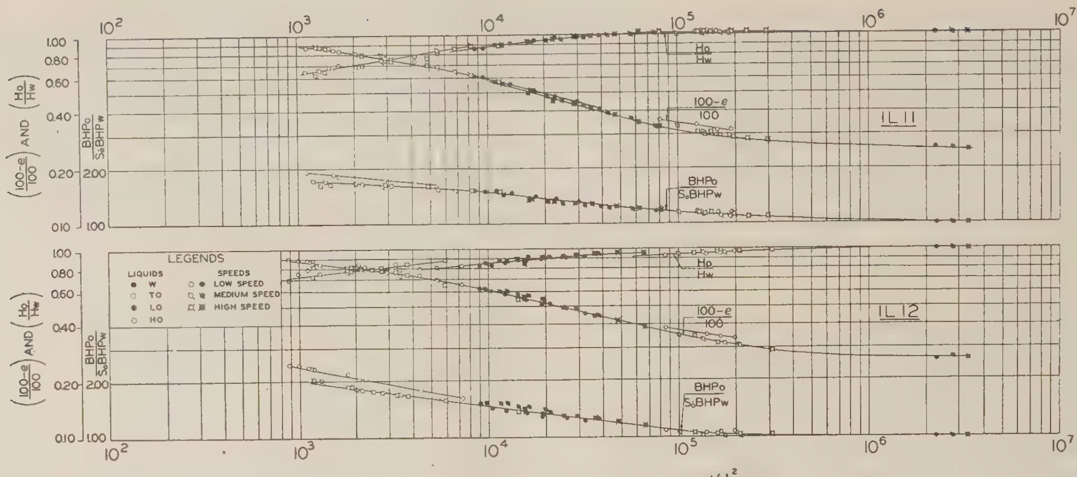

Fig. 14 Results Plotted Against R_D for Pump IL 1

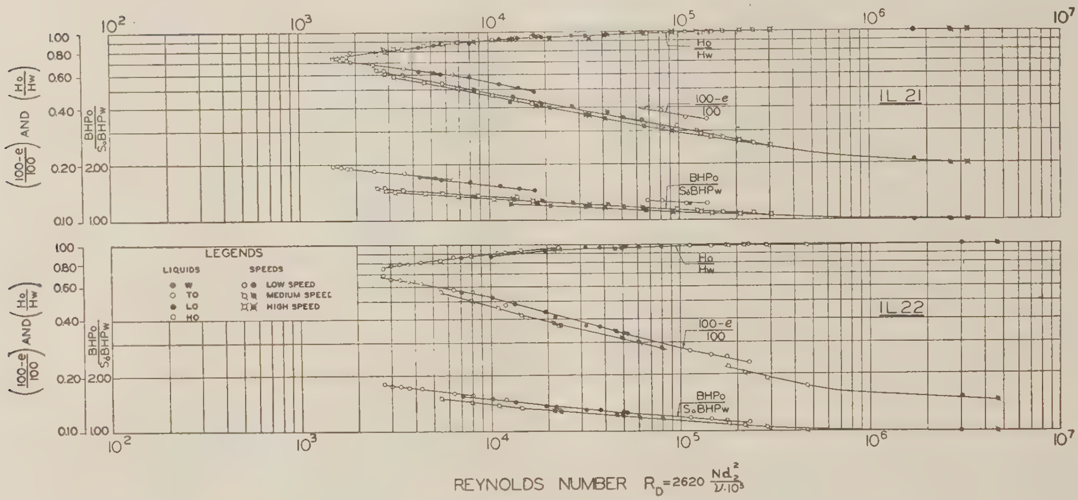

Fig. 15 Results Plotted Against R_D for Pump IL 2

In order to facilitate the conversion from the conventional SSU units for the viscosity to $\nu \cdot 10^5$ in square feet per second as used for the computing of Reynolds number, the equations of conversion may be stated here

$$\text{for SSU} > 100, \quad \nu \cdot 10^5 = \left(0.237\ \text{SSU} - \frac{140}{\text{SSU}}\right) \ldots\ldots [18]$$

$$\text{for SSU} < 100, \quad \nu \cdot 10^5 = \left(0.244\ \text{SSU} - \frac{210}{\text{SSU}}\right) \ldots\ldots [19]$$

In Fig. 17 the Reynolds number characteristics for the "normal" pumps IL 11 and IL 22 have been summarized for the purpose of practical application. It is proposed to use these curves for making general corrections for a range of specific speeds from 800 to 2200 approximately. Values for pumps of different water efficiency may be interpolated.

It is of importance that the experimental material basic to the eventual adoption of correction curves be greatly increased by further analysis and co-ordination of available information and possibly by additional testing. It is naturally desirable to translate eventually all the calculations necessary at present in using these curves into graphical and tabular form for easy application in practice. It was felt, however, that it would be valuable to have first the benefit of an extensive discussion by all those interested in this problem, before too much time is spent in adapting the present results to practical use.

Influence of Design Features on the Reynolds Number Characteristics

The general trend of the Reynolds number characteristics indicates three different zones within which the influence of the major losses is changing in weight.

1 For the range of water and air performance above $R_D \approx 10^6$ the efficiency losses are essentially due to the hydraulic

"through-flow" losses, followed in importance by disk friction and by leakage.

2 For Reynolds numbers $R_D \approx 10^6$ down to $R_D \approx 10^4$ the increase in power input is caused mainly by rapidly growing disk and ring friction losses, while through-flow losses increase at a comparatively low rate. The latter fact is indicated by the relatively small decrease in head and capacity, which proves that turbulent flow persists essentially throughout the pump. Leakage losses have assumed a negligible part.

3 For Reynolds numbers lower than $R_D \approx 10^4$ the through-flow losses increase more rapidly as indicated by a marked downward trend in head and capacity. Laminar-flow conditions are gradually established for the main flow. Disk and ring losses become less dominant and due to the large dissipation of power into heat on account of the latter the general rise in the efficiency losses is retarded.

Due to the complex nature of the flow through a pump it is naturally impossible to fix any Reynolds number R_D for the beginning of laminar-flow conditions for any one pump. It is even less possible to determine such a critical Reynolds number as a general criterion for all pumps. However, it is quite obvious that such a value of R_D would have little critical significance and is in no way analogous in this respect to the critical Reynolds number for pipe flow. The complexity of the losses and the non-uniform character of the flow preclude a sharp break in the line of resistance and point toward a very gradual and smooth transition between the two states of flow. This is indeed the conclusion to be drawn from the curves of Figs. 14–17.

A few additional conclusions may be drawn with respect to design characteristics: The head correction curves show that for low viscosities and low specific speed (pump IL 11) the head may increase at first above that produced for water. This is

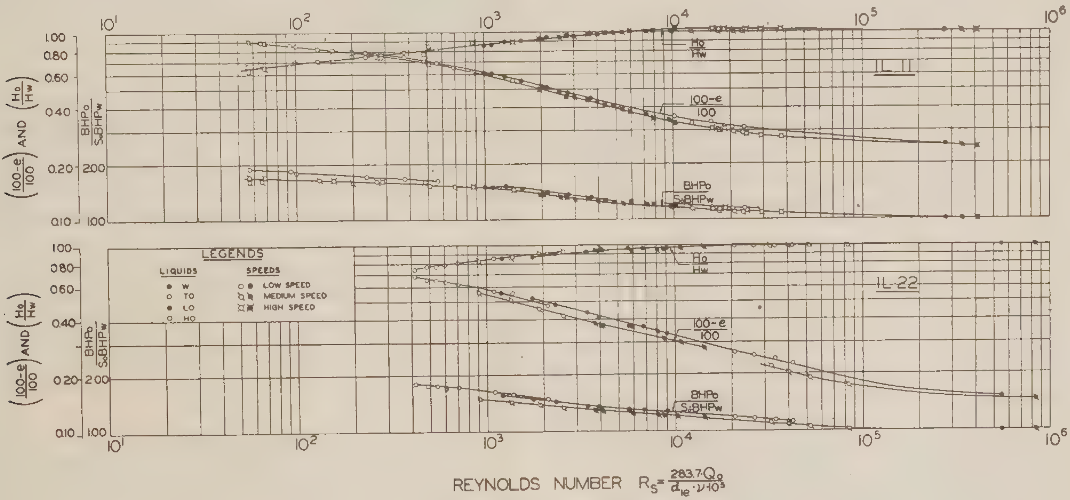

Fig. 16 Results Plotted Against R_S

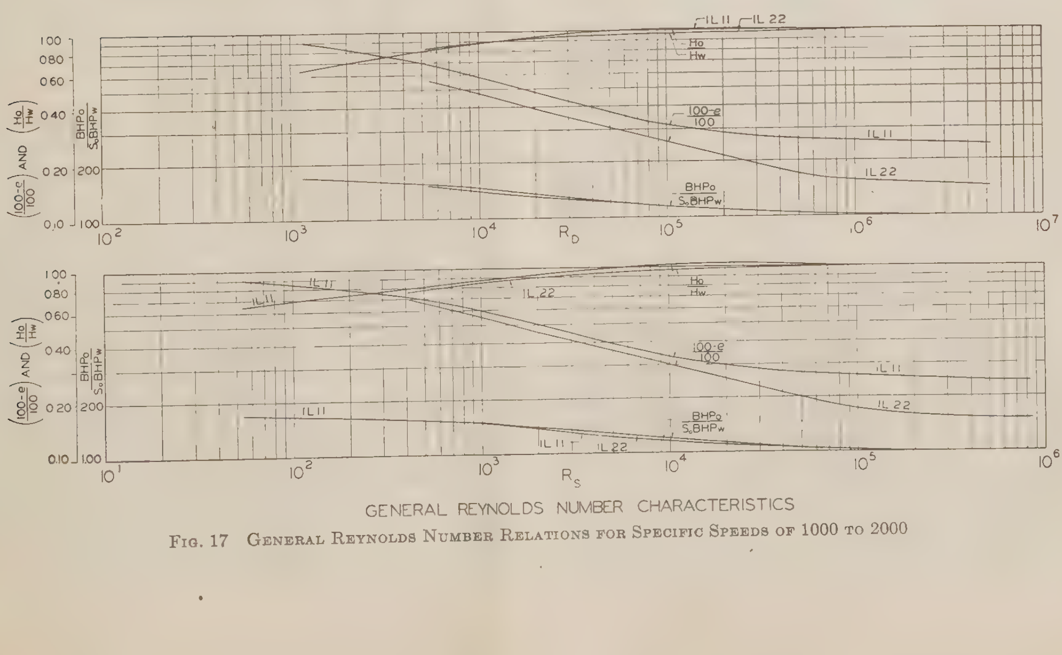

Fig. 17 General Reynolds Number Relations for Specific Speeds of 1000 to 2000

believed mainly due to the large effect of disk action for such pumps, since the pumping by disk action here becomes a considerable percentage of the total capacity. Further proof for this contention is supplied by the curves for the open impeller of the same pump, which show a lower head and lower power in the same range of Reynolds number.

If ring losses are to form a small part of the total losses, abnormally low speeds should be avoided. The useful output of a given pump is increased considerably for viscous conditions of flow by increasing the speed, since the influence of ring losses is diminished. This tendency is exemplified by the curves for the higher-specific-speed pump IL 21, which shows the greatest improvement with higher speeds, since ring and stuffing-box losses form a large percentage of the total losses for the lower speeds.

Individual losses, which are due to "hydraulic friction," to disk action, to ring and stuffing-box friction, are not easily separated at present but will vary apparently within reason in a proportional manner for pumps of conventional design. It may be assumed that a decrease in disk friction for a higher specific speed is compensated for in part by the relative increase in ring friction for this type of pump. However, there is still a net gain in efficiency when compared on the basis of constant Reynolds number and considering only the curves of Fig. 17. Compared on a percentage basis the efficiency losses differ less for low Reynolds numbers than for the water performance runs, since the curves converge rather rapidly for a certain range of R_D values.

A considerable increase in the running temperature for a given pump is caused by ring and stuffing-box friction. While the evidence on hand does not permit more than a qualitative estimate of the influence of the pump temperature on the performance, it may be stated that for a given Reynolds number and a given speed the efficiency is somewhat higher and the head produced is increased, if the running temperature is high. In the range of R_D, where Heavy Oil (HO) and Light Oil (LO) points overlap, the curves show a distinctly higher head for the former, since its temperature is high here while the Light Oil is cold. These remarks are made at this time with the intent of calling attention to the influence of the running temperature rather than with the idea of suggesting definite answers.

Attempts have been made during the evaluation of the data to analyze the individual losses separately and to deduct from the power input the influence of disk and ring friction. While expressions for disk friction give probably the order of magnitude of these losses correctly, the ring friction is not as easily calculated, since it is influenced to a decisive degree by large temperature changes in the ring space. For the time being, therefore, the attempt at isolation of the hydraulic losses from the input power was discontinued. But it is clear that eventually this phase of the problem will have to be solved, if further insight into the relative distribution of these losses is to be gained. This will be a very fruitful subject for further experimental studies.

Recommendations

On the basis of the information made available in this paper it is possible to propose some very definite future steps toward a final solution of the problem under discussion.

1 A considerable amount of detailed information, which up to date seemed irrelevant, should be disclosed by the profession as a result of the material published here. This material should be critically sifted and co-ordinated by a central agency on the basis of all the pertinent variables involved.

2 The information embodied in the performance curves and data, of which the results presented form an important but relatively small part, deserves further analysis. The influence of viscosity on part-load and overload operation should be cleared to a certain extent. Since disk and ring friction losses remain more or less fixed regardless of discharge, the detailed analysis of input and output curves should yield valuable information toward separation of these losses from the hydraulic losses.

3 The problem of ring friction should be studied further, analytically and experimentally. Applications toward improved stuffing-box design and use of mechanical seals may be included here.

4 Analytical studies should be undertaken to determine the influence of relative roughness and of specific speed as variables influencing the efficiency, so that eventually a set of universal performance curves may be plotted including these quantities in addition to Reynolds number.

5 Additional performance tests should be made with pumps of different design, especially with pumps of higher specific speeds. Such tests could be made on a much more economical basis now, since the pertinent variables and trends are established. Effects of changes in design should be systematically studied.

Acknowledgment

The study reported here represents an example of academic-industrial co-operation, which was sustained by the active interest of the administrative officers of Lehigh University and of the Ingersoll-Rand Company in this work. In this connection the author is especially indebted to Mr. W. M. Stanton of Ingersoll-Rand and to Prof. H. Sutherland of Lehigh University. Intimately connected with the technical phases of the work were Messrs. A. P. Brocklebank and H. Hornschuch, to whom the author wishes to express his particular appreciation. Mr. Ming Lung Pei was responsible for a great deal of analytical work. Thanks are due to a considerable number of other associates connected at various periods with this work and to the loyal co-operation of the staff members of the Hydraulic Laboratory of Lehigh University.

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Discussion

N. Tetlow.[3] The writer would like to congratulate the author on his paper, which provides an outstanding contribution to our knowledge of the viscous-flow characteristics of the centrifugal pump. He would like to endorse the author's statement that previous investigations have provided only temporary methods of approach; they have, however, served a very useful purpose indeed pending the accumulation of knowledge which permits a more scientific study.

Past contributions to engineering societies, and in published works, have always given rise to a good deal of discussion in which the theoretical principles associated with the Reynolds number of a pump have been ventilated at considerable length. All previous treatment of this subject, however, has been purely academic and has not been supported by practical evidence.

The researches now recorded by the author have done much to fill in this missing information and permit us to make a more accurate assessment of the influence of Reynolds number on performance. The writer would go so far as to say that this paper provides the first convincing evidence that he has seen in which the influence of running speed is demonstrated.

It is encouraging to learn from the paper that the author proposes, at a later date, to reproduce the results of his observations in a form which will lend itself more easily to the solution of practical problems. It is hoped, therefore, that some suggestions regarding the method of approach will prove helpful.

The method adopted by the author to correlate an almost bewildering series of observations is to plot, in the Reynolds curves, only the pump performances corresponding to maximum efficiency. To do this he has suggested that the locus of the maximum-efficiency point on the head-capacity curves, follows the law $Q/\sqrt{H}$ = constant.

It will be seen from the paper that this is a reasonable method of approach, and it has induced the writer to examine, from this viewpoint, the information in his own possession. This examination suggests that the path of the maximum-efficiency point on the head-capacity curves corresponds more nearly to constant specific speed than to constant $Q/\sqrt{H}$.

As an example, the curve in Fig. 18, taken from a paper[4] by the writer, and plotted on the basis of imperial gallons per minute, shows the characteristics of an 8-in. single-stage volute pump when handling oils of different viscosities. On these curves has been superimposed the constant-specific-speed line $\frac{G_1}{G_2} = \left(\frac{H_1}{H_2}\right)^{3/2}$ which, it will be noted, corresponds with a fair degree of accuracy to the points of maximum efficiency. A study of the examples in Fig. 13 of the author's paper, shows a similar trend. It is suggested, therefore, that the use of constant specific speed not only corresponds more nearly to observed results but, for reasons which are given later in this discussion, may also have other advantages when the available information is applied to practical problems.

The writer has also analyzed the results of a number of viscous-flow tests in a series of curves in which head and efficiency performances are plotted against N_d in the manner suggested in the paper, but now on the basis of constant specific speed. The available number of tests is limited, but they do suggest that a reasonable measure of agreement may become possible, as indeed it should, if pumps of similar specific speeds are compared. There is a fair amount of "scatter" but not more than one might expect, bearing in mind the large number of variables and the ever-present possibility of experimental errors. It is suggested, therefore, that a more complete investigation on this basis may prove illuminating.

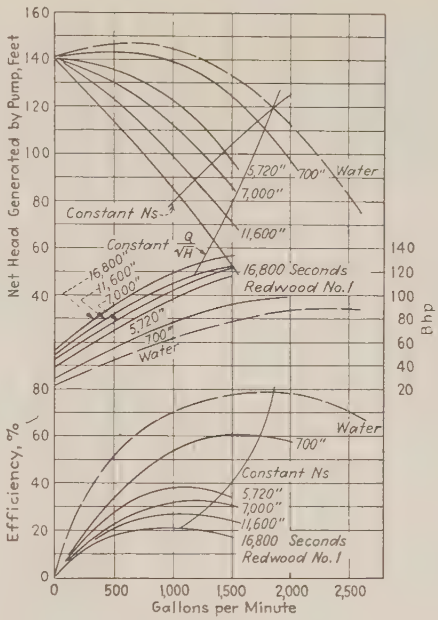

Fig. 18 Test Characteristics of 8-In Volute Pump Handling Viscous Oil Products

If the writer's preliminary observations on this subject are supported by further investigation, then it should be possible to prepare a complete family of Reynolds curves to cover all variations in speed, size of pump, viscosity, etc., each curve corresponding to a particular specific speed. In this event, the advantage of using the constant-specific-speed line instead of the constant $Q/\sqrt{H}$ line proposed by the author will become evident. Given any duty to be performed, once the running speed is determined, it will then be possible to calculate the specific speed, a figure which will establish immediately the appropriate correction curve to be used.

Such a series of Reynolds curves to cover the usual range of specific speeds would provide a concise means of applying the results of the author's tests to practical problems. By their use the size of pump most suitable for the duty required could be established, the problem being simplified by our knowledge that the specific speed will be constant whether handling a viscous fluid or water.

It would be possible also to superimpose on the Reynolds curves the maximum-efficiency duties, when handling water, for any range of pumps available. Not the least advantage of this method of approach is the fact that maximum possible efficiency when handling the viscous fluid could be assured automatically.

The writer would like to have studied this method of approach more carefully, but the combination of postal delays and closing date for communications does not permit this. It does appear possible that there may be a reason, based on fundamental principles, why the constant-specific-speed line should correspond so nearly to maximum-efficiency performance. He would like the author to approach the problem from this angle and would be grateful for any observations he is able to contribute.

The writer has only one criticism to make regarding this

[3] Consulting Mechanical and Electrical Engineer, Manchester, England.

[4] "A Survey of Modern Centrifugal Pump Practice for Oil Field and Oil Refinery Services," by N. Tetlow, Proceedings of The Institution of Mechanical Engineers, vol. 150, 1943, pp. 121–134.

admirable contribution to engineering science. The value of the paper would have been increased enormously if the author had found it possible to append a list of the symbols and the units of measurement used throughout the paper. When dealing with such measurements as viscosity and Reynolds number, it is so easy to introduce an element of confusion that it is hoped he will find it possible to include this additional information in his closure to the paper.

A. Hollander.[5] The author has made an important contribution to a subject which is badly neglected by the manufacturers of centrifugal pumps. It is regrettable that owing to the severe limitation of the length of the paper, some pertinent data were obviously omitted, thus preventing the examination of some aspects of the problem which are different from those of the author or held for future papers.

For example, the writer posed the question some years ago whether the affinity law broadened to include viscous liquids but narrowed down to head and capacity so as to exclude brake horsepower and efficiency and with these the external losses, is valid or not for the same Reynolds number. According to this law, taking a single impeller and having available the complete head-capacity (QH) curve for a given constant speed (N) and viscosity (ν) from shutoff to zero head, for another speed (KN), another complete head-capacity curve could be figured point by point for a viscosity ($K\nu$); the corresponding points of capacities would be KQ and of heads K^2H. Comparing the through-flow at any point of the impeller, the velocity diagrams should be similar for the two speeds of operation. The Reynolds numbers at different points of the impeller or case will vary for a single test point, but for the second test at a different speed and the same viscosity-speed ratio, they should vary the same way, because at any given point of the impeller or case the Reynolds numbers are identical for the two tests at corresponding capacities. For instance, if we have the test of pump IL11 at 2330 rpm and 10,000 SSU, we could figure the head-capacity curve for 1750 rpm, that is, $^3/_4$ speed for 7500 SSU, using as corresponding points $^3/_4$ capacities and $^9/_{16}$ heads. This is the only viscosity for which the foregoing test could be used at 1750 rpm. Going up in speed to 3500, the same test would give a performance for 15,000 SSU viscosity.

This prediction of the complete head-capacity characteristics from a test at a single speed for other speeds and viscosities, which have the same N/ν ratio (that is, the same Reynolds number) would be very desirable even if the best efficiency points were not at corresponding points due to a different change in the external losses. Unfortunately, the published data of the author (Figs. 14 to 17) show only one point of the performance curves at which $\frac{Q_0}{\sqrt{H_0}}$ is the same as the $\frac{Q_w}{\sqrt{H_w}}$ for the best efficiency point for water. Now the affinity law is valid for water (see Fig. 8, where the points coincide), so that

$$Q_{w1} = K_q N_1 \text{ and } \sqrt{H_{w1}} = K_h N_1$$

For any other speed

$$Q_{w2} = K_q N_2 \text{ and } \sqrt{H_{w2}} = K_h N_2$$

so that

$$\frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}} = \frac{K_q}{K_h} \text{ const}$$

means that corresponding points are plotted. What the author's modified affinity law tells us is, that for a given Reynolds number the H_0/H_w points should coincide. This is true for pump IL21, Fig. 15, showing a single line for this curve, and except for a short section for IL22, which shows about a 3 per cent deviation between $R_D = 10^4$ to 2.5×10^4.

The smaller pump IL11 shows a similar, and IL12 a somewhat greater deviation. Of course it would be instructive to see whether or not these deviations remain so small at other $\frac{Q}{\sqrt{H}}$ values for the whole performance curve. This would eliminate any doubt whether the deviations, amounting to something like ± 4 per cent, are due to experimental difficulties in a small region, which might arise if the flow at some point would be going through a Reynolds range between laminar and turbulent, where the repetition of a test point even in plain pipe flow is difficult. Also, it would be desirable to make tests with a much greater speed range, which would permit a similarly greater viscosity range for verification.

If, because of secondary effects, the affinity law were not fully valid, the second question that is posed is whether it would not be valid for geometrically similar pumps operating at speeds and viscosities giving the same Reynolds number against the same heads. In this case the speeds would be inversely proportional with the sizes and the viscosities proportional with the speeds. The velocities (peripheral, relative, and absolute) for corresponding points would be identical at the same heads, so that the corresponding flow rates would be proportional with the areas (square of the size ratio). Having made a set of tests at different speeds and viscosities with a single pump, we could predict the performance of a similar pump of size ratio (K_D) for speeds and viscosities ($1/K_D$) times the test figures.

A confirmation of these laws, which follow from a dimensional analysis, would leave as the only criterion for the prediction of the performance of similar pumps for viscous liquids the equation

$$\frac{K^2_D K_N}{K_\nu} = \text{const}$$

where K_D is the size ratio, K_N the rpm ratio, and K_ν the viscosity ratio.

It should be noted that, strictly speaking, the foregoing considerations are only valid if the leakage through the wearing rings is neglected; otherwise the zero point of the abscissa should be shifted by the amount of leakage, which differs for every head, even for a single Q-H curve at constant speed and viscosity. However, it seems that disregarding this leakage is permissible, because it is a relatively small percentage of the total flow (except near the shutoff point) and particularly for higher viscosities.

Lacking a reasonable theory, the reduction of Q and H at a constant speed and increasing viscosities can be determined only by experiments as the author conducted them. Even with a consistent design of a line, these should be repeated for different types, that is, pumps of different specific speeds over a wider range. Different hydraulic designs, such as single- and double-suction pumps, should come under different headings as primarily they should be compared between themselves, and only secondarily should the comparison be made between the two lines (a) for the same duty (Q, H, N) as the author did, and (b) for the same flow lines in the impeller, that is, taking one half of the capacity of the double-suction impeller, because it consists of two single-suction impellers pasted together with the central bridge omitted well below the outside diameter. For a given duty and particularly for low-specific-speed pumps, the single-suction pump should give better performance, and this even more pronouncedly for small sizes (lowest Reynolds numbers).

Going to the next step, that is, predicting at least one important point of the performance at different viscosities, the writer

[5] Associate Professor of Mechanical Engineering, California Institute of Technology, Pasadena, Calif. Mem. A.S.M.E.

would suggest for this, instead of the point of best efficiency, the point of maximum water or oil horsepower. This is a reasonably definite point, undisturbed by external friction losses, such as disk or ring friction, but dependent substantially upon the flow through the impeller and case only.

This approach would be in line with the author's laudible start of trying to separate the external losses, which could therefore be deducted from the brake horsepower. This figure, compared with the water or oil horsepower, would show the effect of viscosity on the flow or on the hydraulic efficiency of the pump.

The hesitation of the writer to use water or oil horsepower is due to the lack of an acceptable term, which should rightly be "output horsepower." After all, the pump is lifting in unit time a certain weight of liquid to a given height, and the product of these, times a constant, while regularly called "water horsepower," because water is the pumped liquid, is really output horsepower for any liquid.

It seems that for the two complete tests shown, the points of maximum output horsepower are on a line of

$$\frac{Q}{\sqrt[3.3]{H}} = \text{const}$$

Other tests indicate a different exponent of H, as one would expect from greatly varying designs.

These points of maximum output horsepower are for water not far from the best efficiency point, and if they separate further at greater viscosities it is a sign of a different change in the hydraulic and external losses. Any effort to obscure the condition with a single curve for the best efficiency point, as was always done, seems to give a finality to the results which is not warranted and would retard the more fundamental examination that is called for. The omission of the output-horsepower curve from the standard pump-performance curves which show head, efficiency, and brake horsepower as a function of flow rate at constant speed, is even for water objectionable, because it does not show some important characteristics of the pump which would lead to a more intelligent appraisal of its worth.

The value of the different maximum output horsepowers as a function of viscosity is really the function that the designer or user wants, after the QH function for maximum output horsepower is known. With the latter it gives the relative change of Q and H with viscosity. With the external losses determined for different viscosities, the hydraulic efficiency could be figured besides the over-all efficiency as was previously done to give a complete picture. Following this through for different types from zero to shutoff head would possibly shed some light on the very irregular flow at partial and overcapacities, not only on the maximum-output points hitherto compared. With this knowledge then, it might be possible to design better pumps for high-viscosity liquids, which oddly enough was successfully attempted in Professor Daugherty's first and still classical tests (1).

It might be argued and correctly so, that these notes are merely suggestions for a further program rather than criticism of the present paper. It is only fair to add that the author did make a valuable contribution and in more than one direction struck new paths, particularly in calling attention, to the writer's knowledge for the first time, that the wearing ring friction is a major external loss, which cannot be lumped together with the disk friction as a small additional loss.

L. F. Moody.[6] The effect of the viscosity of the fluid on pump performance has remained a problem without a satisfactory answer, and it has not been possible for pump designers to predict with any assurance the modification of water performance to be expected when pumping oils or other fluids. The existing data have been limited to more or less isolated tests and few comparisons under controlled conditions have been available. When looking for the effect of the change of a single variable, reliable conclusions can hardly be deduced from tests on separate pumps, for example, or from any tests where other variables do not remain identical. It is believed that the author's research is the first systematic study of the problem under carefully controlled conditions covering a wide enough range to establish definite conclusions.

The paper provides a wealth of data although, as mentioned by the author, further analysis is still needed to reduce the experimental results to readily usable form. The test data are now at hand, but the complex nature of the problem does not make it easy to apply the results to any specific case. It would be a valuable addition to the material included if sectional views of the pumps tested could be appended to the paper, as this information is necessary for complete analysis.

The method of analysis adopted in the paper is logical and sound, namely, the plotting of the performance with respect to a Reynolds number, which reduces the results to dimensionless form. Several useful forms of Reynolds number are used, one of which, R_D, depends upon the impeller speed and reflects directly the effects of disk friction and clearance friction, while other forms such as R_S and R_P reflect more directly the flow velocities. The plotting of the proportional-energy loss $\frac{100 - e}{100}$, or as the writer would prefer to express it, $1 - e$, (using e as a fraction rather than a percentage) is also a reasonable procedure.

With these data available, it would seem possible to arrive at some simple empirical rule which would permit us to modify the efficiency and head given by water performance to apply to oil, at least for light oils, over a moderate range of viscosities. The writer has found it difficult to read the small-scale logarithmic plottings, in the effort to find a satisfactory functional relation. Perhaps a function of some such form as the following may serve the purpose

$$e_W - e_0 = K(1 - e_W)\left(\log \frac{R_{DW}}{R_D}\right)^2$$

in which e_W = efficiency for water (as a fraction, not a percentage); $R_{DW} = R_D$ for water; and K is a coefficient, depending upon specific speed and perhaps other characteristics of the pump.

It is hoped that the author will be able to continue his own study of the problem, and to give us an easily applied rule which would serve at least as an approximation close enough for practical use.

The fact that the curves flatten out in the region of water performance, so that the behavior then practically ceases to be a function of Reynolds number, confirms the point of view expressed by the writer in the past, namely, that when handling water, the flow in usual sizes of pumps of normal specific speeds, and still more clearly in hydraulic turbines, falls in the zone of practically complete turbulence. Accordingly, the efficiency for water operation should be practically independent of the head, and be a function merely of the relative roughness of the surfaces exposed to the flow.

The research represented by this paper is an example of systematic hydraulic experimentation at its best, and care and thoroughness are evident throughout the undertaking. The results are valuable contributions to our knowledge.

R. L. Daugherty.[7] The author has investigated centrifugal-

[6] Professor of Hydraulic Engineering, Princeton University; Consulting Engineer, Worthington Pump and Machinery Corporation, Harrison, N. J. Fellow A.S.M.E.

[7] Professor of Mechanical Engineering, California Institute of Technology, Pasadena, Calif. Fellow A.S.M.E.

pump performance with oil viscosity as high as 10,000 Saybolt seconds, which is a viscosity of 2200 times that of water. This is probably about as high as there is any practical need for going.

The writer carried on extensive investigations in this field and under the same range of viscosity in 1924 and 1926, and the results were published in two bulletins.[8] Those investigations showed very clearly that capacity, head, and efficiency all diminish with increasing viscosity and that the effects were more noticeable with small pumps than with large ones.

Another factor which must certainly have an influence upon the variation of pump performance with viscosity is the specific speed. The author has made a start in presenting this phase of the subject. However, neither size nor specific speed will tell all the story. Thus in Fig. 19, which the writer has replotted from his earlier data, may be seen the difference in the performances of two pumps of the same size and of substantially the same specific speed when operated with fluids of different viscosities. It is seen that the design, which is more efficient for a fluid of low viscosity, is less efficient for a fluid of high viscosity.

The author has correlated his data on the basis of Reynolds number, but with a separate set of curves for each individual pump. If the rpm were also constant the only variable would be kinematic viscosity such as the writer has used in Fig. 19. Thus in Fig. 19, herewith, two viscosity scales have been added, since viscosity is there the only variable entering into the Reynolds number. One of these viscosity scales is that for Saybolt seconds and the other is kinematic viscosity in centistokes. The former is used here because of its extensive employment in industry, but the latter is a scientific unit with more physical meaning. Thus since at 68.4 F the absolute viscosity of water is 1 centipoise, and its kinematic viscosity is 1 centistoke, the viscosity given in either centipoises or centistokes is at the same time an indication of the viscosity relative to water at that temperature.

The Reynolds number is computed by a formula in which N is revolutions per minute and d_2 is the impeller diameter in feet, while the kinematic viscosity is expressed in square feet per second. This viscosity is equal to the viscosity in centistokes multiplied by 0.00001076. The writer has adhered to this combination because it is the same used by the author, but in the writer's opinion it would be more convenient to compute this by $\dfrac{16.93\ ND_2^2}{\text{centistokes}}$ where D_2 is the diameter of the impeller in inches.

The specific speed has been computed as

$$N_s = \frac{\text{rpm} \times \sqrt{\text{gpm}}}{H^{3/4}}$$

This is obviously the same expression which the author has used for his values, but it is well to state this specifically since cubic feet per second or other units are often used instead of gallons per minute. In the case of a double-suction pump, it seems preferable to use one half of the total pump capacity in such a calculation, but the author does not indicate whether he has followed that practice or not in the values which he gives.

In Table 1 of the paper it would be desirable to define the meaning of the symbols there used since they are not obvious. In connection with Equations [2a, b, c, d], it should be stated that N is revolutions per minute, that d_2 is the diameter of the impeller in feet, although in Fig. 11, D is used for the same value.

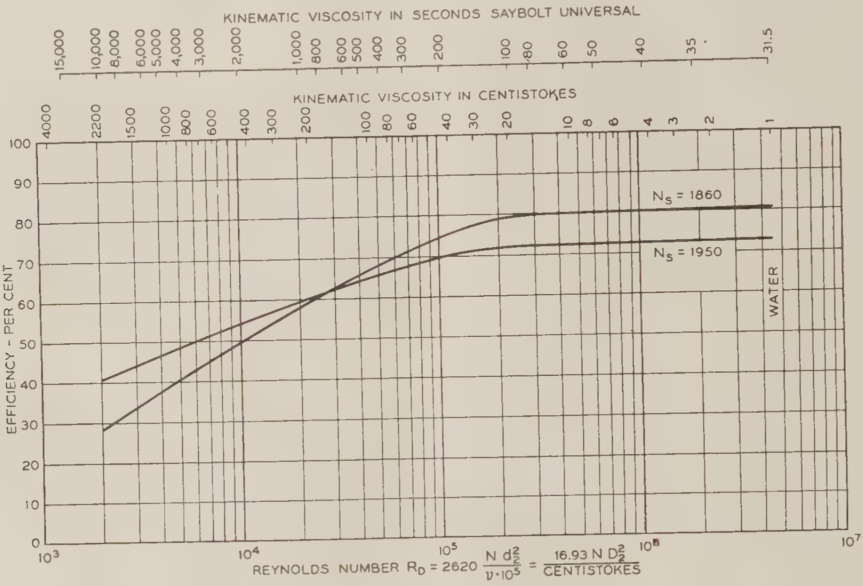

Fig. 19 Performance Curves for Two 8-In. Pumps
(Impeller diameter = 13.5 in.; 1450 rpm.)

It is to be hoped that the author or others may continue to obtain data on centrifugal-pump performances with different viscosities and with a range of sizes, specific speeds, and other design features so that eventually a comprehensive chart may be drawn up which would be a reliable guide for any combination of conditions. However, it is difficult to generalize with only a limited amount of data.

O. H. Dorer.[9] The author's contribution toward the solution of the problem of viscosity effects in centrifugal pumps indicates

[8] Refer to Author's Bibliography (1); an earlier bulletin was published in 1924.

[9] Consulting Engineer, Worthington Pump and Machinery Corporation, Harrison, N. J. Mem. A.S.M.E.

a method of attack attempted by the writer some time ago and since abandoned. By this is meant the head corrections based upon percentage of water performance, and some classification numbering system similar to Reynolds number by using impeller diameter and finally unit speed (n), which is a function of impeller diameter.

Subsequently, a formula for

$$R_N = \frac{14{,}100G^{1/2}}{\gamma} H_v^{1/4}$$

was adopted in which G = gpm, H_v = velocity head, γ = viscosity in centistokes. It is seen that the troublesome size factor has disappeared, and this is possible through substitutions in

$$Q = 0.785\, D^2 \sqrt{2gH_v}$$

the usual pipe formula, where H_v is the head creating the velocity.

In the centrifugal pump the velocity creates the head, so we can use a similar value for over-all Reynolds number. However, since the conception of H_v is from an original velocity determined not by the developed head, but by an internal head which overcame all head losses in the pump, the internal rather than the developed head is used. The internal head corresponds to input horsepower with a deduction for disk friction, bearing and stuffing-box loss. At zero capacity the recirculation capacity is considered in calculating internal head from the input power.

Of the input power, that part overcoming disk friction, bearing and stuffing-box loss is not a direct hydraulic-flow loss. A determination of this loss was made by placing solid dummy impellers in the pump casing having all details like the real impeller, except that the outer band normally used as discharge passage gives extra disk loss. Corrections were determined by extending the

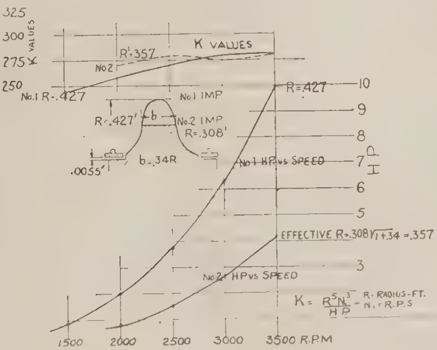

Fig. 20 Water Disk Horsepower of Dummy Impellers

(10-hp dynamometer used; tests include bearings, stuffing boxes, and sealing rings.)

dummy to a rounded-nose affair, as illustrated in Fig. 20 of this discussion. (Water) disk horsepower follows the form of

$$\text{Hp} = \frac{R^5}{K} N_1^3$$

where R = radius, ft; N = rps; K averages 280.

Zur Nedden[10] found with simple disk tests

$$\text{Hp} = \frac{N_1^{2.9} R^{4.9}}{420} \text{ (average value)}$$

However, Gibson and Ryan tests are reported at

$$\text{Hp} = \frac{N_1^{2.9}\, R^{4.9}}{330}$$

Both of these are simple disks, polished, and in rough cast-iron surroundings; and neither of them can be considered as reflecting all the conditions surrounding impeller surfaces, ring, joints, etc. Hence their low values of horsepower are not considered to be applicable to centrifugal pumps.

In order to arrive at proper wall pressures, the dummy impellers were tested under pressures at mid-radius, as recorded by the actual impeller during its operation.

Disk horsepower bears a relationship to specific speed, a low specific speed having relatively more of this loss than higher-specific-speed pumps. The build-up of disk horsepower with viscosity may be very high on a low-specific-speed pump.

While ring-leakage loss is reflected in the net output, it is proper to allow for it in obtaining H_v. Practically, for water performance, H_v may be determined from H_a (actual head) by dividing it by the hydraulic efficiency, here defined as efficiency, not including disk horsepower. In any reasonable size pump ring leakage[11] is a small percentage; it may be neglected.

However, with viscous oils this ring leakage decreases to some extent, and in small pumps its decreasing rate could improve the oil head obtained, above water performance.

The basis for considering effects of viscosity states: (*a*) The increase in power input, corrected to unity specific gravity, is due to increased disk friction, see Fig. 23 of this discussion; (*b*) the decrease in head is due to several features, i. e., the frictions in the passages, the lag in attaining velocity imparted by the blades of the wheel, and finally, the poorer efficiency of velocity-head conversion within the pump.

While equal-size pumps are built with various spiral velocities, some relatively high, others relatively low, for similar specific speeds, it is clear the friction in the large-area spiral will not build up as fast with increase in viscosity as is the case in smaller-area spirals.

A similar situation applies to impeller passages; fewer vanes infer larger hydraulic mean radius and lesser frictions.

With both elements designed for low velocities to reduce frictions, there is an adverse effect in the hydraulic efficiency, and of velocity-head conversion. The problem in viscosity then boils down to which element is most destructive. It will be shown that frictions are more detrimental than the destruction of velocity-head efficiency in some designs of pumps.

Manifestly, each of these elements has a Reynolds number, and various rates of friction loss or head destruction; and while we apparently have a Reynolds number for the particular pump, there does not appear to be an over-all Reynolds-number system applicable to the range of pumps that may be used for viscous-oil applications in a manner similar to the pipe problem.

In this connection, the theory of action in a centrifugal impeller poses another problem. If all head is developed entirely through velocity, H_v is directly determined. However, some of the head is composed of centrifugal force which is at a maximum at the shutoff and at a minimum, at large flows. In fact, when centrifugal force is absent, cavitation begins. Principal formulas for centrifugal force are given as follow

$$H = \frac{YU_2}{g}(YU_2 - W) = \frac{\text{input hp} - \text{disk hp}}{0.1135\,(Q + \text{ring leak } Q)} \quad ..[20]$$

$$\text{Per cent centrifugal force} = 100 \times \left(\frac{H}{1.22\, N_1^2 R^2} - 0.5Y^2\right) \quad ..[21]$$

[10] "Influence of Disk Friction on Turbine-Pump Design," by F. Zur Nedden, Trans. A.S.M.E., vol. 37, 1915, pp. 83–107.

[11] Ring leakage was $2^1/_8$ per cent for the 4-in. pump in which the dummy impeller was tested, and is independent of speed, for a given impeller.

where

U_2 = peripheral velocity, fps
W = horizontal backward component in usual velocity diagram, neglecting contraction
Q = volume delivered, cfs
R = radius impeller, ft
N_1 = revolutions per sec
H = impending head, ft

With the centrifugal-force element deducted, H_v is of a lower value and the viscosity would not affect centrifugal force except through additional lag of vane action on the fluid. In all following comparisons it has been assumed that no lag occurs.

The 8-in. Mather and Platt pump-head results have been investigated for viscosity at three rates of flow, Fig. 21 of this discussion. The sudden bump in the curve may be caused by change of oil, even though the viscosity scale is continuous. It is suspected that the type of oil has some influence. Here the hydraulic efficiencies have been calculated on the assumption that all generated head comes from velocity conversion. At the top of the sheet a series of n exponentials indicates variation with constant apparent Reynolds number.

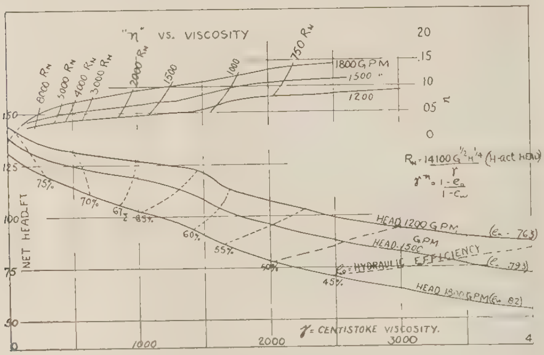

Fig. 21 Pump-Head Results on Mather & Platt 8-In. Pump

Pipe frictions may be compared where viscosity is the only variable by an exponential n of viscosity; it is nearly 0.20. For head conversion in the centrifugal pump, this same exponent of viscosity is seen to be variable and considerably below 0.20.

The low n values of head, compared to frictional n values, infers that the head conversion which is the major part of H, does not decrease as fast with increase of viscosity as frictions increase. With the very heavy viscosities, the increase in n infers greater destruction in conversion of velocity to head than at low viscosities.

In order to substantiate these facts, the frictional slopes for impeller and spiral were calculated and allowed for, as well as the centrifugal force, and the remaining values represent hydraulic efficiency of velocity-head conversion.

$$\text{Friction slope} = \frac{2.7\, f H_v^{5/4}}{G^{1/2}}$$

through substitutions for size indicated in an earlier part of this discussion.

Considering the centrifugal-force element we can write

$$1 - \frac{2.7 f H_v^{5/4}}{G^{1/2}} = e_v = \frac{H_a - \text{CF} + F_r}{H - \text{CF}}$$

H_a = actual developed head
CF = centrifugal-force head

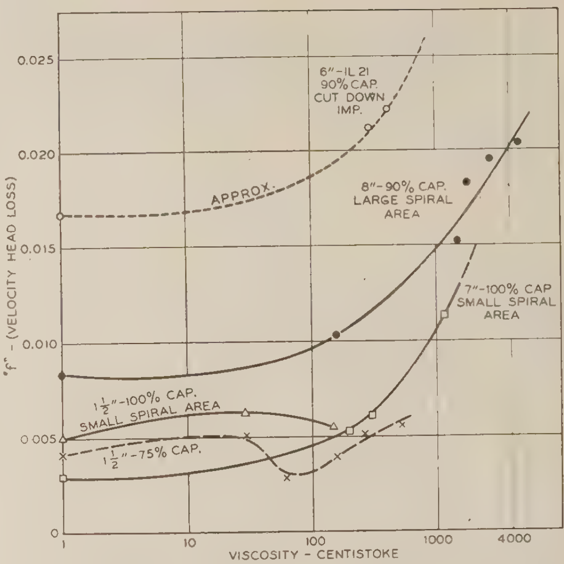

Fig. 22 Frictional Slopes for Pump Impeller and Spiral

F_r = friction loss in spiral and impeller
H = impending head
C_v = velocity-to-head conversion efficiency factor

Here f is similar to the friction factor of pipe calculations, but refers specifically to a friction factor of velocity-head conversion.

Fig. 22 of this discussion gives the results of such an analysis and confirms that velocity-head conversion is not destroyed as rapidly in some designs as friction builds up. The small-spiral-area pumps are more efficient as water pumps having low f values.

The small $1^1/_2$-in. pump seems to have better f values at 75 per cent capacity; however, this illustrates the ring-leakage effects which have been neglected, and which bear a larger percentage to output than is the case in large pumps. The decrease in leakage with increase of viscosity in the small pump is responsible for the apparent improvement in f values.

Confirming that friction and velocity-head destruction of head on viscous oils do not proceed at equal rates, the 8-in. and 7-in. pumps are compared (Table 3 of this discussion) on a common flow, and viscosity, viz. 1500 gpm at 1000 centistokes viscosity. The velocity through the spiral of the 7-in. pump is about $2^1/_4$ times that of the 8-in. pump. (The IL-21 pump is shown at 1050 gpm as 930 centistokes).

TABLE 3 COMPARISON OF 7-IN. AND 8-IN. CENTRIFUGAL WITH 6-IN. IL21 PUMP

	7-in. pump	8-in. pump	6-in. IL21 pump
H water	165	135	80
H oil	108.5	133	$62^1/_2$
Per cent head loss	34.2	16.2	21.9
Hydraulic efficiency, water, per cent	91.3	77.3	81.0
Net efficiency, water, per cent	85.0	73	80
Velocity efficiency, water	90	68	73.3
Velocity efficiency, oil	63.5	59	57.0
Friction, water	1.75	1.0	1.0 (estimated)
Friction, oil	23.8	13.2	9.8
f_w	0.00323	0.0119	0.0168
f_o	0.0118	0.0152	0.271
m exponential	0.302	0.303	0.46
Centrifugal force, ft	49.0	57.0	32.0 (w estimated at 10)

While the friction losses increased by an amount equal to 13.4 per cent of the developed water head, the loss in converted

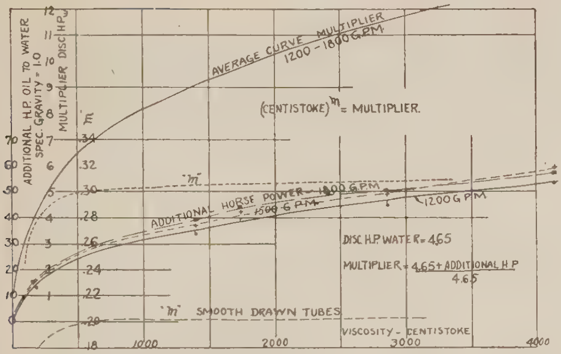

FIG. 23 EFFECTS OF INCREASING VISCOSITY ON PUMP PERFORMANCE

velocity head increased 21 per cent for the 7-in. pump. On the other hand, the 8-in. pump increases are 9.05 per cent and 6.6 per cent, respectively, for these items. The high f_w value of the IL21 pump is undoubtedly due to the large cutdown of impeller, resulting in a relatively long space between impeller and spiral. This space does not appear to affect results as badly on viscosity as in the case of the 7-in. pump, but more so than the 8-in. pump.

The changes in rates by which each element behaves in the different types of designs appear to nullify a universal application of over-all Reynolds numbering system with head and efficiency correction factors applied to the water performance.

The m exponentials affecting power are likewise variable; some high-head per stage pumps showing as low as 0.18. This suggests that rapidly moving disk surfaces create a film of apparently low viscosity or heating at the surfaces.

In conclusion, it appears necessary that more test results be provided to develop acceptable designs for best oil performances; and specific methods of calculation then are a matter of choice.

AUTHOR'S CLOSURE

The discussions submitted have contributed greatly to the further clarification of the complex problems involved in analyzing the influence of viscosity on the behavior of centrifugal pumps. The discussers have raised relatively few controversial points and have contributed instead from their own experiences valuable observations and suggestions for future work which need little additional comment. The author indicated at various times in the paper that the exploitation of the test data was only partly completed. Considerable information should be gained from further critical analysis of the individual performance curves with respect to separation of "external" from so-called "internal" losses, by comparisons with results of other experimenters, and by investigating the influence of such findings on established design procedures. The present paper was intended primarily to establish a common basis of discussion on the strength of extensive experimental evidence, to clarify the basic nature of the various losses, and to estimate their relative influence. The author feels greatly reassured since his line of attack was not seriously questioned. The work may now be continued toward eventual correction curves of more general validity.

A few remarks are in order to answer some specific problems raised by the various contributors. Mr. Tetlow brings up the question whether or not the specific speed N_S should remain constant regardless of viscosity: Curves drawn on the basis of N_S = const (i.e., $\frac{Q}{H^{3/2}}$ = const for a given pump and a given speed) in the head-discharge curves give maximum efficiency points for high viscosities better than the author's method of keeping $\frac{Q}{\sqrt{H}}$ constant. Naturally various methods of defining normal capacity and normal head were discussed and tried during the analysis of the data and the present method was adopted essentially for practical reasons. Up to relatively high viscosities the efficiency corrections obtained by either method are practically identical. Applications where the efficiencies drop below 30 per cent or 25 per cent are not expected to be numerous. The efficiencies can be kept above these values by changing the type or the design of the pumps (see Professor Daugherty's example).

It seems contrary to experience with fluid friction that a single law should be valid from the range of "complete turbulence" through the turbulent transition zone into the completely laminar state of the "through flow." While reduced capacity and head result in a corresponding shift of the peak horsepower output, the higher rate of increase in the so-called external losses, especially disk and ring friction, will tend to produce a slower corresponding shift of the peak efficiency. The author agrees with the discussers who have pointed here to the need of further analysis so that the influence of the internal losses may eventually be separated from that caused by external losses.

For the present the method adopted seems the most useful one, since the assumed proportionality of Q and $\sqrt{H}$ enables the engineer to calculate these values for small changes in speed N and impeller diameter d_2, in accordance with established procedures derived from the affinity laws. It may also be called to mind that the usual concept of specific speed is associated with similar flow conditions in a given pump regardless of speed, resulting in the proportionality of speed to Q and $\sqrt{H}$ with constant proportionality factors. This precludes any influence of the viscosity on the flow conditions. If with increasing viscosities such influences result in a marked change in these proportionality factors, the geometric similarity of the through flow is disturbed though the boundaries and the speed may remain fixed.

For all practical purposes then, the pumping process is changed and it is hard to conceive of specific speed remaining constant under these conditions. The pump type may still be classified, however, by its water-specific-speed, and correction curves may be drawn on that basis. Since the change in the specific speed on the basis of $Q = K\sqrt{H}$ is inversely proportional to the discharge correction factor, the resulting increase in the specific speed remains relatively small. Finally, it may be added that the increasing external losses also have a certain effect on the specific speed, since they affect the location of the maximum efficiency point.

The exact significance of specific speed therefore is not too clear in connection with the problem of viscous influences on pump performance. It represents a classification number for normal water performance, where it remains constant for a given type of pump, since flow conditions remain geometrically similar regardless of speed and diameter. This similarity is due to the almost complete absence of variations in the viscous influences and to the minor variations of the external losses. This similarity will break down if the viscosity increases greatly and it seems reasonable to acknowledge this fact by a change in the specific speed.

The preparation of correction curves is not greatly affected by the definitions and correlations of normal discharge and normal head. The author agrees, however, with Mr. Tetlow in the desirability of further analysis and of theoretical investigations on the foregoing question.

Professor Hollander and Professor Daugherty have inquired into the definition of specific speed concerning single-suction and double-suction pumps. The author calculated the values of N_S

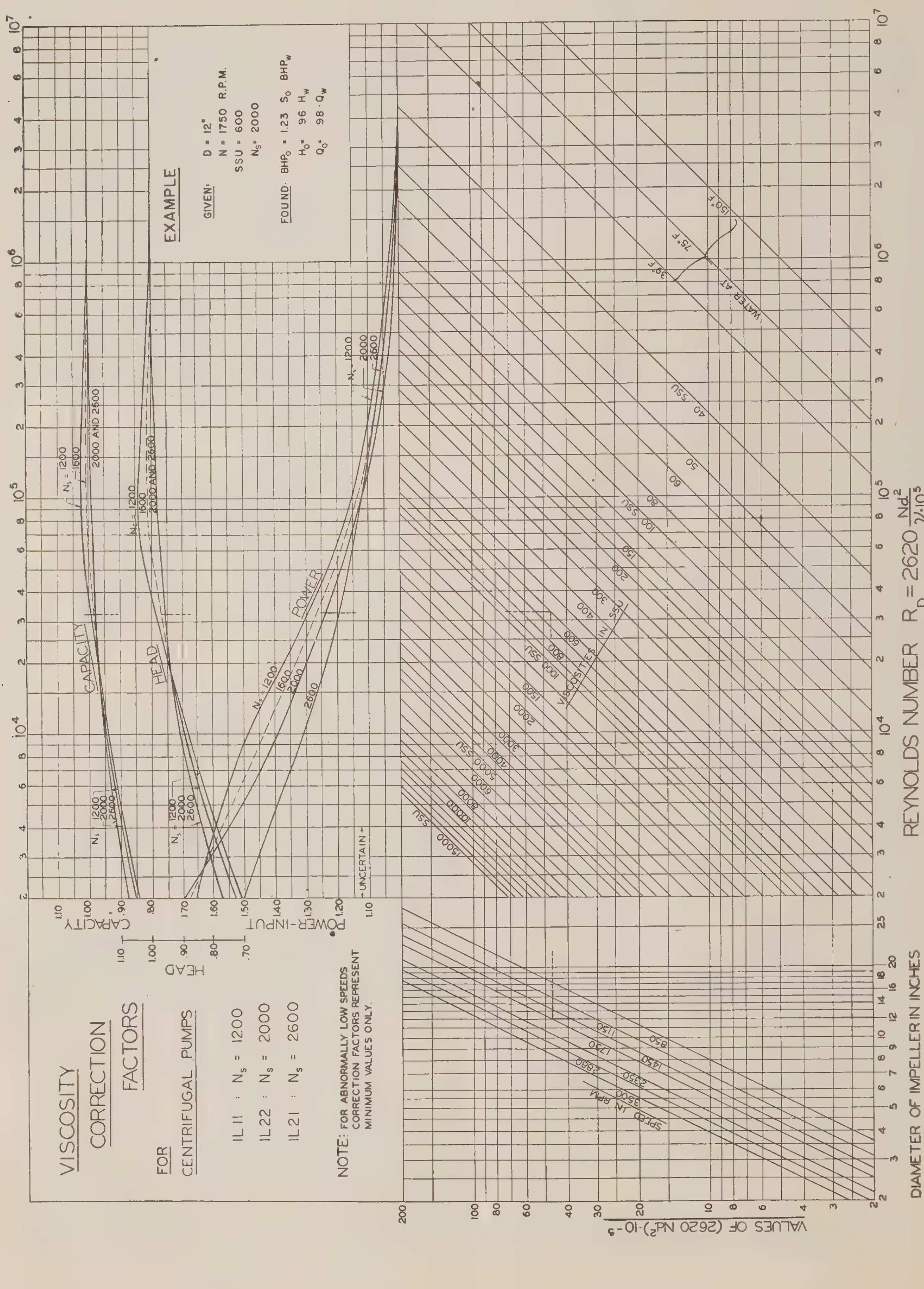

FIG. 24 GRAPHICAL METHOD OF DETERMINING R_D

using total discharge Q in gpm, H in ft, and N in rpm. It is believed that the distinction between single- and double-suction pumps need not be insisted upon in connection with the problem in question. Using only one half the capacity for a double-suction pump to obtain the characteristic specific speed is in order whenever problems concerning entrance conditions and especially cavitation phenomena are investigated. For the problem under discussion it is doubtful whether a better comparison to single-suction pumps would be obtained except for very high specific speeds. If a double-suction impeller were to be replaced by two single-suction impellers, the hydraulic losses would probably be changed very little; however, the horsepower input for each impeller would have to be corrected for greatly increased external losses, since disk- and ring-friction losses would be doubled. The corresponding decrease in the efficiency would offset the correction of the specific speed. Such corrections for external losses for these equivalent impellers could not be made in the present state of knowledge.

For water performance disk- and ring-friction losses form usually only a small part of the total losses, which are essentially hydraulic or internal losses there, so that such corrections would be of negligible order of magnitude. For these reasons it seemed preferable to state the specific speeds for the double-suction pumps (IL21 and IL22) on the basis of their full capacity. Further light on this question must come again from a possible separation of so-called external and internal losses.

Professor Hollander has given an admirable statement and summary of the consequences of the viscous changes on the affinity relations and points out the direction toward the considerable work ahead in this field. It is clear that the conventional thinking on these affinity questions will have to be revised considerably, whenever viscosity plays a major part in pump performance.

Professor Daugherty has plotted his test results against the Reynolds number R_D and since both sets of tests were run at the same speed, this plot is identical with that against the viscosity as the only variable in R_D. It should be noted carefully that the author's curves in Figs. 14 to 16 represent data for three different speeds and therefore correlate the test material for each pump on the basis of corresponding N/ν values, with the diameter being the only constant quantity in R_D. Professor Daugherty's pump of higher efficiency for high viscosities represents the first successful attempt at a special design for viscous influences and is therefore not directly comparable to the normal pump built for highest efficiency with water.

To facilitate further comparisons with tests of other experimenters, a graphical way of determining R_D is given in Fig. 24 of this closure, with corresponding results from the author's tests integrated into three sets of correction curves for head, capacity, and power. The efficiency corrections are not shown, but can be easily obtained for any Reynolds-number value by multiplying the head and capacity corrections and dividing the product by the power-input correction factor. The use of the chart is as follows: From a known value of the impeller diameter in inches on the left scale proceed vertically upward to the intersection with the proper speed line in rpm. From here continue horizontally across to the right to the intersection with the line marked with the given viscosity in SSU (Saybolt seconds Universal). The value of R_D may now be read on the bottom scale. A vertical line extended up will locate immediately the correction factors for capacity, head, and input power.

These correction curves represent the average values from the author's tests obtained for the higher speeds which are normally employed. The individual curves are identified by the corresponding approximate specific speeds. The diagram may thus serve as a more easily applied method for practical approximations, for which Professor Moody and Mr. Tetlow have asked. The user may be reminded, however, that no claim is made as to the general validity of the correction factors, since these are based only upon the tests and the methods of analysis reported in the paper. Material from other sources may be easily compared with these findings.

Professor Moody has offered the form of an efficiency correction function which seems basically sound. The author is especially in favor of his use of $(e_w - e_0)$ therein, which represents the absolute decrease of the efficiency due to the influence of viscosity. From some preliminary graphical studies of tests on many different pumps, it seems that $(e_w - e_0)$ produces the relatively best correlation of the results, again pointing to the predominant increase of external losses especially for lower viscosities.

The author gladly complies with Professor Moody's suggestion of showing the pumps tested in sectional views by courtesy of the Ingersoll-Rand Company. Fig. 25 presents the pump labeled IL11, which indicates the closed impeller, and Fig. 26 shows the pump IL22. It is to be recalled that pump IL21 differed from the latter only by having an impeller of smaller diameter.

Little can be added to Mr. Dorer's remarks which are concerned extensively with the analysis of individual internal and external losses. The author prefers the method of determining disk friction indicated in Fig. 11 of the paper, which is based upon sound analytical and experimental evidence and would like to call attention to the considerable influence of wearing-ring friction. Mr. Dorer's Reynolds number R_N is identical with the author's R_P, as can easily be verified by substituting for $(H_v)^{1/4}$

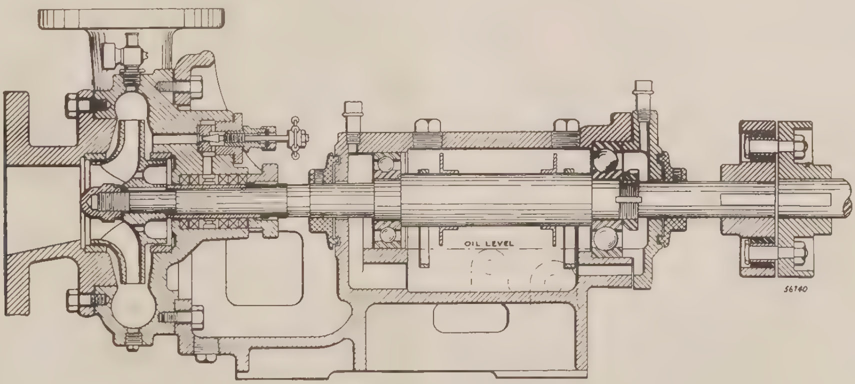

FIG. 25 CLOSED-IMPELLER-TYPE PUMP IL11

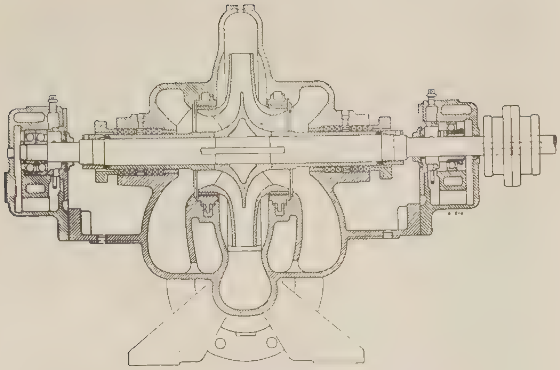

Fig. 26 Double-Suction Centrifugal Pump, IL22

in terms of $\left(\frac{Q_0}{D^2}\right)^{1/2}$ and by changing the constants corresponding to units employed. This Reynolds number differs from R_D only by the proportionality factor relating Q_o to ND^3. Since these proportionality factors are dependent upon the viscosity and Reynolds number, their use in defining a Reynolds number is impractical, because trial-and-error methods have to be employed in solving for the correction factors. Reynolds number R_D is determined by diameter, speed, and viscosity and immediately locates the proper correction. Aside from this practical advantage, R_D is of particular significance for disk and ring friction as pointed out before.

In conclusion the author wishes to express his appreciation for the interesting contributions made in the discussion. They have helped greatly to stimulate and to clarify further thought on the complexities inherent in this problem of viscous effects on centrifugal-pump performance.

APPENDIX

Auxiliary Table of Notations

Tables 1 and 2 (all dimensions in inches):

D_s = diameter of suction pipe
D_d = diameter of discharge pipe
D_1 = diameter of entrance section of impeller
D_2 = diameter of exit section of impeller
D_{sh} = diameter of shaft
B_2 = width of exit section of impeller
D_R = diameter of wearing rings
B_R = width of wearing rings
$2\,S_R$ = diametral clearance of wearing rings

Fig. 7:

$$M = \frac{a_2\sqrt{2g}}{\sqrt{1 - \frac{(d_2)^4}{(d_1)}}}$$

h = differential head of liquid, ft
Subscript (2) refers to orifice and Venturi-throat diameter; subscript (1) to diameter of line of approach. For water flowing through orifice $C_D = 0.6185$, and $K_M = 1.00$

Equations [1 to 17]:

H_w = head produced by pump for water, ft
H_0 = head produced by pump for oil, ft
Q_0' = discharge, cfs
Q_0 = discharge, gpm
d_{i_e} = equivalent eye diameter of impeller, ft
v_i = velocity in equivalent eye section, fps
d_2 = impeller diameter, ft
b_2 = impeller width, ft
ω = angular velocity in radians per sec
N = rpm of disk or impeller
u_2 = tangential velocity at impeller exit, fps
v_2 = relative velocity at impeller exit, fps
r = radius of disk or impeller
δ = thickness of boundary layer near disk
S_D = clearance between disk and housing wall
d_R = diameter of rings, ft
b_R = width of wearing rings, ft
S_R = clearance between rings
γ = specific weight of fluid, lb per cu ft
S_0 = specific gravity of oil

Fig. 10:

$\omega \cdot r$ = tangential velocity of disk
u = tangential velocity of liquid in boundary layer
v_r = radial velocity of liquid in boundary layer
$\tan \alpha_2 = \frac{v_r}{u}$
z = distance from disk surface within boundary layer ($z_{max} = \delta$)

Fig. 11

D = diameter of disk in ft.

The Theory of Moving Sources of Heat and Its Application to Metal Treatments

By D. ROSENTHAL,[1] CAMBRIDGE, MASS.

The theory of moving sources of heat has been instrumental in providing the welding engineer with a scientific criterion of weldability of steels. The author outlines briefly the fundamentals of this theory and derives appropriate solutions for linear, two- and three-dimensional flow of heat in solids of infinite size or bounded by planes. Point, linear, and plane sources of heat are examined. The solutions obtained are then applied to welding problems. It is shown that these solutions are in good agreement with the experimental results, and that they afford a close analysis of the factors governing the heat flow in welding. The most interesting result of the theory, however, is the derivation of a single formula capable of predicting the time and rate of cooling with a fairly good accuracy for a wide variety of thicknesses of steel, ranges of temperature, and welding conditions. An attempt has been made also to show how this theory could be applied to other problems of metal treatment, such as rate of extrusion in continuous casting, or control of flame-hardening and continuous quenching operations.

Introduction

THE theory of heat flow due to a moving source so far has retained little attention in the general treatment of heat flow in metals. Yet the moving source of heat plays an important part in many metallurgical processes, notably in welding. In this latter case the theory of heat flow due to a moving source has been instrumental in establishing a scientific criterion of weldability (1).[2] Similar applications of the theory are feasible in other metallurgical processes, for example, in surface-hardening or continuous casting; but the somewhat involved mathematical analysis makes it rather hard for the industrial engineer to grasp the practical implications of the theory. The purpose of this paper is to overcome this shortcoming by putting more emphasis on applications. The first part deals briefly with the mathematical aspect of the theory; the second part describes the experimental work which has been carried out in connection with the applications to welding; and the third part outlines some of the feasible applications in other metallurgical branches.

Previous Work

Apparently the theory of heat flow due to a moving source originated in connection with arc welding. Many attempts, both experimental and theoretical, had been made to describe the temperature situation created by the arc welding. Most of these early attempts have been reviewed previously (2). Because of their approximate nature, there is little point in discussing them in the present paper.

The exact theory of heat flow due to a moving point source was first applied to arc welding by the author (3) in 1935, using the experimentally established principle of a "quasi-stationary" state (4). A particular case of the general solution was treated independently by Boulton and Lance Martin (5) in 1936. More recently, Bruce (6) applied the method of instantaneous sources to another particular case of welding and, in 1941, Mahla (7) extended this method to a three-dimensional case. The solution derived by Mahla does not differ materially from that obtained previously on the assumption of the quasi-stationary state, but the method is applicable also to nonquasi-stationary states, as shown in a later section. Unfortunately, in most cases the solution becomes too unwieldy for a direct practical application.

1 Theory of Heat Flow Due to a Moving Source

The Differential Equation of the Quasi-Stationary State

Nomenclature. The following nomenclature will be used throughout the paper:

q = rate of heat, for ex., cal/sec
q' = rate of heat per unit length
q'' = rate of heat per unit section
v = speed of source
T = temperature
t = time
k = heat conductivity of metal. When expressed in C.G.S. units the dimensions of k are cal/sec cm deg C
$1/2\lambda$ = thermal diffusivity of the metal.[3] The dimensions of diffusivity are $\frac{\text{sq cm}}{\text{sec}}$ in C.G.S. units. On the other hand, if c is specific heat and ρ density, $2\lambda = \frac{c\rho}{k}$

Assumptions. Assumptions will be made as follows:

(a) The physical characteristics of the metal, i.e., k and λ are independent of the temperature.

(b) The speed v and the rate of heat input q are constant.

The Quasi-Stationary State. The differential equation of heat flow expressed in rectangular co-ordinates (x, y, z) which are referred to a fixed origin in the solid, has the well-known form

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 2\lambda \frac{\partial T}{\partial t} \qquad [1]$$

Assume now that the heat is supplied by a point source of strength q moving with a constant speed v along the x-axis, and determine the temperature distribution around the heat source. This amounts to writing the differential Equation [1] with the point source as origin. To this end, replace in Equation [1] x by $\xi = x\text{-}vt$, where ξ is the distance of the point considered from the point source, and differentiate with respect to ξ. There follows

$$\frac{\partial^2 T}{\partial \xi^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = -2\lambda v \frac{\partial T}{\partial \xi} + 2\lambda \frac{\partial T}{\partial t} \qquad [2]$$

Experiment shows (4) that if the solid is long enough as com-

[1] Massachusetts Institute of Technology.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Heat Transfer Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 17–20, 1946, of The American Society of Mechanical Engineers.

[3] The notation by means of a converse symbol multiplied by 2 is used here for the sake of convenience as will appear later.

pared to the extent of heat, the temperature distribution around the heat source soon becomes constant. In other words, an observer stationed at the point source fails to notice any change in the temperature around him as the source moves on. This state of heat flow is called quasi-stationary, and is defined by Equation [2] in which $\frac{\partial T}{\partial t} = 0$, i.e.

$$\frac{\partial^2 T}{\partial \xi^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = -2\lambda v \frac{\partial T}{\partial \xi} \quad [3]$$

Equation [3] can be simplified by putting

$$T = T_0 + e^{-\lambda v \xi} \varphi(\xi, y, z) \quad [4]$$

where T_0 is the initial temperature of the solid and φ is a function to be determined. After substitution of Equation [4] in [3] and reduction of terms, the following equation is obtained

$$\frac{\partial^2 \varphi}{\partial \xi^2} + \frac{\partial^2 \varphi}{\partial y^2} + \frac{\partial^2 \varphi}{\partial z^2} - (\lambda v)^2 \varphi = 0 \quad [5]$$

or in the usual symbolic form

$$\nabla^2 \varphi - (\lambda v)^2 \varphi = 0 \quad [6]$$

Thus Equation [1] is reduced to a relatively simple form, which provides ready solutions for an infinite and semi-infinite solid, as will be shown presently.

Infinite and Semi-Infinite Solid

Linear Flow of Heat. In this case $\frac{\partial T}{\partial y} = \frac{\partial T}{\partial z} = 0$, and Equation [5] is reduced to

$$\frac{d^2 \varphi}{d \xi^2} - (\lambda v)^2 \varphi = 0 \quad [7]$$

The boundary conditions are

$$\frac{dT}{d\xi} \to 0 \quad \text{for} \quad \xi \to \pm\infty \quad [8]$$

Furthermore, considering a plane source with a rate of heat q'' per unit area

$$-\frac{dT}{d\xi} \times k \to q'' \quad \text{as} \quad \xi \to 0 \quad [9]$$

A general solution of Equation [7] can be written as follows

$$\varphi = C_1 e^{-\lambda v \xi} + C_2 e^{\lambda v \xi} \quad [10]$$

hence

$$T - T_0 = C_1 e^{-2\lambda v \xi} + C_2 \quad [11]$$

With reference to Equations [8] and [9] it is easy to see that $C_1 = 0$ when $\xi < 0$, hence for $\xi < 0$

$$T - T_0 = C_2 \quad [12]$$

Likewise $C_2 = 0$, when[4] $\xi > 0$, hence for $\xi > 0$

$$T - T_0 = C_1 e^{-2\lambda v \xi} \quad [13]$$

Also, since $T - T_0$ must have the same value for $\xi = 0$

$$C_1 = C_2$$

On the other hand, with reference to Equation [9]

$$C_1 2\lambda v k = q''$$

[4] This because $T = T_0$ for $\xi = \infty$.

or, since

$$2\lambda k = c\rho$$

$$C_1 = C_2 = q''/c\rho v \quad [14]$$

hence for $\xi < 0$

$$T - T_0 = \frac{q''}{c\rho v} \quad [15]$$

and for $\xi > 0$

$$T - T_0 = \frac{q''}{c\rho v} e^{-2\lambda v \xi} \quad [16]$$

From Equation [16] it follows that $q''/c\rho v$ is the rise of temperature at the location of the source, i.e., for $\xi = 0$, and corresponds to the maximum value of temperature reached in the solid. Equation [15] shows that once this temperature has been reached, it remains even after the source has moved further. This, of course, is the result of lack of surface losses. Calling T_1 the maximum temperature in the solid, Equations [15] and [16] become finally

for $\xi < 0$

$$T = T_1 \quad [17]$$

for $\xi > 0$

$$T = T_0 + (T_1 - T_0)e^{-2\lambda v \xi} \quad [18]$$

Heat Dissipation; Infinite Rod. If temperature gradients in the cross section of the rod are neglected, Equations [17] and [18] can be made to include heat losses through the surface. Assuming Newton's law of radiation, the rate of heat losses through unit surface area is given by the following formula

$$h(T - T_0) \quad [19]$$

Here T is the temperature of the surface and T_0 is the temperature of the surrounding medium which we shall assume to be the same as the initial temperature of the rod; h is the heat dissipation. In problems of heat flow in metals it is customary (8) to use the ratio of heat dissipation to heat conductivity

$$H = \frac{h}{k} \quad [20]$$

H will be called the dissipation ratio.

With this in mind, the differential equation of linear heat flow with surface losses has the following form (9)

$$\frac{\partial^2 T}{\partial x^2} = 2\lambda \frac{\partial T}{\partial t} + \frac{PH}{A}(T - T_0) \quad [21]$$

Here P is the perimeter and A the area of the cross section of the rod. For the case of the quasi-stationary state this equation can be reduced, as easily seen from Equation [6], to

$$\frac{d^2 \varphi}{d\xi^2} - \left[(\lambda v)^2 + \frac{PH}{A}\right]\varphi = 0 \quad [22]$$

and the solution corresponding to Equations [17] and [18] can be rewritten to read

for $\xi < 0$

$$T = T_0 + (T - T_0)e^{(\sqrt{(\lambda v)^2 + PH/A} - \lambda v)\xi} \quad [23]$$

and for $\xi > 0$

$$T = T_0 + (T - T_0)e^{-(\sqrt{(\lambda v)^2 + PH/A} + \lambda v)\xi} \quad [24]$$

The rate of heat per unit area delivered to the rod is correspondingly

for $\xi < 0$

$$q_1'' = k(T_1 - T_0)[\sqrt{(\lambda v)^2 + PH/A} - \lambda v] \ldots\ldots [25]$$

for $\xi > 0$

$$q_2'' = k(T_1 - T_0)[\sqrt{(\lambda v)^2 + PH/A} + \lambda v] \ldots\ldots [26]$$

Various applications of the foregoing equations will be found in Parts 2 and 3 of this paper.

Two-Dimensional Flow of Heat. In this case the heat flows in two directions, one of which is the x- or ξ-direction. The other direction will be taken along the y-axis, and there will be no flow in the z-direction. Thus $\partial T/\partial z = 0$, and Equation [5] becomes

$$\frac{\partial^2\varphi}{\partial\xi^2} + \frac{\partial^2\varphi}{\partial y^2} = (\lambda v)^2\varphi \ldots\ldots\ldots\ldots [27]$$

The boundary conditions are

$$\left.\begin{array}{ll} \partial T/\partial\xi \to 0 & \text{as } \xi \to \pm\infty \\ \partial T/\partial y \to 0 & \text{as } y \to \pm\infty \end{array}\right\} \ldots\ldots\ldots\ldots [28]$$

and, by considering a circle $2\pi r$ drawn around the heat source with $r = \sqrt{\xi^2 + y^2}$

$$-\frac{\partial T}{\partial r} 2\pi rk \to q' \quad \text{for } r \to 0 \ldots\ldots\ldots\ldots [29]$$

where q' is the rate of heat of a linear source.

Because of the nature of the boundary conditions and the symmetrical form of Equation [27] with respect to ξ and y, φ depends only upon the distance r from the heat source, hence Equation [27] becomes in cylindrical co-ordinates[5]

$$\frac{d^2\varphi}{dr^2} + \frac{1}{r}\frac{d\varphi}{dr} - (\lambda v)^2\varphi = 0 \ldots\ldots\ldots\ldots [30]$$

The solution of Equation [30] satisfying the boundary Condition [29] is known; it is the so-called modified Bessel function of the second kind and zero order, and is represented by the symbol $K_0(\lambda vr)$ (10).

It can be shown that this function tends to $\ln r$ as r tends to zero, hence $\frac{dK_0}{dr} \times r$ tends to a constant value. Thus $K_0(\lambda vr)$ fulfills the Condition [29]. On the other hand, this function tends to $\sqrt{\frac{\pi}{2\lambda vr}}\, e^{-\lambda vr}$ as r tends to infinity, whereby the boundary Condition [28] also is satisfied. The solution of the two-dimensional case therefore is

$$T - T_0 = \frac{q'}{2\pi k} e^{-\lambda v\xi} K_0(\lambda vr) \ldots\ldots\ldots\ldots [31]$$

This solution will be discussed in Part 2 in connection with the welding of thin plates.

Thin Plates; Surface Losses. If the thickness g of the plate is small enough to neglect the temperature gradient in the z-direction, Equation [31] can be made to include heat losses through the surface. Calling H and H' the dissipation ratios at top and bottom face of the plate, respectively, and noting that PH/A appearing in Equation [22] becomes simply $\frac{H + H'}{g}$, there follows

$$\frac{d^2\varphi}{dr^2} + \frac{1}{r}\frac{d\varphi}{dr} - \left[(\lambda v)^2 + \frac{H + H'}{g}\right]\varphi = 0 \ldots\ldots [32]$$

hence the solution

$$T - T_0 = \frac{q}{2\pi kg} e^{-\lambda v\xi} K_0\left[\sqrt{\lambda v^2 + \frac{H + H'}{g}}\, r\right] \ldots\ldots [33]$$

where q/g, which appears instead of q', is the rate of heat per unit thickness, and q is the total rate of heat of the linear source.

Linear Source of Variable Strength; No Surface Losses. In previous discussion it has been assumed that the rate of heat q' per unit thickness was a constant. If this rate varies along the thickness, an appropriate solution similar to Equation [31] can be obtained by means of cosine series. To this end develop function $q'(z)$ in Fourier series, between $z = 0$ and $z = g$. Suppose this function reads as follows

$$q'(z) = q'(0)\sum_0^n A_n \cos\frac{\pi nz}{g} \ldots\ldots\ldots\ldots [34]$$

Since $q'(z)$ is a cosine function of z, the same must be true for φ by virtue of Condition [29]. Thus the differential Equation [30] must be completed by adding the second derivative with respect to z, to read as follows

$$\frac{\partial^2\varphi}{\partial z^2} + \frac{\partial^2\varphi}{\partial r^2} + \frac{1}{r}\frac{\partial\varphi}{\partial r} - (\lambda v)^2\varphi = 0 \ldots\ldots\ldots [35]$$

It is easy to see that this equation is satisfied by a function of the form

$$\cos\frac{\pi nz}{g} K_0\left(\sqrt{\lambda v^2 + \left(\frac{\pi n}{g}\right)^2}\, r\right) \ldots\ldots\ldots [36]$$

Hence with reference to Equation [34]

$$T - T_0 = \frac{q_0'}{2\pi k} e^{-\lambda v\xi} \sum_0^n A_n \cos\frac{\pi nz}{g} K_0\left(\sqrt{\lambda v^2 + \left(\frac{\pi n}{g}\right)^2}\, r\right) \ldots\ldots [37]$$

Solution [37] has been applied to the problem of flame-cutting, and the reader is referred to a previous publication for further developments (11).

Three-Dimensional Flow. In this case the boundary conditions are

$$\left.\begin{array}{ll} \partial T/\partial\xi \to 0 & \text{for } \xi \to \pm\infty \\ \partial T/\partial y \to 0 & \text{for } y \to \pm\infty \\ \partial T/\partial z \to 0 & \text{for } z \to \pm\infty \end{array}\right\} \ldots\ldots\ldots [38]$$

and by considering a spherical surface $4\pi R^2$ drawn around the heat source with radius $R = \sqrt{\xi^2 + y^2 + z^2}$

$$-\frac{\partial T}{\partial R} 4\pi R^2 k \to q \quad \text{for } R \to 0 \ldots\ldots\ldots [39]$$

where q is the rate of heat of a point source.

Because of Conditions [38] and [39], and the symmetrical form of the differential Equation [5], φ depends only upon the distance R, hence using polar co-ordinates Equation [5] can be rewritten as follows[5]

$$\frac{d^2\varphi}{dR^2} + \frac{2}{R}\frac{d\varphi}{dR} - (\lambda v)^2\varphi = 0 \ldots\ldots\ldots\ldots [40]$$

It is easy to see that

$$\frac{d^2\varphi}{dR^2} + \frac{2}{R}\frac{d\varphi}{dR} = \frac{1}{R}\frac{d^2(R\varphi)}{dR^2}$$

hence Equation [40] becomes

$$\frac{d^2(R\varphi)}{dR^2} - (\lambda v)^2 R\varphi = 0 \ldots\ldots\ldots\ldots [41]$$

[5] Refer to (9), p. 12.

A solution of Equation [41] satisfying the boundary Conditions [38] obviously is

$$R\varphi = Ce^{-\lambda vR} \qquad [42]$$

or

$$\varphi = \frac{Ce^{-\lambda vR}}{R}$$

This solution also satisfies Conditions [39] since $\frac{d\varphi}{dR} \times R^2$ tends to a constant value as $R \rightarrow 0$. Hence the corresponding temperature distribution

$$T - T_0 = \frac{q}{4\pi k} e^{-\lambda v\xi} \frac{e^{-\lambda vR}}{R} \qquad [43]$$

This solution will be discussed in detail in connection with welding in Part 2.

Solid Bounded by Planes

The Method of Images. If the quasi-stationary state is to be preserved, the solid cannot be bounded by planes perpendicular to the direction of motion. However, the state can still remain quasi-stationary if the bounded planes are parallel to the direction of motion. This circumstance leaves out the linear flow, but retains the two- and three-dimensional flows. Assuming there is no heat loss through the bounding planes, the method of images can be used conveniently to obtain the required solutions.[6] It will be recalled that in this method fictitious sources are associated with the real source of heat in such a way that the bounding planes become planes of symmetry. Such an arrangement fulfills the condition of no radiation, and it is accomplished by locating the fictitious sources at the mirror reflections of the real source, the bounding planes acting as the corresponding plane mirrors.

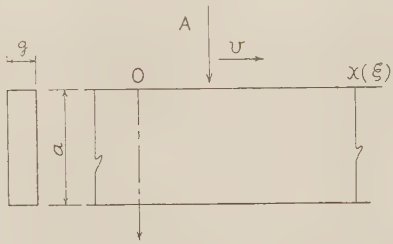

Fig. 1 Solid Bounded by Planes Paralleled to Direction of Motion

Two-Dimensional Flow. With reference to Fig. 1, the problem consists of determining the temperature distribution in a plate of width a and thickness g, heated by a linear source traveling along one of its edges. In accordance with the method of images just explained, imagine the plate extended to the size of a semi-infinite plate. Then the temperature due to the source q distributed over the thickness g would read, with reference to Equation [31]

$$T - T_0 = \frac{q}{\pi kg} e^{-\lambda v\xi} K_0(\lambda vr)$$

To this value now add the contributions of the mirror reflections of the source with respect to mirrors placed at $y = 0$ and $y = a$. These contributions are of the form

$$\frac{q}{\pi kg} e^{-\lambda v\xi} K_0(\lambda vr_n) \qquad [44]$$

[6] Reference (9) p. 158.

where

$$r_n = \sqrt{\xi^2 + (y \pm 2na)^2}$$

Hence the solution becomes

$$T - T_0 = \frac{q}{\pi kg} e^{-\lambda v\xi} \sum_{n=-\infty}^{n=+\infty} K_0(\lambda vr_n) \qquad [45]$$

From the way this solution has been built it is obvious that

$$\frac{\partial T}{\partial y} = 0 \qquad [46]$$

for $y = a$ and $y = 0$, which fulfills the boundary conditions along the two edges. Condition [29] also is fulfilled, for of all the terms $K_0(\lambda vr_n)$ which compose the solution, Equation [45], only the term $K_0(\lambda vr)$ contributes to the value of $q' = q/g$, the derivatives of the other vanishing as r when being multiplied by $2\pi r$. It remains to prove that

$$\frac{\partial T}{\partial \xi} \rightarrow 0 \quad \text{as} \quad \xi \rightarrow \pm\infty \qquad [47]$$

This can be done best by transforming Equation [45] into a Fourier series. By proceeding as explained in Appendix 2, the following expression is obtained

For $\xi > 0$

$$T - T_0 = \frac{q}{c\rho agv}\left[e^{-2\lambda v\xi} + \sum_1^\infty \frac{2}{\mu_n} e^{-(\mu_n+1)\lambda v\xi} \cos\frac{\pi ny}{a}\right] \qquad [48]$$

and for $\xi < 0$

$$T - T_0 = \frac{q}{c\rho agv}\left[1 + \sum_1^\infty \frac{2}{\mu_n} e^{(\mu_n-1)\lambda v\xi} \cos\frac{\pi ny}{a}\right] \qquad [49]$$

where

$$\mu_n = \sqrt{1 + \left(\frac{\pi n}{\lambda va}\right)^2} \qquad [50]$$

and $c\rho$ is the volume specific heat.

By differentiating Equations [48] and [49] with respect to ξ, it is seen that the derived expression tends to zero for ξ tending to both + and — infinity as required by Conditions [47].

The characteristic feature of solution, Equation [49], is that it tends to a constant temperature, which is different from the initial temperature T_0, after the source has moved away from the section considered. Calling T_2 the new temperature, there follows from Equation [49] that, if $\xi \rightarrow -\infty$

$$T_2 - T_0 = \frac{q}{c\rho agv} \qquad [51]$$

The meaning of Equation [51] is very simple. It represents a uniform rise of temperature produced by the heat q in a volume of plate covered in unit time and can be explained as follows: Because of the quasi-stationary state the temperature situation around the heat source remains unchanged, and since there are no surface losses the amount of heat delivered in unit time must be employed to raise the temperature of an additional volume of plate = agv far behind the source.

Putting Equation [51] in Equations [48] and [49], there follows

For $\xi > 0$

$$T - T_0 = (T_2 - T_0)\left[e^{-2\lambda v\xi} + \sum_1^\infty \frac{2}{\mu_n} e^{-(\mu_n+1)\lambda v\xi} \cos\frac{\pi ny}{a}\right] \qquad [52]$$

and for $\xi < 0$

$$T - T_0 = (T_2 - T_0)\left[1 + \sum_1^\infty \frac{2}{\mu_n} e^{(\mu_n - 1)\lambda v \xi} \cos\frac{\pi n y}{a}\right]. \quad [53]$$

By comparing Equations [52] and [53] with Equations [17] and [18], it is seen that the two-dimensional flow in a solid of a limited cross section tends to become linear at some distance from the source, and from Equation [50] it is apparent that this distance is so much shorter as the value of $\lambda v a$ is smaller; for example, as the plate is narrower. This trend is represented in Fig. 2 by

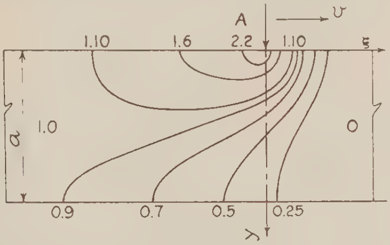

Fig. 2 Temperature Distribution in Thin and Narrow Plate Due to Heat Source Moving Along One of Its Edges
(Initial temperature taken as zero and final temperature as 1.)

means of isotherms, which for convenience have been marked in terms of the relative temperature $U = (T - T_0)/(T_2 - T_0)$. The value of $\lambda v a$ has been taken arbitrarily as 2. It is seen that in the upper half of the plate the relative temperature rises from a value $= 0$ in front of the source to a maximum which is greater than 1 in the vicinity of the source and then drops progressively down to 1 behind the source. The maximum temperature is so much more in excess of 1 as the layer under consideration is closer to the edge passed by the heat source. On the other hand, layers of metals which are close to the opposite edge never exceed 1. They approach this value asymptotically as the flow becomes more linear. This last situation is virtually reached in Fig. 2 at a distance which is smaller than twice the width of the plate.

For many practical applications, especially flame-hardening and quenching, solutions, Equations [52] and [53], have the disadvantage of becoming infinite at the location of the source. This, of course, is a direct consequence of the assumption that the heat source is reduced to a line. A more practical solution may be obtained by removing this restriction as will be shown later.

Two-Dimensional Flow; Plane Source. The inconvenience of having an infinite value of temperature at the location of the source disappears if the latter instead of being a single line is spread over a certain area, i.e., if it becomes a plane source. To demonstrate this proposition, suppose a source with a rate of heat $= q''$ per unit area covers the edge of the plate, Fig. 1, from point $\xi' = 0$ to point $\xi' = l$. This source can be represented as being composed of an infinite number of infinitely small linear sources $q''d\xi$ set side by side from $\xi' = 0$ to $\xi' = l$. If q'' is constant, then $q'' = \dfrac{q}{gl}$ where q is the rate of heat of the plane source, hence $q''d\xi' = \dfrac{qd\xi'}{lg}$. On the other hand, according to Equations [48] and [49], a linear source of strength $\dfrac{qd\xi'}{lg}$ placed at a distance ξ' from the origin contributes to the temperature of a point ξ, y in amount of

$$\frac{qd\xi'}{c\rho agvl}\left[e^{-2\lambda v(\xi - \xi')} + \sum \frac{2}{\mu_n} e^{-(\mu_n + 1)\lambda v(\xi - \xi')} \cos\frac{\pi n y}{a}\right] \quad [54]$$

for $\xi > \xi'$, and in amount of

$$\frac{qd\xi'}{c\rho agvl}\left[1 + \sum \frac{2}{\mu_n} e^{(\mu_n - 1)\lambda v(\xi - \xi')} \cos\frac{\pi n y}{a}\right] \quad [55]$$

for $\xi < \xi'$. Three cases may be distinguished, according to whether the point ξ, y is located outside or inside the portion of plate covered by the plane source.

Case 1. If $\xi > l$, Equation [54] applies to all elementary sources from $\xi' = 0$ to $\xi' = l$, and by integrating Equation [54] from $\xi' = 0$ to $\xi' = l$, there follows

$$T - T_0 = \frac{q}{c\rho agv}\left[\frac{(e^{-2\lambda vl} - 1)}{\lambda vl} e^{-2\lambda v\xi} + \sum \frac{2}{\mu_n(\mu_n + 1)} \frac{(e^{\lambda vl(\mu_n + 1)} - 1)}{\lambda vl} \times e^{-(\mu_n + 1)\lambda v\xi} \cos\frac{\pi n y}{a}\right] \quad [56]$$

Case 2. If $\xi < 0$, it is Equation [55] which applies to all elementary sources; hence by integrating Equation [55] from $\xi' = 0$ to $\xi' = l$, there follows

$$T - T_0 = \frac{q}{c\rho agv}\left[1 + \sum \frac{2}{\mu_n(\mu_n - 1)} \times \frac{1 - e^{-\lambda vl(\mu_n - 1)}}{\lambda vl} \times e^{(\mu_n - 1)\lambda v\xi} \cos\frac{\pi n y}{a}\right] \quad [57]$$

Case 3. Finally, if $l > \xi > 0$, Equation [54] applies to all elementary sources for which $0 < \xi' < \xi$, and Equation [55] applies to all sources for which $\xi < \xi' < l$. By integrating each of these expressions within the limits of its applicability, we have

$$\left.\begin{aligned} T - T_0 = \frac{q}{c\rho agv}\Bigg[&\frac{1 - e^{-2\lambda v\xi}}{\lambda vl} + \left(1 - \frac{\xi}{l}\right) \\ &+ \sum \frac{2}{\mu_n(\mu_n + 1)} \frac{1 - e^{-(\mu_n + 1)\lambda v\xi}}{\lambda vl} \cos\frac{\pi n y}{a} \\ &+ \sum \frac{2}{\mu_n(\mu_n - 1)} \frac{1 - e^{-(\mu_n - 1)\lambda v(l - \xi)}}{\lambda vl} \cos\frac{\pi n y}{a}\Bigg] \end{aligned}\right\} \quad [58]$$

It is easy to see that the maximum value of temperature occurs at $\xi = 0$, $y = 0$. For this point

$$T_1 - T_0 = \frac{q}{c\rho agv}\left[1 + \sum \frac{2}{\mu_n(\mu_n - 1)} \frac{1 - e^{-(\mu_n - 1)\lambda vl}}{\lambda vl}\right] \quad [59]$$

On the other hand, it follows from Equations [57] and [51] that for $\xi \to -\infty$, the temperature tends to the same constant value T_2 as in the case of a linear source. Therefore Equation [59] can be rewritten to read

$$T_1 - T_0 = (T_2 - T_0)\left[1 + \sum \frac{2}{\mu_n(\mu_n - 1)} \frac{1 - e^{-(\mu_n - 1)\lambda vl}}{\lambda vl}\right] \quad [60]$$

If the heat instead of being delivered is absorbed by the source, i.e., if the latter is replaced by a sink, then $T_0 > T_2 > T_1$, and Equation [60] becomes

$$T_0 - T_1 = (T_0 - T_2)\left[1 + \sum \frac{2}{\mu_n(\mu_n - 1)} \frac{1 - e^{-(\mu_n - 1)\lambda vl}}{\lambda vl}\right] \quad [61]$$

Both expressions, Equation [60] as well as [61], can find suitable applications in metal treatments, the former in flame-hardening, the latter in continuous quenching, as will be shown later in Part 3.

Remark. By comparing Equation [61] with Equation [53] it is seen that the former is represented by a series of the form

$1/n(n-1)$, while the latter is of the form $1/n$. But the series $1/n(n-1)$ is known to be absolutely convergent, whereas the series $1/n$ is divergent, if all terms are positives, and this is what happens at the location of the source, for which $\xi = 0$ and $y = 0$.

Three-Dimensional Flow. Solid bounded by planes $z = 0$ and $z = g$; no surface losses.

Following the pattern developed in the preceding section, imagine first the solid extended in the z-direction from $z = 0$ to $z = \infty$, covering one half of the space. Then the temperature due to the source of strength q located at $\xi = 0$, $y = 0$, $z = 0$ would read with reference to Equation [43]

$$T - T_0 = \frac{q}{2\pi k} e^{-\lambda v \xi} \frac{e^{-\lambda v R}}{R} \qquad [43a]$$

The factor 2 in Equation [43a], as compared to 4 in Equation [43] accounts for one half of the space.

To Equation [43a] add now the contributions of the mirror reflections of the source with respect to planes $z = 0$ and $z = g$. These contributions are of the form

$$\frac{q}{2\pi k} e^{-\lambda v \xi} \frac{e^{-\lambda v R_n}}{R_n} \qquad [62]$$

where

$$R_n = \sqrt{\xi^2 + y^2 + (z \pm 2ng)^2}$$

Hence the solution reads

$$T - T_0 = \frac{q}{2\pi k} e^{-\lambda v \xi} \sum_{-\infty}^{\infty} \frac{e^{-\lambda v R_n}}{R_n} \qquad [63]$$

From the way this solution has been built it is obvious that

$$\frac{\partial T}{\partial z} = 0 \qquad [64]$$

for $z = 0$ and $z = g$, which fulfills the condition of no radiation. Condition [39] also is fulfilled, since the derivative of all terms of Equation [63] tends to zero as R, when being multiplied by R^2, except the derivative of the term $n = 0$, which becomes a constant when being multipled by R^2.

Finally it is easy to see that

$$\left.\begin{array}{ll} \partial T/\partial \xi \rightarrow 0 & \text{as } \xi \rightarrow \pm\infty \\ \partial T/\partial y \rightarrow 0 & \text{as } y \rightarrow \pm\infty \end{array}\right\} \qquad [65]$$

which fulfills the first two Conditions [38].

By a procedure similar to that explained in Appendix 2, solution of Equation [63] can be transformed into a Fourier series to read

$$T - T_0 = \frac{q}{2\pi k g} e^{-\lambda v \xi} \left\{ K_0(\lambda v r) + 2 \sum_{1}^{\infty} K_0 \left[r \sqrt{\lambda v^2 + \left(\frac{\pi n}{g}\right)^2} \right] \cos \frac{\pi n z}{g} \right\} \qquad [66]$$

The similarity between Equations [66] and [31] is obvious. In fact, both solutions become practically identical for large values of r. This means that the solid behaves more and more like a thin plate as the distance from the heat source increases. Conversely, for values of r which are small as compared to the thickness g, Equation [63] differs little from Equation [43] which has been derived for an infinitely thick plate. This twofold aspect of solution [63] or [66] is well evidenced in the behavior of butt-welded plates, as will be demonstrated in Part 2.

Solid Bounded by a Cylinder

Because of the condition of the quasi-stationary state, the axis of the cylinder must be parallel to the direction of motion, and in most cases of practical interest it will coincide with the path of the moving source. Even so, it does not appear possible to obtain a simple solution of this problem. Recently, R. H. Cameron and the author have worked out a solution which, although far from being simple, is believed to be practically usable. However, because of the somewhat involved mathematical treatment of the solution, the presentation of their work is reserved for a separate publication.

2 APPLICATION TO ARC WELDING

Rate of Fusion of Electrode

In the process of welding, the heat generated by the arc causes the electrode to melt, thus providing the metal necessary for joining the parts to be welded. Assuming the electrode is long enough with respect to its diameter and neglecting the heat generated by the Joule effect, the process of melting for the first few inches of the electrode can be looked upon as being of a quasi-stationary nature, in so far as the temperature distribution around the welding arc is concerned. The rate of melting of the electrode is then the speed with which the arc moves along the electrode, and the temperature T at a point located at a distance ξ from the arc is equal, according to Equation [18], Part 1

$$T = T_0 + (T_1 - T_0)e^{-2\lambda v \xi} \qquad [67]$$

Here, T_0 is the initial temperature, T_1 is the temperature at the location of the arc, i.e., the temperature of fusion, 2λ is the converse of thermal diffusivity, and v is the rate of fusion.

Equation [67] assumes that there are no surface losses, in other words, that the electrode is perfectly insulated. This assumption is an approximation but it offsets to a great extent the error introduced by neglecting the Joule effect in the electrode. That this is so is proved by the fact that the relation between the rate of fusion v and current density i, derived from the foregoing assumptions, is in good agreement with the experiment. This relation is obtained as follows:

It has been shown in Part 1 that in the absence of surface losses the rate of heat q'' absorbed in the electrode per unit area is

$$q'' = (T_1 - T_0)c\rho v \qquad [68]$$

$c\rho$ is the specific volume heat content (c specific heat, ρ density), and T_1, T_0, and v, as previously, the temperature of fusion, the initial temperature, and the rate of fusion, respectively. On the other hand, q'' is a fraction of the heat generated by the arc, which is proportional to the product VI, where V is the voltage drop across the arc, and I is the current intensity. Experiment shows that for a given type of electrode the voltage drop V is fairly constant within a large range of current densities, and that for various types of coated-steel electrodes it varies between 25 and 35 volts. Thus for a given type of electrode of a diameter d, it is possible to write

$$q'' \frac{\pi d^2}{4} = fI \qquad [69]$$

where f is a constant (13).

Substituting in Equation [68], there follows

$$\frac{fI}{\pi d^2/4} = (T_1 - T_0)c\rho v$$

hence

$$v = f_0 i \qquad [70]$$

where

$$f_0 = \frac{f}{(T_1 - T_0)c\rho}$$

Relation, Equation [70] has been checked experimentally and has been found to hold within a reasonable degree of accuracy for moderate current densities and for the first 3 to 4 in. of the electrode. The values of f_0 reported in the literature for a mild-steel electrode lie between 1.24×10^{-3} and 1.56×10^{-3} cu in. per amp per min.

Substituting Equation [70] in [67] there follows

$$T = T_0 + (T_1 - T_0)e^{-2\lambda f_0 i \xi}$$

or putting

$$i = I \Big/ \frac{\pi d^2}{4}$$

$$T = T_0 + (T_i - T_0)e^{-f' I \xi} \qquad [71]$$

where

$$f' = \frac{2\lambda f_0}{\pi d^2/4} = \frac{8\lambda f_0}{\pi d^2} \qquad [72]$$

In Fig. 3 the ratios of $(T - T_0)/(T_1 - T_0)$ have been plotted against the distance ξ from the location of the arc in inches, for four different values of current density, namely, 145, 165, 185, and 240 amp. These values may be considered as being within workable range of current intensities for a $^5/_{32}$-in. mild-steel electrode. The following values were used to calculate the factor f' appearing in Equation [71]

$$2\lambda = 1.00 \text{ sq in. per min}$$

$$f_0 = 1.25 \times 10^{-3} \text{ cu in. per amp per min}$$

$$d = {}^5/_{32} \text{ in.}$$

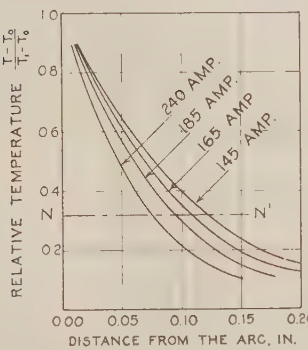

Fig. 3 Temperature Distribution in Mild-Steel Electrode for Various Current Intensities

It appears from Fig. 3 that the higher the current intensity the shorter is the range over which the electrode is affected by the heat of the arc. This condition is well illustrated in Fig. 4, which shows macroetched sections made through the axis of a $^5/_{32}$-in. electrode submitted to the current intensities mentioned. Because of the recrystallization process the part of the electrode which has been heated beyond some 950 F responds more readily to the etching (10 per cent solution of nitric acid) than the rest of the electrode. It is seen that in accordance with diagrams, Fig. 3, the extent of the recrystallized zone is greater for the low current intensity than it is for high current intensity, see also line NN', Fig. 3, corresponding to the temperature of recrystallization of mild steel ($T_1 - T_0/T_1 - T_0 \approx 0.32$).

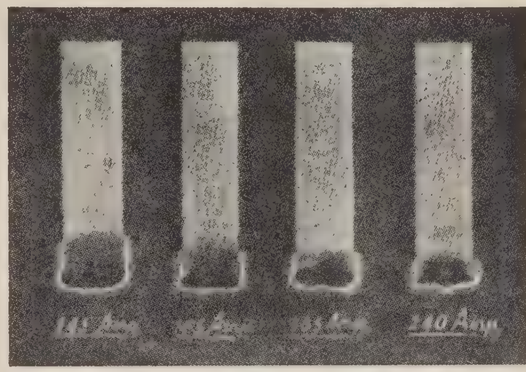

Fig. 4 Heat-Affected Areas in Mild-Steel Electrode for Various Current Intensities
(Courtesy *The Welding Journal.*)

Arc-Welding of Thin Plates

Temperature Distribution in an Edge-Welded Plate. The validity of Equation [31], Part 1, in the case of arc welding was tested experimentally by Schmerber and the author (12). They deposited a weld layer at a constant rate along one of the edges of a flat bar, 5.5 in. wide, and measured the variation of temperature in the bar produced by this operation. Assuming no losses of heat through the surface, the temperature can be taken as being uniform through the thickness. Hence the heat flow is of a two-dimensional nature, and Equation [31], Part 1, is applicable. For the purpose of computation, a constant value of heat conductivity k was adopted = 0.1 cal per deg C per cm per sec, but the diffusivity coefficient $1/2\lambda$ was made to vary according to a linear law with the temperature. Under these circumstances, the agreement between the measurement and computation was quite satisfactory, except at the immediate neighborhood of the arc, Fig. 5. There, the computed isotherms were closer to the origin than the measured values. This discrepancy could be explained by the fact that the theory considered the source of heat as a line, whereas in practice the source of heat had a finite size.

Rate of Cooling in Thin Butt-Welded Plates. On the basis of the foregoing experiment it can be expected that Equation [31] will be applicable also to the case of thin butt-welded plates. Hess and his co-workers (14) determined experimentally the rates of cooling produced very near to the weld in plates of various thicknesses. If the temperature is assumed constant through the thickness, these rates of cooling can also be derived from Equation [31] by differentiating the latter with respect to time. In so doing, attention is called to the fact that in the present application the rates of cooling are measured at some distance from the location of the heat source. Hence, according to the theory of Bessel functions (10), Equation [31] can be reduced to a more simple expression to read

$$T - T_0 = \frac{q}{2\pi k g}\sqrt{\frac{\pi}{2\lambda v r}}\, e^{-\lambda v \xi - \lambda v r}$$

$$= \frac{q}{2\pi k g}\,\frac{e^{-\lambda v(\xi + r)}}{\sqrt{vr}}\sqrt{\frac{\pi}{2\lambda}} \qquad [73]$$

where T is the temperature measured at a distance $r = \sqrt{\xi^2 + y^2}$ from the source, T_0 the initial temperature, q the rate of heat input, v the speed of the source, and k, $c\rho$, and $1/2\lambda = k/c\rho$ as

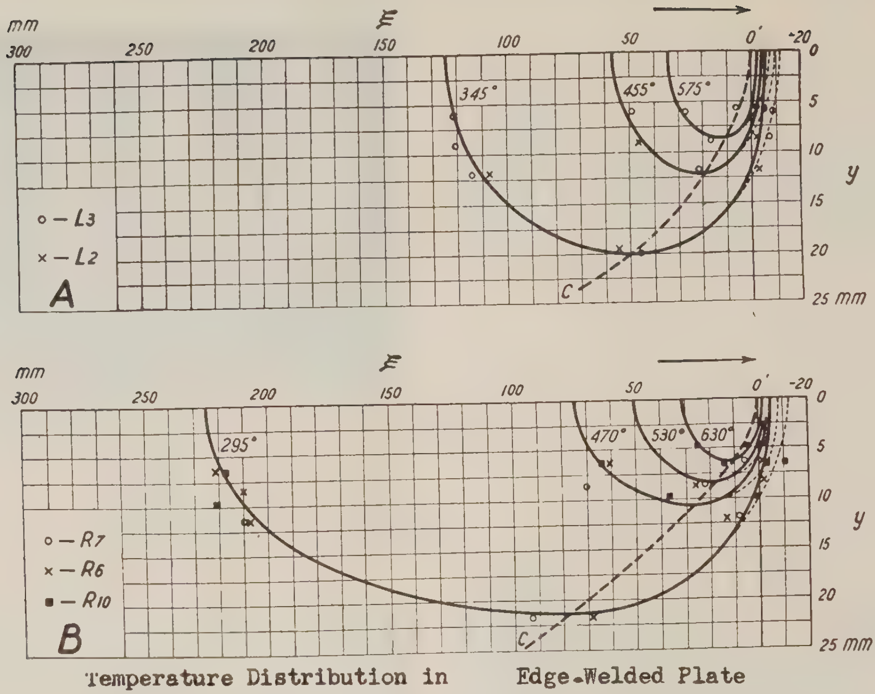

Fig. 5 Temperature Distribution in Edge-Welded Plate

(Continuous lines, calculated; broken lines, experimental. A, Current, 93 amp; welding speed, 0.21 cm per sec or 4.95 ipm. B, Current, 141 amp; welding speed, 0.43 cm per sec or 10.1 ipm. L2, L3, R7, R6, R10 are different specimens. Courtesy *The Welding Journal*.)

previously, the heat conductivity, volume specific heat, and heat diffusivity, respectively.

Along the weld, $y = 0$, hence $r = /\xi/$; also $\xi = -vt$, hence Equation [73] can be further simplified as follows

$$T - T_0 = \frac{1}{2\sqrt{\pi k c\rho}} \times \frac{q}{gv} \times \frac{1}{\sqrt{t}} \quad [74]$$

Expression q/gv represents, as it is easy to see, the amount of heat per unit section of plate, measured in the direction of welding. In practice, the welding conditions are defined by another value more readily determined, namely, the total amount of energy J/gv per unit section of plate supplied by the arc, expressed in joules per sq in. This value is related to the former by a coefficient which defines the heat efficiency of the arc, and which Hess and co-workers represented by the symbol (I.F.) input factor. Thus

$$T - T_0 = \frac{(\text{I.F.})}{2\sqrt{kc\rho\pi}} \times \frac{J}{gv} \times \frac{1}{\sqrt{t}} \quad [75]$$

or

$$(T - T_0)\sqrt{t} = C \text{ (independent of time)} \quad [76]$$

By differentiating with respect to time there follows

$$\frac{dT}{dt}\sqrt{t} + \frac{1}{2}\,\frac{T - T_0}{\sqrt{t}} = 0$$

or

$$\frac{dT}{dt} = -\frac{1}{2}\,\frac{T - T_0}{t} \quad [77]$$

Equation [77], due to Hess and co-workers (14), states that the rate of cooling at a given temperature T is proportional not only to this temperature but also to the converse of time t that it has taken for the point considered to cool down to the temperature T after the welding arc has passed through this point. In this respect the rate of cooling in welding is quite different from the rate of cooling in quenching, for which no dependence on time is assumed (15). As will appear later, the factor 1/2 at the right-hand side of Equation [77] is characteristic for the two-dimensional problem of welding.

If time t is eliminated from Equations [77] and [75], the following expression is obtained

$$\frac{dT}{dt} = -\frac{2\pi k c\rho}{(\text{I.F.})^2} \times \left(\frac{gv}{J}\right)^2 \times (T - T_0)^3 \quad [78]$$

or

$$\left(\frac{dT}{dt}\right)^{1/2} \times \frac{J}{vg} = \frac{\sqrt{2\pi k c\rho}}{(\text{I.F.})} \times (T - T_0)^{3/2} \quad [79]$$

Since the right-hand side of Equation [79] is a constant for a given type of steel, electrode, and temperature T, the product on the left-hand side also must be constant, no matter what are the individual values of rates of cooling, energy inputs, and thickness. This condition has been tested for the case of $^1/_4$-in. and $^1/_2$-in. butt-welded plates using experimental results of Hess and co-workers (14), obtained very near to the weld.[7] When the products figuring on the left-hand side of Equation [79] are plotted as a function of rates of cooling at 1300 F, the diagram, Fig. 6, is obtained. From this diagram it appears that the $^1/_4$-in. butt-welded plate behaves like a thin plate for all rates of cooling at 1300 F, while the $^1/_2$-in. plate behaves like a thin plate only for rates of

[7] It can be shown that the rate of cooling very near to the weld is not appreciably different from that in the weld itself.

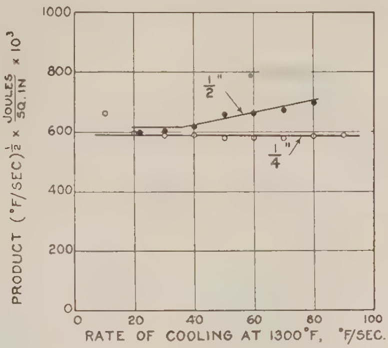

FIG. 6 TWO-DIMENSIONAL HEAT FLOW IN $1/4$- AND $1/2$-IN. BUTT-WELDED PLATES AS EVIDENCED BY CONSTANT VALUE OF PRODUCT INDICATED

cooling at 1300 F which are smaller than 50 deg F per sec. The reason for this discrepancy will be discussed later.

ARC WELDING OF THICK PLATES

Rate of Cooling in Butt-Welded Plates. Following the ideas developed in the section on the arc welding of thin plates, it may be anticipated that since Equation [31] has approximated fairly well the conditions of heat flow in butt-welded plates, Equation [43] may approximate likewise the conditions of heat flow in thick butt-welded plates. If the observation is limited to the surface of the weld, the distance R, figuring in Equation [43], is simply the distance $/\xi/$ covered by the arc in a given period of time t; thus

$$T - T_0 = \frac{q}{2\pi k} \times \frac{1}{\xi} \qquad [80]$$

Equation [80] does not contain explicitly the speed v, and it does not depend upon the diffusivity of the material. In fact the same expression is obtained, if the point source instead of being in motion remains stationary, except that in this case ξ represents not only the distance behind the source, but the distance from the source measured in any direction.[8] On the other hand, if ξ is expressed in terms of the distance covered by the arc in time t, then $\xi = vt$, and Equation [80] becomes

$$T - T_0 = \frac{1}{2\pi k} \times \frac{q}{v} \times \frac{1}{t} \qquad [81]$$

The quantity (q/v) represents the heat input per unit length and is to be compared with a similar quantity (q/vg) used in the study of thin plates. As in the latter case, we shall write

$$q/v = (\text{I.F.})J/v \qquad [82]$$

where J/v represents the total energy input of the arc per unit length and (I.F.) the heat efficiency of the arc. With reference to Equation [82], Equation [81] may be rewritten as follows

$$(T - T_0) \times t = C \text{ (independent of time)} \qquad [83]$$

Hence by differentiating

$$\frac{dT}{dt} = -\frac{T - T_0}{t} \qquad [84]$$

This expression, also due to Hess and co-workers (14), differs from Equation [77], derived previously for the rate of cooling in a thin plate, only by the absence of the factor 1/2. Thus if conditions of welding are such that the same temperature T is reached at the same time t in both the thin and thick plates, the rate of cooling still will be twice as great in the thick plate as in the thin one.

If time t is eliminated from Equations [84] and [81], then with reference to Equation [82] the following formula is obtained

$$\frac{dT}{dt} = -\frac{2\pi k}{(\text{I.F.})} \times \frac{v}{J} \times (T - T_0)^2 \qquad [85]$$

or

$$\frac{dT}{dt} \times \frac{J}{v} = -\frac{2\pi k}{(\text{I.F.})} \times (T - T_0)^2 \qquad [86]$$

The right-hand side of Equation [86] is a constant for a given temperature and a given material, hence the product on the left-hand side also must be constant for any particular value of rate of cooling, energy input, and speed.

Although, strictly speaking, this condition holds only for the points of weld deposit for which $R = \xi$, points of the base metal very close to the weld may be expected to satisfy the condition mentioned with a reasonable degree of accuracy. With this in mind, experimental results of Hess and co-workers (14) on 1 and $1^1/_2$-in. steel plates have been used to compute the products figuring on the left-hand side of Equation [86] for various rates of cooling at 1300 F. When plotted as a function of these rates, they gave a fairly constant value, Fig. 7. The values for the $1^1/_2$-in. plate are in general more consistent than those for the 1-in. plate. Thus the $1^1/_2$-in. plate and to some extent also the 1-in. plate behave under the particular condition of cooling like plates of infinite thickness.

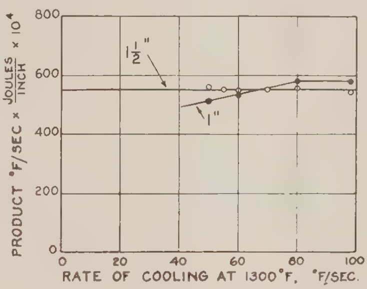

FIG. 7 THREE-DIMENSIONAL HEAT FLOW IN 1- AND $1^1/_2$-IN. BUTT-WELDED PLATES AS EVIDENCED BY CONSTANT VALUE OF PRODUCT INDICATED

Determination of Heat Diffusivity From Rates of Cooling in Thin and Thick Butt-Welded Plates. If rates of cooling have been determined for a particular temperature T and known conditions of welding in a thin and thick butt-welded plate, then assuming the same heat efficiency (I.F.) in both cases, there follows when Equation [75] is divided by Equation [85]

$$\frac{\left[\left(\frac{dT}{dt}\right)^{1/2}\left(\frac{J}{vg}\right)\right]}{\left[\frac{dT}{dt} \times \frac{J}{v}\right]} = \sqrt{\frac{c\rho}{2k}} \times \frac{1}{\sqrt{\pi}} (T - T_0)^{-1/2} \qquad [87]$$

but

$$\frac{c\rho}{2k} = \lambda$$

hence the diffusivity

$$\frac{1}{2\lambda} = \frac{1}{2\pi(T - T_0)} \times \frac{\left[\frac{dT}{dt} \times \frac{J}{v}\right]^2 \text{ thick plate}}{\left[\left(\frac{dT}{dt}\right)^{1/2} \times \frac{J}{vg}\right]^2 \text{ thin plate}} \qquad [88]$$

[8] Reference (9) p. 151.

Substituting in Equation [88] the values taken from Figs. 6 and 7, for the $^1/_4$-in. and $1^1/_2$-in. plates, respectively, the value of the diffusivity becomes

$$\frac{1}{2\lambda} = \frac{1}{2\pi(1300 - 68)} \times \frac{(550 \times 10^4)^2}{(590 \times 10^3)^2} = 0.0112 \frac{\text{in.}^2}{\text{sec}}$$

$$= 0.072 \frac{\text{cm}^2}{\text{sec}}$$

as against the value of 0.0685, computed for pure iron from the data of the International Tables for 1300 F (16).

Considering the uncertainty attached to the determination of the heat conductivity k, and the probable error of measurements, this agreement appears quite satisfactory. However, it must be borne in mind that the assumption on which Equation [88] is based precludes any variation of the diffusivity with temperature, hence the value of 0.072 sq cm per sec must be regarded as an average rather than as a particular value of the diffusivity.

Temperature Distribution in Thick (and Thin) Arc-Welded Plates. On the basis of the foregoing results it can be assumed that the application of Equation [43] will yield a satisfactory picture of the temperature distribution during welding in thick butt-welded plates. More specifically, the conception will apply to a situation created by a top weld layer when the plates have been allowed to cool down to the room temperature. This situation is represented in Fig. 8 for various conditions of welding by means of a family of isotherms drawn around the instantaneous position of the electrode (11). Because of the quasi-stationary state of heat flow, the same figure may be used to depict the variation of temperature at a given section of plate and at various instances of welding.

For the purpose of computation, the following values were substituted in Equation [43], using C.G.S. units

q = 0.239 (I.F.) $\times V \times I$ where 0.239 is the thermal equivalent of watts
(I.F.) = heat efficiency of arc assumed = 0.65 (12)
V = voltage drop in arc assumed = 25 volts (12)
I = current intensity in amp, variable
k = heat conductivity—0.1 cal per cm per deg C per sec, for steel and 0.485 cal per cm per deg C per sec for aluminum
λ = one half of the converse value of diffusivity, assumed = 5 sec per sq cm for steel and 1.2 sec per sq cm for aluminum

The main characteristics of the temperature distribution obtained under these conditions, Fig. 8, are as follows:

1 The rise of temperature in front of the heat source is much steeper than the fall of temperature behind the heat source.

2 Because of the fact that the heat does not propagate instantly in metals, the temperature does not pass through the maximum at the same time at various points of the same cross section. There is a lag at points which are farther away from the weld, as shown by curve n-n connecting the points of maximum temperature. This lag is a function of the speed of welding and diffusivity of metal (compare Fig. 8, A, B, and D).

3 Considering an instantaneous position of the electrode, curve n-n separates points in the solid with rising temperature from points in the solid with falling temperature.

As for the influence of the welding conditions, it is seen that:

1 An increase in the speed of welding, Fig. 8, A and B, produces a greater lag in the temperature distribution at a given cross section, hence the heat is more concentrated around the heat source. However, the temperature distribution along the line of welding, behind the electrode, remains unchanged, in accordance with Equation [80].

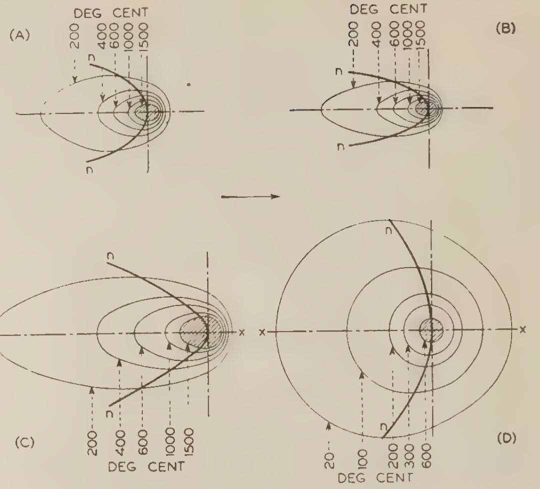

Fig. 8 Temperature Distribution in Thick Butt-Welded Plate
(For details see the text. Courtesy of *The Welding Journal.*)

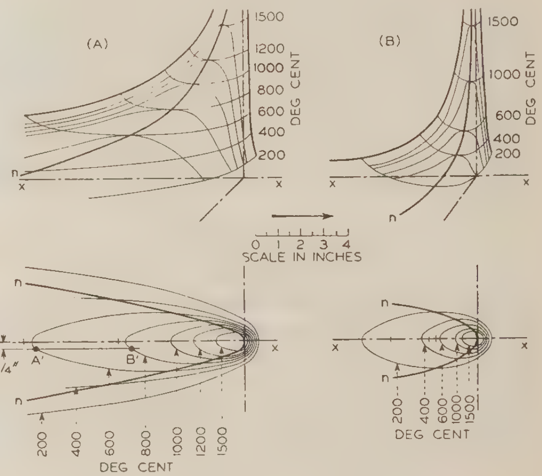

Fig. 9 Temperature Distribution in Thin and Thick Butt-Welded Plate
(*Left*, thin plate, 0.2 in. thick; *right*, thick plate. Same material, i. e., mild steel, and same welding technique in both cases, i.e., 200 amp, 8.3 ipm. Top view isometric projection; bottom view topographic projection. Courtesy of *The Welding Journal.*)

2 An increase in the current intensity, Fig. 8, A and C, extends the range of heat in the solid, but does not affect the shape of isotherms.

3 An increase in the heat conductivity and diffusivity of metal, Fig. 8, C and D, affects both the shape and the size of isotherms. The increase of the range of heating in front of the electrode is especially noteworthy.

4 An increase in the initial temperature, i.e., the preheating, causes a corresponding increase in the numerical value of each isotherm, but does not change its shape or size. However, the preheating has a marked effect upon the rate of cooling, as shown by Equation [85]. This effect is discussed in more detail in the experimental work on weldability (14) and (17).

Basically the same conclusions apply to thin welded plates, as can be easily demonstrated by means of Equation [73]. However, there is a fundamental difference in the conditions of welding of a thin and thick plate, due to the different nature of heat

flow. This difference is well apparent in the shape and size of isotherms drawn in Fig. 9.

Application to Actual Welding Practice

Determination of Cooling Time. In so far as welding practice is concerned, neither of the solutions already discussed is quite satisfactory. Because of the finite thickness of plates, the conditions of welding seldom correspond to that of a very thin or very thick plate. At a short distance from the arc there is a considerable temperature gradient through the thickness, but as the arc moves away from the section considered, the temperature becomes more uniform across the thickness. Thus the heat flow changes from a three-dimensional state to a two-dimensional one, as the distance from the arc increases. This condition has been discussed in Part 1, and an appropriate solution for this case has been obtained in the form of Equation [63].

In actual welding practice it is important to determine the time required for the weld to cool down to a given temperature, rather than to obtain a complete pattern of the temperature distribution around the electrode. Under these circumstances Equation [63] can be simplified by writing for a given spot of the weld bead, t seconds after it has been deposited on a plate of thickness, g

$$\xi = -vt;\ y = 0;\ z = 0$$

$$R_0 = |vt|;\ R_n = \sqrt{(vt)^2 + (2ng)^2},\ n = 1, 2, 3 \ldots \quad [89]$$

Substituting in Equation [63], there follows

$$T - T_0 = \frac{1}{2\pi k} \times \frac{q}{v} \times \frac{\mu}{t} \quad [90]$$

where

$$\mu = 1 + 2\sum \frac{e^{-\lambda v^2 t(s_n - 1)}}{s_n} \quad [91]$$

and

$$s_n = R_n/vt$$

As previously, the rate of heat delivered to the plate will be replaced by the rate of energy input of the welding arc, J multiplied by the heat efficiency of the arc (I.F.), thus

$$T - T_0 = \frac{(\text{I.F.})}{2\pi k} \times \frac{J}{v} \times \frac{\mu}{t} \quad [92]$$

Equation [92] can be made to correspond "formally" to Equation [81] derived for a very thick plate, by introducing a "fictitious" time of cooling $t^* = t/\mu$, which for convenience we shall call the "reduced" time of cooling. With this in mind Equation [92] can be rewritten as follows

$$T - T_0 = \frac{(\text{I.F.})}{2\pi k} \times \frac{J}{v} \times \frac{1}{t^*} \quad [93]$$

As seen from Equation [91], the reduced time of cooling depends upon the thickness of the plate g, on the speed of welding v, and the diffusivity of metal $1/2\lambda$. Actually, however, the thickness is the major factor, and for the usual range of welding speeds, from 6 to 12 ipm, the influence of speed can be neglected.

Fig. 10 shows the relation between the true and reduced time of cooling for various thicknesses of steel, assuming a diffusivity of 1 sq in. per min and a welding speed of 7 ipm. It is seen that as the time of cooling increases, the value of the reduced time of cooling, t^* differs more and more from that of the true time of cooling t, and so more so as the thickness is smaller. Since Equation [93] formally represents the conditions of a very thick plate and of a three-dimensional state of heat flow, the discrepancy between t and t^* is a measure of the departure from a three-dimensional heat flow. On the basis of Fig. 10 it may be predicted that in the vast majority of cases the $1/4$-in. plate will behave like a very thin plate, and the $1\,1/2$-in. plate like a very thick plate, and this prediction is in accordance with the findings of this and the preceding section. Also, it may be safely stated that plates over $1\,1/2$ in. thick will always behave like a very thick plate. Thus for plates thicker than $1\,1/2$ in. the reduced and the true cooling times are the same.

How well Equation [93] fits the actual conditions of welding is best determined by comparing the time of cooling computed from this formula using the chart, Fig. 10, with the time of cooling

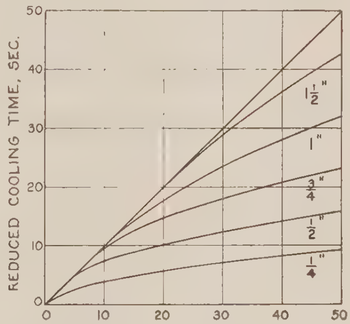

Fig. 10 Relation Between Cooling Time and Reduced Cooling Time for Plates of Various Thickness

measured in the experimental work of Hess and co-workers (14) on steel plates. In doing so, it must be remembered that the actual measurements of temperature were made not in the weld bead itself, but at a small distance from the bead. However, it can be shown that the error due to this factor is small after the first 5 to 10 sec of cooling. Another important factor is the proper choice of the heat efficiency (I.F.), and the heat conductivity k. From previous work (12), the I.F. was assumed to be 0.65. As for the heat conductivity k, a few trials showed that the value of $k = 0.080$ cal per deg C per sec fits best the experimental results at 1300 F. This value is commensurate with that determined previously, Equation [88].

Using the values given and expressing the temperature T in deg F, the energy input J/v in joules per in., and the reduced time t^* in seconds, Equation [93] reads for the case of butt-welded steel plates as follows

$$T - T_0 = 0.22\left(\frac{J}{v}\right)\frac{1}{t^*}$$

hence the reduced time of cooling

$$t^* = \frac{0.22}{T - T_0}\frac{J}{v} \quad [94]$$

Figs. 11 to 15, inclusive, show the agreement between computed and measured times of cooling for various thicknesses using three or four different energy inputs. The agreement in the important range of temperatures from 1300 to 575 F, is in general better than 5 per cent if no preheat is employed. The agreement is not as good with preheated plates, probably because the preheating operation was not maintained during welding (14). Notwithstanding this discrepancy, the general validity of Equation [93] in the range of temperatures from 1300 to 575 F may be considered as sufficiently established for practical purposes.

Determination of Rates of Cooling. By differentiating Equation [92] with respect to time there follows that (18)

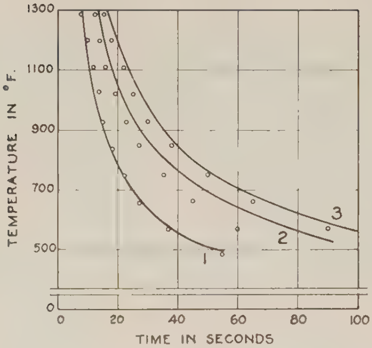

Fig. 11 Weld-Cooling Curves in 1/4-In. Plate at Room Temperature

(Continuous lines computed; isolated circles measured by Hess and coworkers. 1, 17,000 joules per in.; 2, 22,800 joules per in.; 3, 28,000 joules per in.)

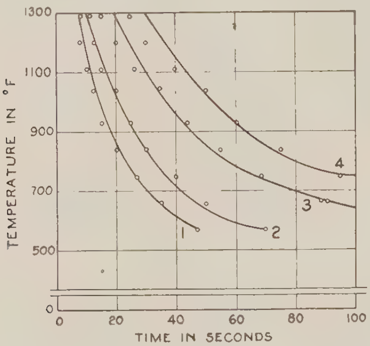

Fig. 12 Weld-Cooling Curves in 1/2-In. Plate at Room Temperature

(Continuous lines computed; isolated circles measured by Hess and coworkers. 1, 34,900 joules per in.; 2, 41,000 joules per in.; 3, 56,100 joules per in.; 4, 68,700 joules per in.)

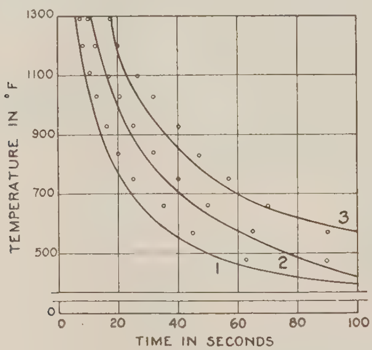

Fig. 13 Weld-Cooling Curves in 3/4-In. Plate at Room Temperature

(Continuous lines computed; isolated circles measured by Hess and coworkers. 1, 46,000 joules per in.; 2, 65,000 joules per in.; 3, 72,000 joules per in.)

$$\frac{dT}{dt} = -\frac{(\mathrm{I.F.})}{2\pi k}\frac{J}{v}\frac{\mu'}{t^2} \dots\dots\dots\dots\dots [95]$$

where

$$\mu' = 1 - 2\sum \frac{e^{-\lambda v^2 t(s_n - 1)}}{s_n^{\,2}}\,[\lambda v^2 t(s_n - 1)s_n - 1] \dots [96]$$

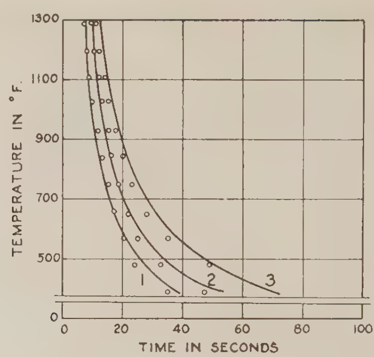

Fig. 14 Weld-Cooling Curves in 1-In. Plate; Temperature 37.4 F

(Continuous lines computed; isolated circles measured by Hess and coworkers. 1, 42,400 joules per in.; 2, 52,700 joules per in.; 3, 67,000 joules per in.)

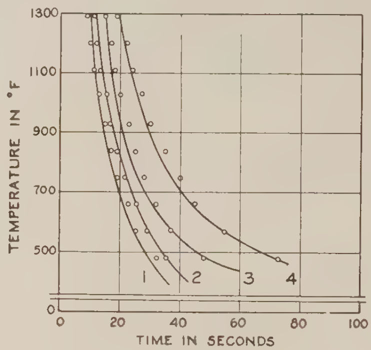

Fig. 15 Weld-Cooling Curves in 1 1/2-In. Plates at Room Temperature

(Continuous lines computed; isolated circles measured by Hess and coworkers. 1, 53,000 joules per in.; 2, 63,500 joules per in.; 3, 76,400 joules per in.; 4, 100,500 joules per in.)

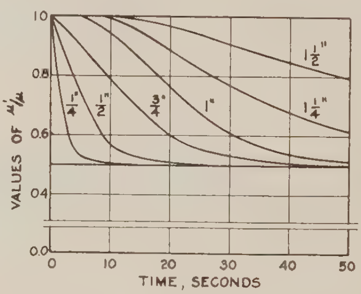

Fig. 16 Relation Between Weld-Cooling Time t and Coefficient $\left(\frac{\mu'}{\mu}\right)$ for Various Thicknesses of Steel

hence by comparing Equation [96] with Equation [92]

$$\frac{dT}{dt} = -\frac{T - T_0}{t}\frac{\mu'}{\mu} \dots\dots\dots\dots\dots [97]$$

Equation [97] is of the same type as Equations [77] and [84] for the thin and thick plates, respectively; the only difference is that the coefficient (μ'/μ) instead of being a constant is a function of time and thickness, and to a much lesser degree of speed of welding and diffusivity. Assuming as previously, the value of 1 sq in. per min for the diffusivity and 7 ipm for the speed of welding, Fig. 16 shows the dependence of (μ'/μ) on time for various

thicknesses of plate. As expected, the values of (μ'/μ) vary between $^1/_2$ and 1, the line $^1/_2$ denoting conditions identical to that of a very thin plate and the value 1, conditions identical to that of a very thick plate. In accordance with previous results it is seen that the $^1/_4$-in. plate behaves like a thin plate less than 5 sec after the cooling process has started, whereas the $1^1/_2$-in. plate remains practically under the conditions of a very thick plate up to 30 sec of cooling.

Equations [93] and [97] along with charts, Figs. 10 and 16, may be used conveniently for a speedy determination of rates of cooling.

Suppose, for example, it is desired to determine the rates of cooling in a $^3/_4$-in. plate when using an energy input of 50,000 joules per in. The three following temperatures of preheat will be examined: 72 deg, 200 deg, and 400 deg F.

From Equation [93] the corresponding reduced times of cooling are 11.8, 13.7, and 18.4 sec.

Referring to the chart, Fig. 10, we find for a $^3/_4$-in. plate the true cooling times, namely, 13.5, 18.0, and 30.5 sec, respectively, and from the chart, Fig. 16, the corresponding values of $(\mu'/\mu) =$ 0.72, 0.64, and 0.54, respectively.

Substituting in Equation [97], the following rates of cooling are obtained: 49.5 deg F per sec, 27.2 deg F per sec, and 10.5 deg F per sec, as compared to the values determined by Hess and coworkers, namely, 51 deg F per sec, 32 deg F per sec, and 15 deg F per sec. As expected, the computed rates of cooling are lower than those actually measured for higher temperatures of preheat.

3 MISCELLANEOUS APPLICATIONS OF THEORY OF MOVING SOURCES

Introduction

Because of lack of experimental data, the applications described in this part are of a more or less tentative character. The purpose of this discussion is not so much to arrive at numerical results as it is to find a scientific approach to the solution of the problems presented. In view of the success which such an approach has had in the problems of welding, it is believed that the considerations which follow will be of more than academic interest.

Rate of Extrusion in Continuous Casting. Continuous casting is a metallurgical process in which liquid metal is fed into a die and extruded in the form of a solid rod at a constant rate by means of a pair of rollers placed behind the die (19). To make the extrusion operation possible the rod must leave the die at a temperature at which the metal can be hot-worked and which consequently represents a given fraction of the temperature of fusion. In other words, the die must be capable of removing the heat from the cast metal to bring about the required drop of temperature. If radial temperature gradients in the rod are neglected, this drop of temperature may be regarded as being produced by a linear flow of heat with surface losses. On the other hand, by the process of extrusion each section of the rod is moved gradually from the inlet of the die where it is at the temperature of fusion T_1 to the outlet of the die, where it has been cooled down to the temperature of extrusion T_2. Thus the heat flow is of a quasi-stationary nature and Equation [23] is applicable. If the die is kept at a constant temperature T_0 and if the length of the die from the inlet to the outlet is L, then Equation [23] may be rewritten as follows

$$T_2 - T_0 = (T_1 - T_0)e^{-(\sqrt{\lambda^2v^2+PH/A} - \lambda v)L} \quad \ldots\ldots [98]$$

The remaining symbols in Equation [98] have the following meaning

λ = one half of the converse value of diffusivity
v = rate of extrusion
P = perimeter of rod
A = cross section of rod
H = dissipation ratio, see Equation [20]

For a circular rod of diameter D, the fraction $P/A = 4/D$. Putting further

$$\ln \frac{T_2 - T_0}{T_1 - T_0} = -m \quad \ldots\ldots [99]$$

Equation [98] becomes

$$m = \left[\sqrt{(\lambda v)^2 + \frac{4H}{D}} - \lambda v\right] L \quad \ldots\ldots [100]$$

or

$$m^2 = 2m\lambda vL + \frac{4HL^2}{D}$$

or

$$2\lambda vD = \frac{4HD}{m} \times \frac{L}{D} - m\Big/\frac{L}{D} \quad \ldots\ldots [101]$$

Assuming a constant value of m, Equation [101] gives the relation between the rate of extrusion v and the mechanical and thermal characteristics of the extrusion. The former is represented by the ratio L/D which is a function of the friction between the die and the cast.

Since the load-carrying capacity of the rod at the outlet of the die has a limiting value, the higher the friction coefficient, the smaller the ratio L/D. On the other hand, the thermal characteristics of the extrusion process are determined (*a*) by the coefficient 2λ, which is the ratio of the volume heat capacity of the rod to its heat conductivity k, i.e., $\frac{c\rho}{k}$ (c = specific heat and ρ = density); and (*b*) by the coefficient H, which is the ratio of the heat dissipation h, and the heat conductivity of the rod, k. Thus both 2λ and H contain k in the denominator. This fact will be brought out later in the discussion.

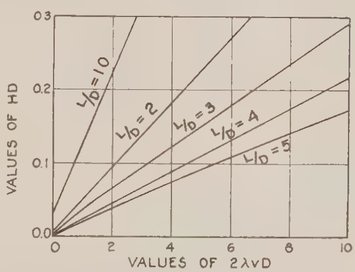

Fig. 17 Relation Between Rate of Extrusion and Heat-Dissipation Ratio in Continuous Casting

Fig. 17 shows the relation between the rate of extrusion and the surface transfer coefficient of the interface between the die and the rod for various values of L/D, computed according to Equation [101]. For the purpose of computation, the fraction $T_2 - T_0/T_1 - T_0$ has been taken as 0.7; thus ln 0.7 = —0.36. Likewise, dimensionless numbers rather than specific values of speed and dissipation ratio have been plotted in order to make the discussion more general. Because of the fact that for values of $L/D > 2$ the second term on the right-hand side of Equation [101] becomes very small as compared to the first one, the speed of extrusion increases almost proportionally to the coefficient H and the relative length of the die L/D. It also increases with the ratio $H/2\lambda$, which as pointed out previously, is nothing else but the ratio of the heat dissipation h to the volume heat capacity

of the metal. The fact that the heat conductivity of the cast metal does not appear in these relations is noteworthy. It shows, for example, that other conditions being equal, the rate of extrusion of copper wire will not be substantially greater than that of iron wire, in spite of the much higher heat conductivity, as the volume heat capacity is not very much different for these metals.

To obtain an approximate idea of the order of magnitude of the rate of extrusion which can be expected, assume the following values for an iron wire 1 cm diam ($\sim$ 3/8-in.) when cast in a metallic die

h = 0.012 cal per sq cm per deg C per sec (20)
k = 0.08 cal per cm per deg C per sec (16)
$c\rho$ = 1.2 cal per cu cm (16)[9]

hence

$$HD = 0.15 \text{ and } 2\lambda D = 15 \text{ sec per cm}$$

Assuming further L/D = 2, 3, and 4, respectively, the following values of $2\lambda vD$ are obtained from Fig. 17: 3.15, 4.8, and 6.6, hence

$$v = 0.2 \text{ cm per sec, } 0.33 \text{ cm per sec, and } 0.44 \text{ cm per sec}$$

Actually, these values may be substantially increased by improving both the heat dissipation h and the lubrication condition between the mold and cast, i.e., the ratio L/D, when using graphite instead of metal for the dies (19). At present, none of these coefficients is very accurately known to permit more than a qualitative statement. However, a method of obtaining a value of some of these coefficients is proposed in the following section.

Method of Determining Value of Diffusivity and Heat-Dissipation Ratio. Equations [23] and [24] provide a convenient method of obtaining, in a single experiment, the coefficients λ and H, the former being one half of the converse value of diffusivity, the latter the dissipation ratio. To this end the metal to be tested is formed into a thin-walled cylinder and surrounded by the medium whose value of H with respect to the metal in question is to be determined. A source of heat in the form of a coil is then moved inside the tube at a rate which will practically produce a uniform temperature distribution in the thickness of the tube, and of sufficient intensity to bring the temperature of the tube to the desired maximum value. The temperature of the surrounding medium is kept at a constant temperature T_0.

By means of thermocouples spot-welded to the tube, the temperature at a given cross section of the latter is taken for three positions of the heat source as follows:

1 Position x_1, for which the temperature has reached the maximum value T_1.

2 Position x_2, preceding the position x_1 and for which the temperature had a given value $T_2 < T_1$.

3 Position x_2' following the position x_1, and for which the temperature has dropped back to the value T_2.

Calling g the thickness of the tube, the ratio of the perimeter P to the cross section A, entering Equations [23] and [24] is simply $1/g$ thus

$$\frac{T_2 - T_0}{T_1 - T_0} = e^{-[\sqrt{(\lambda v)^2 + H/g} - \lambda v](x_1 - x_2)} \qquad [102]$$

and

$$\frac{T_2 - T_0}{T_1 - T_0} = e^{-[\sqrt{(\lambda v)^2 + H/g} + \lambda v](x_2' - x_1)} \qquad [103]$$

These two equations can be rewritten in a more simple form by putting

$$\ln \frac{T_2 - T_0}{T_1 - T_0} = -m \qquad [104]$$

[9] Strictly speaking, coefficient 1.2 must also include the latent heat of fusion.

$$\frac{x_1 - x_2}{g} = u; \quad \frac{x_2' - x_1}{g} = u' \qquad [105]$$

hence

$$\frac{m}{u} = \sqrt{(\lambda vg)^2 + Hg} - \lambda vg \qquad [106]$$

and

$$\frac{m}{u'} = \sqrt{(\lambda vg)^2 + Hg} + \lambda vg \qquad [107]$$

Solving for λvg and Hg there follows

$$\lambda vg = \frac{m}{2}\left(\frac{1}{u'} - \frac{1}{u}\right) \qquad [108]$$

and

$$Hg = \frac{m^2}{uu'} \qquad [109]$$

hence finally the average diffusivity between the temperatures T_1 and T_2

$$\frac{1}{2\lambda} = \frac{4v}{m} \frac{(x_2' - x_1)(x_1 - x_2)}{(x_2 + x_2' - 2x_1)} \qquad [110]$$

and the average dissipation ratio between $(T_1 - T_0)$ and $(T_2 - T_0)$

$$H = \frac{m^2 \times g^2}{(x_2' - x_1)(x_1 - x_2)} \qquad [111]$$

Characteristically, the expression for diffusivity does not depend on the thickness of the tube and the expression of H does not depend on speed.

Control of Factors Governing Flame-Hardening and Continuous-Quenching of Steel Products. The purpose of flame-hardening is to produce a hardened case in steels of proper carbon and alloy content by raising the temperature of the case above the upper transformation point of steel Ac_3, and by quenching it as rapidly as possible in water. When dealing with parts of some length, such as gear rims or rails, the rise of temperature is accomplished by means of an oxyacetylene torch moving at a constant speed v, and the quenching, by a water sprinkler which moves closely behind the torch.

In what follows it will be assumed that the part has a rectangular section and that the flame of the oxyacetylene torch covers uniformly the surface over its whole width g, but that it extends over only a small portion l of its length. Thus the heat flow may be considered as being produced by a narrow plane source of heat moving at a constant speed, v. This case has been examined in Part 1, and Equations [51] and [60] have been derived to describe the temperature distribution around the heat source. Of these, Equation [51] represents an almost uniform rise of temperature which will result in the heated part throughout its thickness, at some distance behind the source, if there are no losses to the surrounding. This rise of temperature is then merely the consequence of the storing up of the heat, and if the latter is delivered at a rate of q thermal units per unit time, the rise of the temperature T_2 behind the source over the initial temperature T_0 is in accordance with Equation [51]

$$T_2 - T_0 = \frac{q}{c\rho agv} \qquad [112]$$

As previously $c\rho$ (c=specific heat, ρ=density) represents the volume specific heat, and v the speed at which the source moves. As for the symbols a and g, their previous meanings must be inverted in order to keep up with the present description of things.

They are accordingly; a = the thickness, and g = the width of the heated part.

As has been shown in Fig. 2, for the case of a very narrow plane source, T_2 also represents the temperature at some depth below the location of the heat source. This temperature is quickly approached by the bulk of the metal as the heat source moves away from the section considered. For convenience, temperature T_2 will be called the "bulk" temperature. With this in mind, Equation [60] represents the relative rise of the maximum surface temperature T_1, as compared to the rise of the bulk temperature of the piece. Fig. 18 shows the same relation graphically. Here for convenience the converse of the relative rise of temperature, namely $(T_2 - T_0)/(T_1 - T_0)$, has been plotted as a function of a dimensionless number λva, whose significance will be explained shortly, and for various lengths l of the source referred to the thickness a of the piece l/a. Assuming that in flame-hardening the surface temperature is required to reach a constant value T_1 and that T_0 is the practically constant room temperature, Fig. 18 shows that the higher the number λva the smaller will be the rise of the bulk temperature T_2. The same result is obtained by decreasing the relative length of the source l/a.

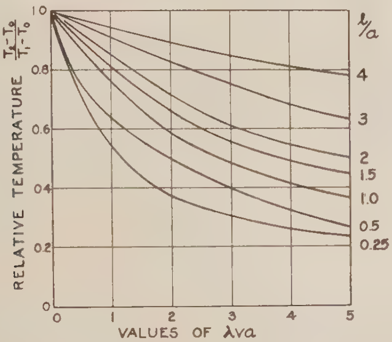

Fig. 18 Relation Between Rise of Surface Temperature T_1, and the Rise of Temperature of Bulk of Material T_2, for Various Conditions of Flame-Hardening

In order to interpret the number λva, attention is called to Equation [112]. If k is the heat conductivity of the material, it will be recalled that by definition

$$\frac{c\rho}{k} = 2\lambda \qquad [113]$$

hence Equation [112] can be rewritten as follows

$$\lambda va = \frac{q}{g} \times \frac{1}{2k(T_2 - T_0)} \qquad [114]$$

In this relation q/g is the rate of heat per unit width and it may be considered as a characteristic of the heat source employed. When comparing Equation [114] with the chart, Fig. 18, it is found that λva is a parameter which allows expression of the maximum surface temperature T_1, or the bulk temperature T_2, whichever is made variable, as a function of the heat characteristic (q/g) of the torch, and also another characteristic of the torch represented by the relative length l/a. Neither of these characteristics can be measured directly, but indirectly both can be determined by using Equation [114] and chart, Fig. 18, in reverse.

To this end, a trial experiment can be set up using a sample of the material to be treated in the form of a rectangular bar and measuring the temperature at a certain point of the surface and below the surface, while the latter is being heated, by passing over it the torch whose characteristics are to be determined. The following measurements are made:

1 Initial temperature, T_0.
2 Maximum temperature T_1 at the surface.
3 Temperature T_2, for which the surface temperature is almost the same as the depth temperature.
4 Speed of motion, v of the torch.
5 Width g, and thickness a of the bar.

Assuming an average constant value for λ and k, the value of q is thus determined from Equation [112] and the value of l from the chart, Fig. 18. For example, if $T_1 - T_0$ has been measured to be 800 C (1440 F) and $T_2 - T_0$ = 400 C (720 F), and if, furthermore, the speed at which the torch moved was 0.85 cm per sec, (0.336 ips), and the section of the bar 5 cm wide and 1 cm thick (about 2 × $^3/_8$ sq in.), then assuming, according to previous discussion, k = 0.08 cal per cm per deg C per sec, and λ = 6 sec per sq cm for steel, the value of q is found from Equation [114] as

$$q = 6 \times 0.85 \times 1 \times 5 \times 2 \times 0.08 \times 400 = 1640 \text{ cal per sec}$$

and l/a from the chart, Fig. 18, with $(T_2 - T_0)/(T_1 - T_0) = 0.5$ and $\lambda va = 5.1$, as 2; hence l = 2 cm (= 0.79 in.).

If each torch thus has been rated for a given value of $\bar{q}$ and l, it then becomes possible to determine for what thickness of steel it is best fitted. As a usual practice, it may be attempted to keep the maximum surface temperature slightly above Ac_3 and the bulk temperature slightly below Ac_1, thus the ratio $(T_2 - T_0)/(T_1 - T_0)$ will be around 0.66, and the value of λva, according to Equation [114] around 0.01 (q/g), if q/g is expressed in cal per cm per sec. Assume, for example, a torch is used for which q/g = 400 cal per cm per sec, and l = 2 cm. It is required to find the thickness a for which it is fitted and the speed of motion v.

From Equation [114] $\lambda va = 0.01 \times 400 = 4.00$ and from the chart, Fig. 18, the intersection of $(T_2 - T_0)/(T_1 - T_0) = 0.66$, and $\lambda va = 4$, gives $l/a = 3$. Thus $a = {}^2/_3$ cm = 0.26 in. and since $\lambda va = 4$, the speed v = 1 cm per sec, or about 23 ipm. If the bulk of the material is to be kept much below Ac_1, for example around 600 F, then $(T_2 - T_1)/(T_1 - T_0)$ is around 0.33 and $\lambda va = 0.005$ (q/g). Hence with the same type of torch as before, $\lambda va = 2$, and $l/a = 0.2$; $a = 2/0.2 = 10$ cm (4 in.). The speed, however, is reduced as much as the thickness has been increased, becoming only 1.5 ipm.

Continuous Quenching. If the heat, instead of being delivered to the piece, is being withdrawn at a rate of q thermal units per unit time, under conditions similar to that of flame-hardening, then Equation [114] and the chart, Fig. 18, can be used to control the final temperature T_2 to which the surface of a piece previously heated to a temperature T_0 will be automatically reheated after having been quenched down to the temperature T_1. This problem can be solved in a manner similar to that outlined for the flame-hardening. To make the conditions more specific, suppose it is desired to improve the hardness of the top surface of a rail by means of water-quenching, followed by tempering at 500 C (950 F). Using the theory of a moving-plane source, this treatment can be attempted in one continuous and single operation as follows: After the rail has left the rolling mill at some 980 C (1800 F), it is quenched by means of a water sprinkler moving at a constant speed v. The problem is then reduced to the determination of the characteristics of the water sprinkler in such a way that the surface of the rail after being quenched to the temperature of 20 C (86 F) is reheated to a bulk temperature of 500 C (950 F). To this end, suppose that the head of the rail is 2 in. wide and $1^1/_2$ in. thick, or approximately 5 × 4 sq cm, and that the heat content of the web can be neglected. Then by assuming that the water spray covers a surface of 5 × 5 sq cm, the value

of l/a is $5/4 = 1.25$. On the other hand, $(T_2 - T_0)/(T_1 - T_0) = \frac{500 - 980}{20 - 980} = 0.5$; hence from the chart, Fig. 18, $\lambda va = 3.2$. If this value is substituted in Equation [114], then by putting as previously $k = 0.08$ cal per cm per deg C per sec, and $\lambda = 6$ sec per sq cm, the value of q becomes

$$q = -3.2 \times 5 \times 0.16 \times 480 = -1240 \text{ cal per sec}$$

Since $\lambda va = 3.2$, the speed of motion is

$$v = 3.2/6 \times 4 = 0.132 \text{ cm per sec} = 3.15 \text{ ipm}$$

A similar computation will show that if it is desired to obtain a tempered martensite at 100 C (= 212 F) at the surface, the heat must be removed from the surface at a rate of 2280 cal per sec with the same speed of motion.

Before closing, emphasis again is put on the tentative character of the foregoing computation. There is obviously need for some preliminary experimentation before the theory expounded can be used as a practical tool much in the same way as this has been done for welding.

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Appendix 1

Use of Instantaneous Sources for Solution of Heat Flow Due to Moving Sources

The solution of heat flow in an infinite solid due to a source moving in a given direction may be obtained by adding the contributions of an infinite number of small instantaneous sources placed one behind the other at infinitely small intervals of time, along the direction of motion. If the direction of motion is taken as the x-axis, then the contribution of an infinitesimal instantaneous point source dQ placed at a distance x' from the origin at a time t', may be represented as follows in so far as the increase of temperature dT of a point (x, y, z) at any moment $t \geqslant t'$ is concerned[10]

$$dT = \frac{\sqrt{2\lambda}\, dQ}{8\pi^{3/2}k} \times \frac{1}{(t-t')^{3/2}} \times e^{-\frac{\lambda}{2(t-t')}[(x-x')^2+y^2+z^2]}$$

where dQ is the quantity of heat of the point source, k is the heat conductivity, and 2λ is the converse of heat diffusivity. However, if the law of motion is given, x' is a known function of t'; and the same is true for the amount of heat delivered by the moving source; thus

$$Q = \phi(t') \text{ or } dQ = \phi'(t')dt'$$

and

$$x' = f(t')$$

hence

$$T - T_0 = \frac{\sqrt{2\lambda}}{8\pi^{3/2}k}\int_0^t \frac{\phi(t')}{(t-t')^{3/2}} e^{-\frac{\lambda}{2(t-t')}\left\{[x-f(t')]^2+y^2+z^2\right\}} dt' \quad \ldots\ldots\ldots [A]$$

This is the most general equation of heat flow in an infinite solid due to a moving-point source. Similar expressions can be derived for moving line and plane sources.

As a particular case, put

$$Q = qt'$$

and

$$x' = vt'$$

where q and v are constants.

Then Equation (A) becomes

$$T - T_0 = \frac{\sqrt{2\lambda}}{8\pi^{3/2}}\frac{q}{k}\int_0^t \frac{dt'}{(t-t')^{3/2}} e^{-\frac{\lambda}{2(t-t')}[(x-vt')^2+y^2+z^2]} \quad \ldots\ldots [B]$$

Put

$$t - t' = \tau^2;\ x - vt = \xi;\ \xi^2 + y^2 + z^2 = R^2$$

There follows

$$T - T_0 = -\frac{\sqrt{2\lambda}}{8\pi^{3/2}}\frac{q}{k}\int_0^t \frac{2\tau d\tau}{\tau^3} e^{-\frac{\lambda}{2\tau^2}[R^2+2v\xi\tau^2+v^2\tau^4]}$$

$$= \frac{\sqrt{2\lambda}}{4\pi^{3/2}}\frac{q}{k} e^{-\lambda v\xi}\int_0^t \frac{d\tau}{\tau^2} e^{-\frac{\lambda R^2}{2\tau^2}-\frac{\lambda\tau^2}{2}}$$

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[10] Reference (9), p. 150.

or letting $t \longrightarrow \infty$

$$T - T_0 = \frac{\sqrt{2\lambda}}{4\pi^{3/2}}\frac{q}{k}\, e^{-\lambda v\xi}\int_0^\infty \frac{d\tau}{\tau^2}\, e^{-\frac{\lambda R^2}{2\tau^2}-\frac{\lambda\tau^2}{2}} \ldots\ldots [C]$$

The infinite integral appearing in Equation $[C]$ can be solved by means of the Bessel function $K_{1/2}(x)$. To this end put a new variable α, such that

$$\tau = \alpha R\sqrt{2\lambda} \ldots\ldots\ldots\ldots\ldots\ldots [D]$$

The substitution of Equation $[D]$ in Equation $[C]$ gives

$$T - T_0 = \frac{q\sqrt{2\lambda}}{4\pi^{3/2}\,k}\, e^{-\lambda v\xi} \times \frac{1}{R\sqrt{2\lambda}}\int_0^\infty e^{-(\lambda vR)^2\alpha^2-1/4\alpha^2}\,\alpha^{-2}d\alpha$$

But according to the theory of Bessel functions (10)

$$\int_0^\infty e^{-(\lambda vR)^2\alpha^2-1/4\alpha^2}\,\alpha^{-2}d\alpha = \sqrt{2\lambda R}\; K_{1/2}(\lambda vR)$$

and furthermore

$$K_{1/2}(\lambda vR) = \frac{\sqrt{\pi}}{\sqrt{2\lambda vR}}\, e^{-\lambda vR},$$

hence finally

$$T - T_0 = \frac{q}{4\pi k}\, e^{-\lambda v\xi}\,\frac{e^{-\lambda vR}}{R}$$

This expression has been derived previously in Part 1 in a more simple way for a three-dimensional case.

Using similar procedures for the line and plane instantaneous sources, Expressions [31] and [18] can be derived for the two-dimensional and linear heat flow, respectively.

Appendix 2

Expansion of Solution [45] in a Fourier Series

As shown by Carslaw,[11] if $f(y)$ is an even function of y which can be expanded, as also $f(y \pm 2na)$, in a Fourier series of cosines of multiples $\pi y/a$, then

$$\sum_{-\infty}^{\infty} f(y \pm 2na) = \frac{1}{a}\int_0^\infty f(y)dy + \frac{2}{a}\sum_1^\infty \cos\frac{\pi ny}{a} \int_0^\infty f(y')\cos\frac{\pi ny'}{a}\, dy$$

Let $f(y) = K_0(\lambda v\sqrt{\xi^2 + y^2})$, then

$$\sum_{-\infty}^{\infty} K_0\left(\lambda v\sqrt{\xi^2 + (y \pm 2na)^2}\right) = \frac{1}{a}\int_0^\infty K_0\left(\lambda v\sqrt{\xi^2 + y^2}\right) dy + \frac{2}{a}\sum_1^\infty \cos\frac{\pi ny}{a} \times \int_0^\infty K_0\left(\lambda v\sqrt{\xi^2 + y'^2}\right)\cos\frac{\pi ny'}{a}\, dy'$$

But, by virtue of the integral representation of Bessel function (10)

$$\int_0^\infty K_0\left(\lambda v\sqrt{\xi^2 + y^2}\right)dy = \frac{1}{\lambda v}\int_0^\infty e^{-(\lambda v\xi)^2t^2-1/4t^2}\,t^{-1}dt \times \int_0^\infty e^{-(\lambda vy)^2t^2}\,d(\lambda vy) = \frac{1}{\lambda v}\frac{\sqrt{\pi}}{2}\int_0^\infty e^{-(\lambda v\xi)^2t^2-1/4t^2}\,t^{-2}dt = \frac{\pi}{2\lambda v}\, e^{-\lambda v|\xi|}$$

Likewise

$$\int_0^\infty K_0\left(\lambda v\sqrt{\xi^2 + y'^2}\right)\cos\frac{\pi ny'}{a}\, dy' = \frac{1}{\lambda v}\int_0^\infty e^{-(\lambda v\xi)^2t^2-1/4t^2}\, t^{-1}dt \times \int_0^\infty e^{-(\lambda vy')^2t^2}\cos\left(\frac{\pi n}{\lambda va}\lambda vy'\right) d(\lambda vy')$$
$$= \frac{\sqrt{\pi}}{2\lambda v}\int_0^\infty e^{-(\lambda v\xi)^2-\mu_n^2/4t^2}\,t^{-2}dt = \frac{\pi}{2\lambda v\mu_n}\, e^{-\lambda v\mu_n|\xi|}$$

where
$$\mu_n = \sqrt{1 + \left(\frac{\pi n}{\lambda va}\right)^2}$$

hence recalling that

$$2\lambda = \frac{c\rho}{k}$$

we obtain expression, Equation [48] or [49], according to whether ξ is positive or negative.

Discussion

R. H. Cameron.[12] This paper is of considerable interest to the writer as a mathematician, because it shows that data of important practical significance can be calculated by the use of higher mathematics and higher mathematical techniques. While undoubtedly many of the author's conclusions have been or will be checked directly by experiment, his methods make it possible quickly to calculate data and plot curves which could be obtained directly only by a very large number of time-consuming and costly experiments. His theory of quasi-stationary states simplifies many practical problems to the point where calculation is possible and provides a method of attack on other problems which have not yet been put on a mathematical basis.

The author's technique changes a problem involving moving heat sources into one involving only fixed sources or singularities, and thus extends the classical methods of treating fixed sources to the more difficult problems of moving sources. These methods apply to such apparently diverse problems as arc welding, the rate of extrusion of a continuous casting, continuous quenching, and the heating effects of the passage of a bullet in a gun barrel.

Finally, his methods can be used in connection with experimental work to suggest the design of experiments, the direction they should take, and the interpretation of the results. His method for determining the values of diffusivity and heat-dissipation ratio is an example in point.

M. W. Rubesin[13] and R. C. Martinelli.[14] The author has presented an extremly interesting and useful contribution to the field of heat conduction. It is interesting to note that, due to the quasi-stationary state existing about the moving source, the method of relaxation can be readily applied to the problem.

A heat balance is made on a lattice of sides δ and depth $\delta/2$ situated directly below the moving source and moving with the source at a velocity v. Consideration is taken of the heat generated by the source, the heat conducted into the lattice by conduction, and the heat carried into the lattice by the movement of material through the lattice. Heat balances are made on other lattices surrounding the one about the source and a set of relaxation patterns established. Calculations show that the results of this numerical method check reasonably with the analytical results of the author.

[11] Reference (9), p. 159.

[12] Professor of Mathematics, University of Minnesota, Minneapolis, Minn.

[13] University of California, Berkeley, Calif.

[14] Assistant Professor of Mechanical Engineering, University of California. Mem. A.S.M.E.

Heat losses from the surface can also be accounted for by the method of relaxation.

Author's Closure

Professor Cameron's comments coming from a mathematician are very much appreciated. The technical man neither ignores nor belittles the value of the mathematics, but the thing he is primarily interested in is the final numerical value of the solution and not the way in which the latter has been obtained. Unfortunately, many a mathematician does not appreciate this point of view as fully as Professor Cameron does. They are inclined to consider the ultimate step of the analysis which consists of putting the solution in a form suitable for numerical computation as some sort of "puka" mathematics to be left to the computing machines. While the latter certainly are tremendous time savers, they hardly afford as complete a view of the problem as a mathematical analysis does, especially when it comes to the consideration of singularities. Some of the problems of heat flow require the use of the highest mathematical techniques, and it is only through the co-operation of mathematicians who are skilled in those techniques that a substantial progress in the theory of heat flow can be made.

The emphasis on the mathematical analysis does not preclude the use of numerical methods for particular applications. From this point of view the remarks made by Mr. Rubesin and Professor Martinelli are most welcome. The author is not enough familiar with the relaxation method to see the advantages that it offers in the treatment of a general problem of heat flow, like the one developed in the present paper, but he is aware of the possibilities which it affords in solving particular problems with specific boundary conditions. For example, he would be very much interested to see the discussers treat the following two problems which are of importance in arc welding:

1 Heat flow due to a moving plane source of a circular shape, and

2 Heat flow in a plate heated by a moving point source on one face and cooled by liquid medium on the other face.

The solution of the first problem may give a better insight into the phenomena of temperature distribution in the molten pool, and the second is of especial interest in ship-repair welding.

In closing the author wishes to thank all discussers for their kind interest in his paper.

Cutting Action of Reamers

By T. F. GITHENS,[1] CLEVELAND, OHIO

The cutting action of reamers resembles in many ways the cutting action of boring tools or twist drills. There is a common misconception that a reamer somehow gets into a hole and then scrapes the hole to size by means of its longitudinal cutting teeth. What really happens is that a reamer, a small amount larger than the hole to be finished, must cut its way into the hole by its entering teeth; and then produce a smoothly finished hole, round, straight, in proper alignment, and as near to the standard reamer size as possible. Reamers are provided with three kinds of clearance, point, peripheral, and longitudinal. Reamers can be made with combinations of positive, negative, and zero radial and axial rakes. The angle of the point and the feed per revolution affect the true rake angle of the cutting edge. Reamers must be kept sharp. The standard reamer made accurately to size and carefully used to produce a standard hole has been the most important factor in interchangeable manufacture. All classes of fit can be made by proper allowance on the shaft and the maintenance of the dependable standard hole produced by the standard reamers. These and other facts concerning reamers are explained in this paper.

Machine and Hand Reamers

A MACHINE reamer is a tool used for enlarging a hole, previously formed, and for freely cutting this hole as round, smooth, straight, in proper alignment, and as close to the standard or exact size as possible (1).[2]

A reamer is accomplishing its purpose when it cuts freely a round, smooth, straight hole its own size.

The principal parts of a machine reamer are shown in Fig. 1. These definitions and those for hand reamers are explained in the American Standard for Reamers (2).

The principal parts of a hand reamer are shown in Fig. 2.

Reamer Clearances

A reamer without clearance cannot cut because its cutting edges cannot get under the surface of the metal to lift or sever the chip from the metal. If forced into a hole it merely rubs or burnishes the surface of the hole and either heats the tool sufficiently to draw the temper, or tears the hole, or does both. A reamer having proper clearance cuts freely and smoothly (1).

There are three kinds of clearance on reamers, as follows:

1 Clearance on the entering ends of the teeth. This clearance is illustrated in Fig. 3 and is sometimes called "point" clearance.

2 Clearance along the lands or peripheral part of the reamer. This is sometimes called "radial relief" or clearance. This is illustrated in Fig. 5.

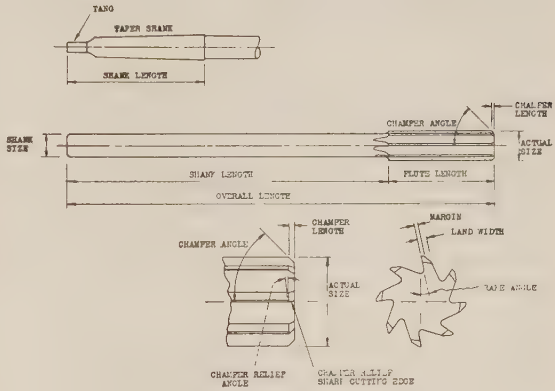

Fig. 1 Machine Reamer

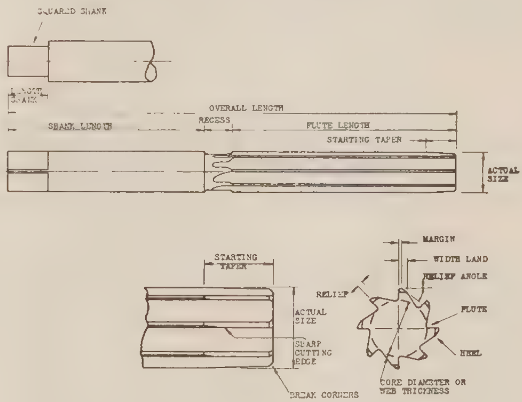

Fig. 2 Hand Reamer

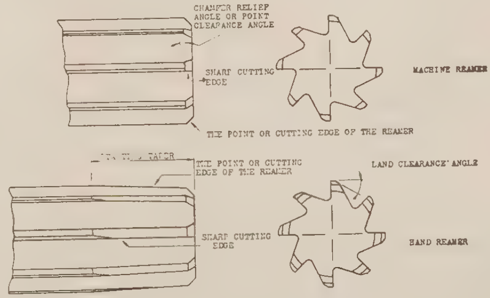

Fig. 3 Cutting Ends or Points, Showing Cutting Edges and Clearances

3 "Longitudinal relief" or "back taper," which is a very slight taper of about 0.0001 in., which makes the reamer smaller toward the shank in order to prevent the back end from enlarging the hole or dragging and thereby roughing up the finish of the hole, Fig. 7.

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[1] Mechanical Engineer, Cleveland Twist Drill Company. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Research Committee on Cutting Data and Bibliography and presented at the Semi-Annual Meeting, Detroit, Mich., June 17–20, 1946, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

Point Clearance

The entering cutting edges on chucking reamers are usually beveled at an angle of from 40 to 50 deg for the purpose of keeping the extreme cutting corner as obtuse as possible, which helps to keep it sharp for a longer time. This cone shape also aids the reamer in centering itself in the work. These cone-shaped edges must be ground to a sharp edge with clearance for end-cutting, Fig. 3.

Hand reamers have a much longer taper, about 0.015 in. per in., and they are used for finishing purposes only. These reamers are but slightly smaller at the entering end, as it is necessary to remove only a small amount of metal in the finishing process. This long taper allows a scraping cut to be taken in such a manner that very accurate and smooth holes are produced. The taper edges must be ground to a sharp edge with clearance. A bevel of about 45 deg is also put at the long end of the tapering point to aid the hand reamer in entering the hole, but there should be so little stock left in the hole that this bevel should do no cutting, Fig. 3.

Fluted chucking reamers which are sometimes required for very accurate and smooth holes often have a slight second bevel in addition to that already mentioned, which acts like the taper on a hand reamer, thus making a smoother hole than would otherwise be the case. This is shown in Fig. 4; the length of this second taper need be only $^1/_{16}$ in. and must be ground to a sharp edge with proper clearance (1).

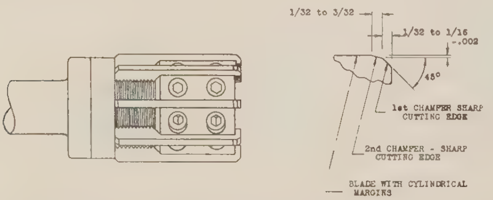

Fig. 4 Second Bevel on Point or Cutting Edges of Adjustable-Blade Reamers

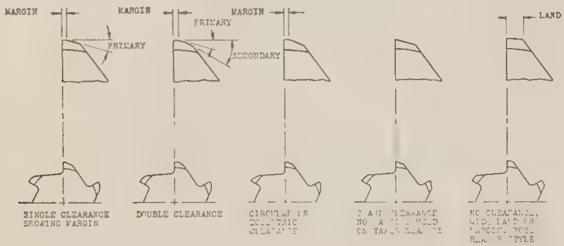

Fig. 5 Types of Side Clearance or Radial Relief

Side Clearance or Relief

Side clearance is illustrated in Fig. 5 and is formed by grinding away a portion of the land, usually leaving a slight cylindrical margin. The action of a reamer without side clearance is best understood by comparing a single tooth of the reamer with a boring tool having no clearance behind the cutting edge, as shown in Fig. 6, in which H is the hole to be bored, D the cutting edge, A is the side of the tool or peripheral land, with small sector B exactly concentric with the hole. Clearance is shown on the end of the boring tool at C.

It will be seen that great pressure would have to be exerted in the direction indicated by arrow X to hold the tool against the work, because the tool can cut the metal only at edge D, which enters the metal in the direction of the length of the hole, and therefore there is no cutting action accomplished by the part shown at B. This part only rubs and as there is some spring to the tool it does not cleanly sever the chip produced by cutting edge D. Therefore part B is forced outward at each succeeding revolution as the tool is advanced, which results in producing a tapering hole as shown on dotted line E. This could be overcome only by exerting enough pressure in the direction of arrow X to hold sector B absolutely to its line of travel and by having the surface so hard and smooth that it could not tear the wall of the hole (1).

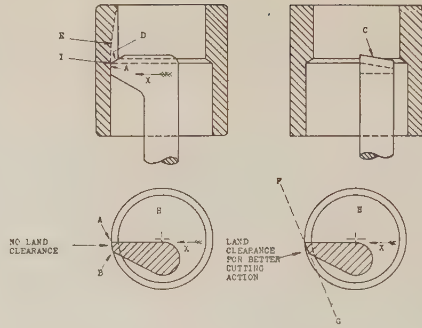

Fig. 6 Cutting Action of Boring Tool
(Land clearance improves performance.)

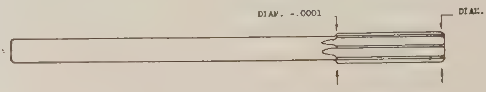

Fig. 7 Longitudinal Clearance or Back Taper

Another factor that also tends to produce a tapered hole and that requires additional pressure in the direction of the arrow is the dulling or rounding over of the extreme corner of the tool at 1. This dulling action is very severe at this point. The easiest and surest way to overcome these difficulties is to relieve the surface B along a line represented by FG. This reduces the required pressure in the direction shown by arrow X, and also lessens the trouble due to the dulling of the extreme cutting corner, and makes it possible to bore a hole the sides of which are practically parallel. The hole will also be smoother, as the side of the tool will scrape out the roughness left by the dulled corner as it follows after it. It must therefore be evident that a reamer without radial or side clearance is working under great difficulties and the chances for accurate work are small. It is also evident that as the surfaces of the reamer, which correspond to B, advance into the metal they should be lubricated and this can be done by lubricating the cutting edges as well.

Side relief is necessary for accurate reaming, yet a slight margin or cylindrical portion of the land must be left on the reamer to aid it to maintain correct size of the hole and allow the reamer to be sharpened many times without losing its original size. If the margin can be prevented from abrading or wearing away, the reamer can be sharpened back indefinitely. Of course some abrading and wear cannot be prevented, and commercial reamers are usually made a few ten-thousandths oversize so as to allow longer tool life and still produce accurate holes.

Rake Angles

The "rake angle" of a tool is the angle between the top cutting surface of a tool and a plane which is perpendicular to the surface of the work and to the direction of motion of the tool with respect to the work.

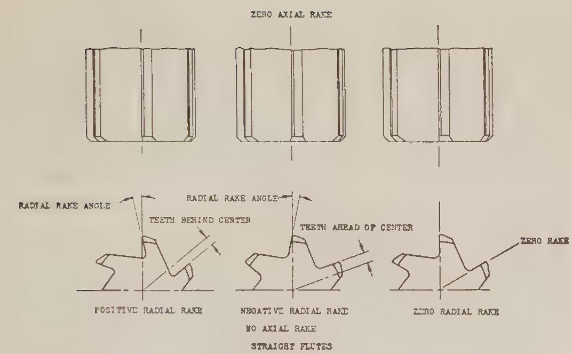

Fig. 8 Types of Radial and Axial Rake

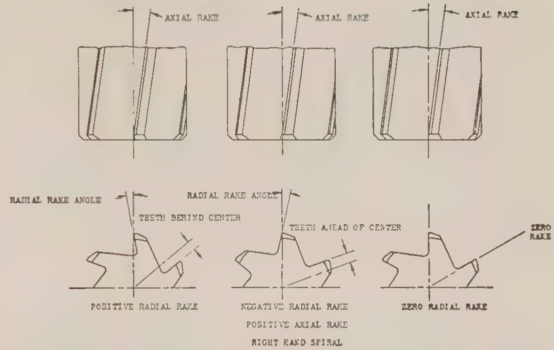

Fig. 9 Types of Radial and Axial Rake

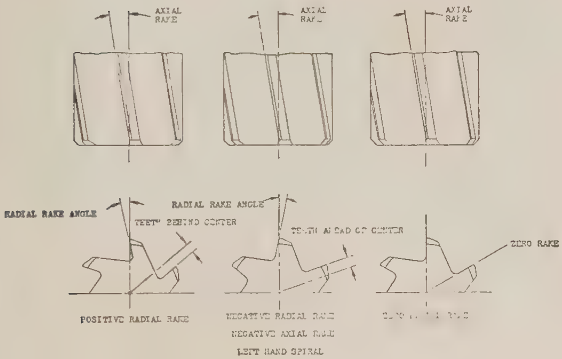

Fig. 10 Types of Radial and Axial Rake

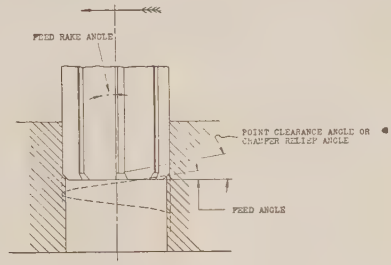

Fig. 11 Rake Angle Due to Feed

(The angle that the reamer makes with the work as it feeds into the hole. Point clearance angle or chamfer relief angle should be greater than the feed angle shown. The action of the feed increases the rake angle of the reamer by the amount shown by the "feed-rake angle.")

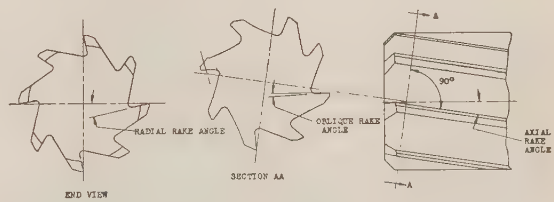

Fig. 12 Rake Angle Normal to Helix Angle or Oblique Rake Angle

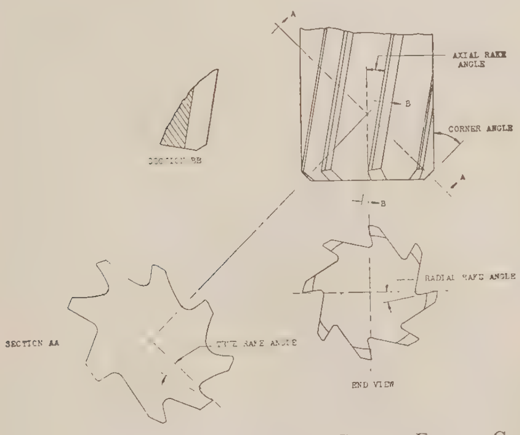

Fig. 13 True Rake Angle, Normal to Cutting Edge of Chamfer, Resulting From a Given Combination of Radial-Rake Angle, Axial-Rake Angle, and Corner Angle or Point Chamfer

The "axial-rake" angle is the same as the "helix angle." The "radial-rake" angle is the angle between the face of the flute and a radial line drawn to edge of the margin.

The best rake angle for each material, speed, feed, and machine-operating condition is a matter of trial and experiment.

Figs. 8, 9, and 10 show some possible combinations of axial and radial rakes with which reamers can be made.

Fig. 11 shows that the manner in which the reamer approaches

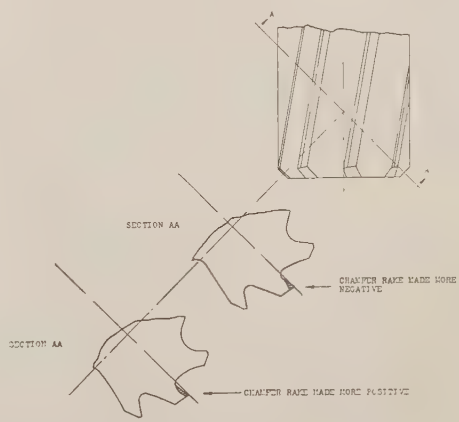

Fig. 14 Positive or Negative Rake May Be Varied on Cutting Edges of Chamfer

the work affects the rake angle. The larger the feed per revolution the larger is the rake angle due to the feed.

Fig. 12 shows how the axial- and radial-rake angles combine to affect the oblique rake angle shown (3).

Fig. 13 shows a section at right angles to the point cutting edge at the actual and cutting edge of the reamer. This true rake angle depends upon the combined values of the radial rake, the axial rake, and the point angle or bevel.

Fig. 14 shows how in special cases with great care in grinding, the cutting rake of the point cutting edges may be varied for different materials and cutting conditions.

Form of Flutes

The style of flute is of the greatest importance, as its shape and area determine the relative strength of the tooth and the ability to carry away the chips. Large chip space and strength are desirable in all reamers but they are of greater importance in reamers designed for taper-reaming and for use in machines. The cutting face of a reamer tooth should have a fillet, see Fig. 15,

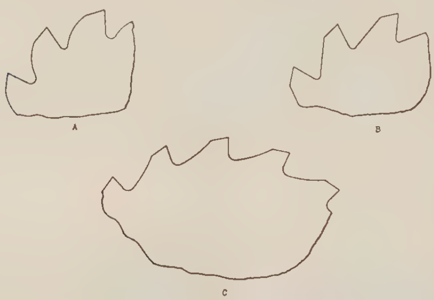

Fig. 15 Different Forms of Flutes Used for Various Types of Reamers

beginning about midway between the top and bottom of the tooth where great strength is desired, as in machine reamers, and nearer the bottom where the strength is of less importance, as in hand reamers. This fillet does not reduce the chip area materially because the chips tend to curl in most metals and this curl conforms to the shape of the fillet. Theoretically, the shape of the back of the tooth should be a convex parabola from the cutting edge to the base or junction of the fillet to obtain maximum strength. For manufacturing reasons a straight line or a concave curve which is tangent to the fillet and which lies outside the parabola is standard practice.

At A in Fig. 15 is shown a tooth with convex parabolic flank which is rarely used except on heavy chucking reamers where the cuts are heavy; at B is shown the straight flank; while at C is shown the concave-curve flank largely used by most manufacturers. Shape C provides a large chip area, and where uneven spacing is practiced the lands are most easily kept at uniform width (1).

Prevention of Chatter

Another desirable feature in a reamer, and especially a taper or hand reamer, is that it be able to cut a round smooth hole without chattering. There is an erroneous opinion prevalent that an even number of flutes chatter more than an odd number. This is a fallacy and our experience shows that an odd number of flutes will chatter as readily as an even number, especially when the reamer has more than four flutes.

Chatter may sometimes be eliminated by reducing the amount of clearance. To make reamers suitable for reaming most kinds of material met with in ordinary practice a considerable amount of clearance must be given, and this would produce chatter by biting in unless it is offset by some other features.

Chatter may sometimes be reduced by making the setup as rigid and strong as possible and by using pilots and guide bushings.

Chatter may also be reduced by cutting down the speed of the reamer. Speeds must not be so high as to permit chatter (5). Too low a feed may in some cases cause chatter due to glazing of the hole. Too much positive rake angle or too much negative rake angle may also cause chatter.

Reamers as commercially made usually have the cutting teeth unequally spaced. Fig. 16 shows a reamer with even spacing and a reamer with one type of uneven spacing. The reason for providing this uneven spacing is to reduce the possibility of chatter.

Fig. 17 shows a chucking reamer about to enter a hole. If the axis of rotation A-A of the reamer is exactly in line with the axis of the hole B-B, and all cutting conditions correct, such as speed not too high, feed not too low, amount of stock sufficient for a chip, rake not too hooked or too negative, the work and reamer rigidly held and guided if possible, there should be no chatter and a round, smooth, accurately sized hole should be produced.

However, if the axis of the reamer is out of line with the axis of the hole, or if a hard spot in the steel is cut, or if the reamer teeth are not sharpened accurately so that they are all at the same angle and length; one tooth of the reamer may cut a little deeper than the others, and in addition to the motion of rotation about its axis, the reamer will also acquire a secondary form of rotation as shown in Fig. 18.

This shows the situation greatly exaggerated. Tooth 1 has cut deeply into the metal and for an instant the reamer will rotate about the instantaneous center 1, until tooth 2 bites into the metal. Tooth 2 now becomes the instantaneous center of rota-

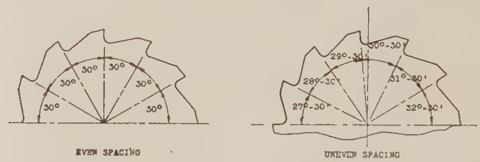

Fig. 16 Unequal Spacing of Flutes to Avoid Chatter

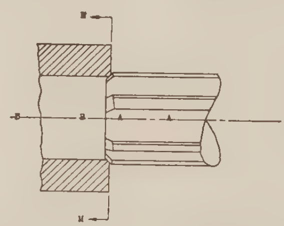

Fig. 17 Unequal Spacing of Flutes to Avoid Chatter

Fig. 18 (*left*) Unequal Spacing of Flutes to Avoid Chatter
(Section M-M of Fig. 17 greatly exaggerated. Reamer while rotating about its axis rotates also about center 1, a small amount, until tooth 2 cuts.)

Fig. 19 (*right*) Unequal Spacing of Flutes to Avoid Chatter
(Section M-M of Fig. 17 greatly exaggerated. Reamer while rotating about its axis rotates also about center 2, a small amount, until tooth 3 cuts.)

tion (see Fig. 19) and the reamer rotates until tooth 3 bites in, and so on all around the reamer. This continues while the reamer is rotating about its axis. A very bad condition such as described would result in chatter. If the teeth are unevenly spaced so that the distances between 1 and 2, 2 and 3, etc., are not the same, the jump or biting-in of the teeth will be unevenly distributed, and the cut gradually may become smooth and chatterless.

Cutting Action

The cutting action of a reamer has been shown in Fig. 6 to be similar in many respects to that of a boring tool. In Fig. 20 is shown a boring tool about to enlarge or finish-cut a rough hole. It must cut it sway into the hole by its sharp, cleared, end cutting edge *D*. Similarly the reamer must cut its way into the small rough hole by its sharp, cleared, end cutting beveled teeth or point.

The cutting action of a reamer is in some respects similar to that of a drill. Fig. 21 shows a drill about to cut through and enlarge a rough hole. The drill is larger than the rough hole and in order to enter, the drill must cut its way in. Similarly, in the same figure the reamer must be larger than the rough hole and must cut its way into the hole by its end cutting or beveled teeth. The cutting action of the longitudinal margins or lands is explained in connection with Fig. 6.

Fig. 22 shows the similarity of the cutting action of a drill and a hand reamer. The cutting action of a reamer is in some respects similar to that of an end mill as shown in Fig. 23, assuming that the direction of feed for both tools is helically axial.

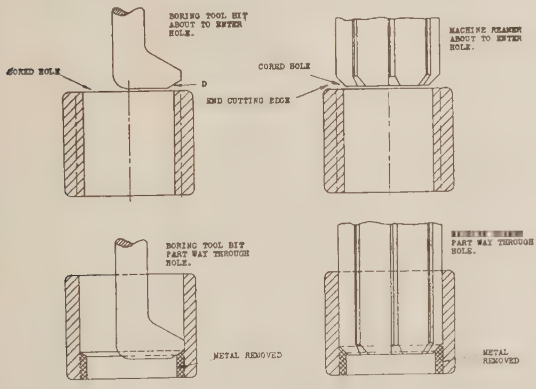

Fig. 20 Similarity of Cutting Action of a Boring-Tool Bit and a Reamer

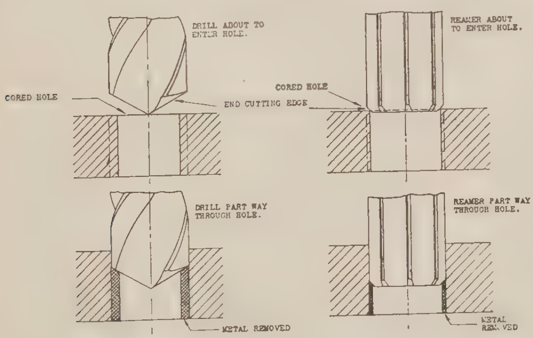

Fig. 21 Similarity of Cutting Action of a Twist Drill and a Machine Reamer

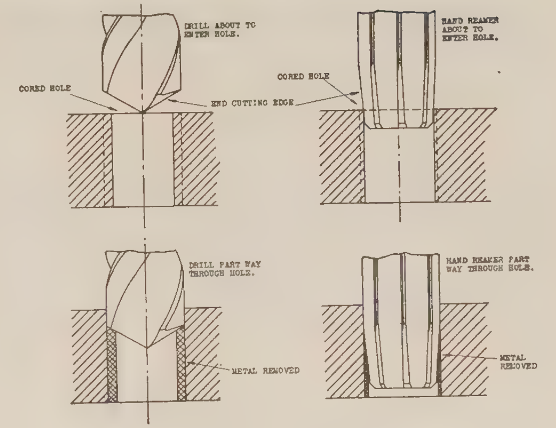

Fig. 22 Similarity of Cutting Action of a Twist Drill and a Hand Reamer

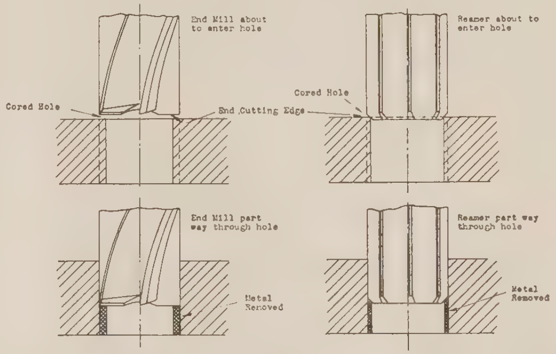

Fig. 23 Similarity of Cutting Action of an End Mill and a Machine Reamer

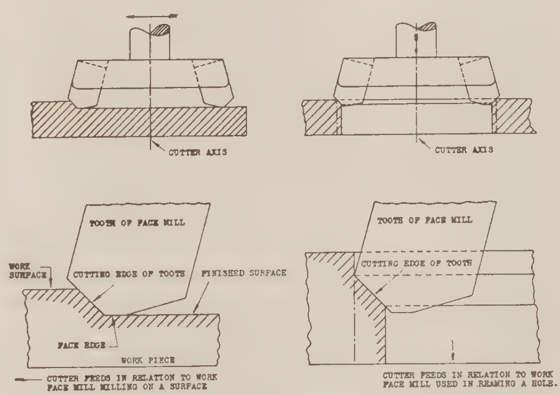

Fig. 24 Similarity of Cutting Action of a Face Mill and a Machine Reamer

The bodies of both tools fit in the holes they cut. Of course, in an end mill the direction of feed is usually perpendicular to the axis of the tool, not along the axis. Also an end mill will not usually produce as accurate a hole as a reamer because its longitudinal teeth are backed off sharp with no margins or lands.

Fig. 24 shows how a face mill, usually used for cutting along the surface of a piece of work, might be used to enlarge a hole.

This shows that the cutting action of face-milling teeth is essentially similar to that of reamer end cutting teeth. The lands or margins on a reamer serve the same purpose of smoothing the cut that the faces do on a face mill.

Reamer Sharpening

After a reamer has been used for a number of holes it will show signs of wear. These signs should be carefully watched and studied and at the proper time the reamer should be resharpened. In general, higher speeds will cause the reamer to dull faster, but higher speeds also give higher production. The finish required also determines the speed possible.

It will be noticed that the reamer will become dull along its once sharp edges *A-B*, Figs. 25 and 27, with the greatest wear at the corners *A*. After a number of holes have been reamed the corner *A* will become completely rounded off. If the reamer is continued in use after this corner has become worn, the sharp edge along the margin *A-C* will also become worn and abraded starting at *A* and working back along the margin or land *A-C*. If the reamer is kept in use in this dull condition it will be completely ruined, the margins all the way back to *D* will become abraded, and the reamer cannot ream a smooth hole to the correct size.

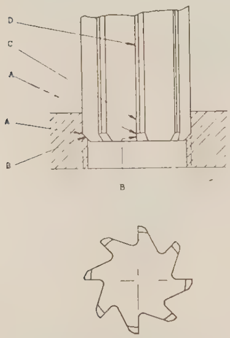

Fig. 25 Wear of Reamer Cutting Edges

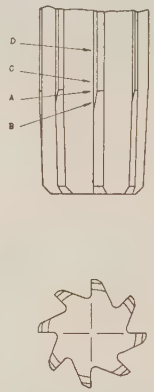

Fig. 27 Wear of Hand-Reamer Cutting Edges

The first result of using a dull reamer will probably be undersize holes, the reamer will be actually forced through the hole without cutting its true size. This of course, causes unnecessary wear on the reamer lands. If at the first sign of holes coming smaller in size the reamer is sharpened, holes up to size will be again produced.

To sharpen the reamer it is necessary to grind only the point or cutting edges back as shown at *A-D*, Figs. 26 and 28, with the proper clearance. The edges are ground back until the corner and marginal dullness between *A* and *C* is removed.

The importance of frequent reamer point sharpening for accurate and smooth holes cannot be overemphasized, especially as it is as simple an operation as drill point sharpening, and a very similar one. Frequent sharpening prolongs the life of the accurately sized reamer margins and lands, and guards against their abrading or wearing undersize and producing undersize holes.

Just how often a reamer should be sharpened or what tool life a reamer should have in minutes of use per grind or number of inches of holes reamed per grind must be determined by trial for each particular application.

In general, the higher the cutting speed the shorter will be the tool life, but of course the high cutting speed has the advantage of high production, if the finish and accuracy of the holes produced are satisfactory.

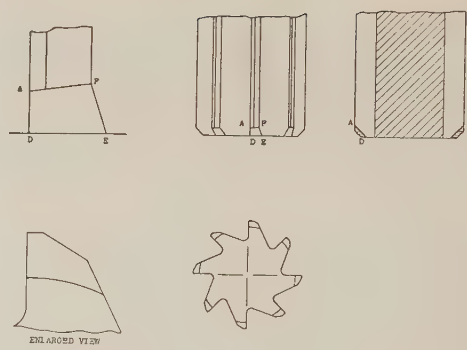

Fig. 26 Sharpening of Reamer-Point Cutting Edge

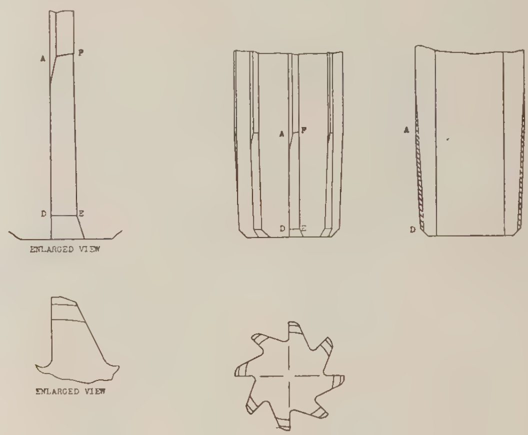

Fig. 28 Sharpening of Hand-Reamer Point Cutting Edge

On single-operation machines where it is a simple matter to take out a tool and resharpen it, low tool life and high production are desirable. On a multispindle automatic it might be uneconomical to change tools more than once a shift, and here slower speeds and longer tool life on a particular operation may be better.

In any case, the use of a dull tool is very poor economy. It is destructive of the reamer and productive of poor and unaccepta-

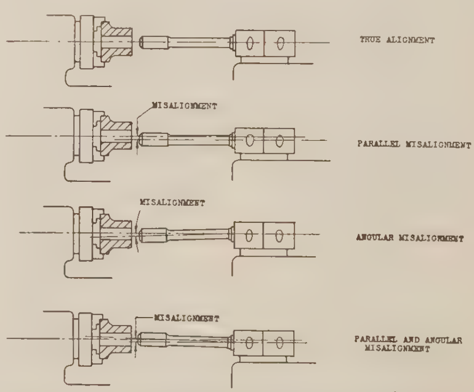

Fig. 29 Floating Toolholders and Alignment

ble work. Some methods for sharpening reamers are shown in Fig. 28.

Floating Toolholders

It should always be borne in mind that a reamer produces its best work when taking light cuts. If the holes are being finish-reamed in a machine, good work has been done by holding the reamer in a chuck, collet, or holder that will float smoothly without jumping. Some good work is done with holders that allow the reamer to float perpendicular to its axis as well as at an angle (1). This floating action is necessary because it is almost impossible to hold the reamer so that it is exactly concentric with the hole, and at the same time have the line of travel of the axis of the reamer coincide or line up with the axis of the hole. A reamer must align itself with the hole it is cutting for extreme precision. The lands or margins of the reamer must be a snug fit in the hole that the reamer is finishing. The axis of the rotating reamer must coincide within a very small amount with the axis of the tool.

This floating alignment and rigidity of cut are both difficult to attain but are necessary for good work.

Speeds, Feeds, and Lubricants

No definite rules can be given to cover the speeds and feeds that should be used in reaming. Several factors govern the correct feed and speed. The material to be reamed and the material of the reamer are perhaps the most important governing factors. The finish desired in the hole and whether hand feed or machine feed is used, govern the selection of the proper feed. The type of lubricant used, or whether the reamer is piloted or guided in any way, also influence the decision.

In general, reamers should be run at about one half to two thirds the speed of the corresponding drill. It is always best to start the reamer somewhat slower than this, and gradually increase the speed until the best condition is found. If the reamer is run at too slow a speed the production will suffer, while if run too fast, the wear on the reamer will dull it too soon. Somewhere between these two extremes is the proper speed which gives a high productivity with a long reamer life.

Feeds should be from 2 to 3 times the feed of the corresponding drill. The higher feed causes increased production and reduced tool wear. If too fine a feed is used, the reamer will be subjected to unnecessary wear. Too coarse a feed will produce feed marks in the hole. The desired finish must be balanced against the production per hour.

Feeds are governed by the size of the reamer and the material reamed. A general rule is first to try a feed of 0.002 to 0.004 in. per revolution for reamers smaller than $^1/_8$ in.; 0.004 to 0.008 for reamers $^1/_8$ to $^1/_4$ in.; 0.008 to 0.014 for reamers $^1/_4$ to $^1/_2$ in.; 0.014 to 0.030 for reamers $^1/_2$ to 1 in.; and 0.030 to 0.050 for reamers larger than 1 in. Alloy and hard steels should generally be reamed with the lighter feeds, while cast iron, brass, and aluminum are sometimes reamed with the heavier feeds. Suggested speeds for reamers are given in Table 1.

TABLE 1 SUGGESTED SPEEDS FOR HIGH-SPEED REAMERS

Material reamed	Speed, fpm
Aluminum and its alloys	130–200
Bakelite	70–100
Brass and bronze, ordinary	130–200
Bronze, high-tensile	50–70
Cast iron, soft	70–100
Cast iron, hard	50–70
Cast iron, chilled	20–30
Malleable iron	50–60
Magnesium and its alloys	170–300
Monel metal	20–30
Steel, machinery (0.2 to 0.3C)	50–70
Steel, annealed (0.4 to 0.5C)	40–50
Steel, tool (1.2C)	30–40
Steel, forgings	30–40
Steel, alloy	30–50
Steel, stainless, free-machining	40–50
Steel, stainless, hard	20–30

Lubricants

Lubricants have many functions, several of which are as follows:

1 To cool both the cutting edges of the tool and the work being machined. This can best be done by directing as large a volume of the coolant as possible on the cutting edges. On thin-walled work it often helps to allow a large volume of flow onto and around the piece.

2 To lubricate the chips; this aids in chip clearance.

3 To improve the finish of the work. The selection and proper application of the lubricant will materially influence the machined finish.

It is suggested that lubrication problems be referred to a reputable manufacturer of cutting oils. The following list of lubricants should be used as suggestions only:

1 Aluminum and its alloys: Soluble oil, kerosene, and lard-oil compounds, light nonviscous neutral oil, kerosene, and soluble-oil mixtures.

2 Brass: Dry, soluble oil, kerosene, and lard-oil compounds, light nonviscous neutral oil.

3 Copper: Soluble oil, winter-strained lard oil, oleic-acid compounds.

4 Cast iron: Dry or with a jet of compressed air for a cooling medium.

5 Malleable iron: Soluble oil, nonviscous neutral oil.

6 Monel metal: Soluble oil, sulphurized mineral oil.

7 Steel, ordinary: Soluble oil, sulphurized oil, high E.P. value mineral oil.

8 Steel, very hard and refractory: Soluble oil, sulphurized oil, turpentine.

9 Steel, stainless: Soluble oil, sulphurized mineral oil.

10 Wrought iron: Soluble oil, sulphurized oil, high animal-oil-content mineral-oil compound.

Regular Reamers and Basic Standard-Hole System of Tolerances

Varying degrees of accuracy result from the use of different tools employed in the drilling, boring, and reaming of holes. The cost of obtaining the accuracy required by some kinds of work would be prohibitive for others. Therefore the proper tools for producing holes in any class of work are those best adapted to produce the required accuracy from the standpoint of an over-all economy. By an over-all economy is meant the greatest number of suitably finished holes that can be produced for each dollar paid out for tools, labor, and machine expense.

Drilled holes are neither round, straight, nor of uniform diameter when compared with the accurate holes required by most industries, although they may be sufficiently accurate to meet the requirements of holes for rough bolts, rivets, and similar classes of work. When a close fit is required holes are usually finished by reaming. When holes of extreme accuracy are required it is necessary to use two reamers in order to obtain the desired results. For all such work as holes for pins, rods, or finished bolts, one reaming operation is sufficient, and even in many grades of machines the holes for journals are finished with a single machine-reaming operation. Before selecting a type of reamer for any class of work, it is advisable to study the cutting action of different reamers in order to select the one best adapted to each class.

In some shops the practice is to produce round, smooth, straight holes, as close to the standard or exact size as possible. All the tolerance or allowable error is above the basic or exact size. In other words, the hole must be slightly oversize, not

undersize. This enables holes to be produced in large numbers, which are sure to make the proper fits. Many shops insist that all holes in parts belonging to good work must be hand-reamed, and in such a manner that the work is subjected to no distortion due to holding or clamping. This is sometimes done by holding the hand reamer in a vise and passing the work over the reamer with a screwlike motion, the reamer scraping only a few thousandths out of the hole. Other shops insist that the holes be finished on the chucking machine, and in this case the chucking reamers must be accurate and the work not distorted by the chuck if good results are to be expected.

The standard reamer and the standard basic plug-and-ring gage have been the foundation of interchangeable manufacturing.

A standard basic 1-in. hole may be defined as one which will just receive a 1-in. plug gage which measures exactly 1 in. diam as measured on a measuring machine. The hole which receives this plug gage will also receive an accurately finished shaft of the same size (4).

In practical manufacturing the "go" gage is not usually exactly 1 in. diam but has a plus wear allowance and a plus manufacturing tolerance, which means that the basic hole in practical manufacturing is slightly large.

Commercial reamers are made slightly oversize because the hole made by the reamer must be large enough to receive the "go" working gage. Reamers as made commercially have also a slight plus wear allowance and a slight plus manufacturing tolerance (2). This assures that all holes carefully reamed as near to the exact size of the reamer as possible will all accept standard commercial "go" plug gages and will all receive finished shafts made to the correct sizes, with proper allowances between hole and shaft depending upon the class of fit wanted.

Many parts made by interchangeable mass-production methods depend upon the precision and uniformity of the reamer sizes to maintain hole sizes which will be basic, with a very small tolerance oversize. The holes are sure to be large enough and yet not too large to produce undesirable results.

American Standard for Allowances, Tolerances, and Gages for Metal Fits

Since 1926 the foregoing standard has been in use as a guide for the correct dimensions of holes made for interchangeable manufacturing.

This standard is based upon the same principles as the successful American Standard for Screw Threads ASA B1.1-1935. These are the same dimensions as published in Handbook H-28 Screw Thread Standards for Federal Services.

In both of these standards the minimum hole is taken as the basic or standard size. This means that the interference point or place where metal-to-metal fit occurs between hole and shaft will be at this basic dimension no matter what class of fit is specified. This is illustrated in Figs. 30 and 31.

The advantage of the standard-hole system is that it enables standard basic plug gages to be carried in stock and establishes definite minimum hole sizes, below which the holes cannot go, and maintains universal interchangeability.

If it is necessary to change tolerances or allowances on a given fit, this can be done in the basic-hole system without changing the "zero," "base," or "interference line," as the minimum hole is left constant, and the maximum hole, or minimum or maximum shaft sizes are changed. The standard plug gage always rejects any hole below basic size which might destroy interchangeability.

It is easier to produce plug gages, blocks, and measuring rods to high grades of precision, for use in measuring hole members than it is to produce gages for measuring shaft members. Hole members are best measured with fixed-size gages and also are best produced with fixed-size or solid reamers, or broaches. Shaft

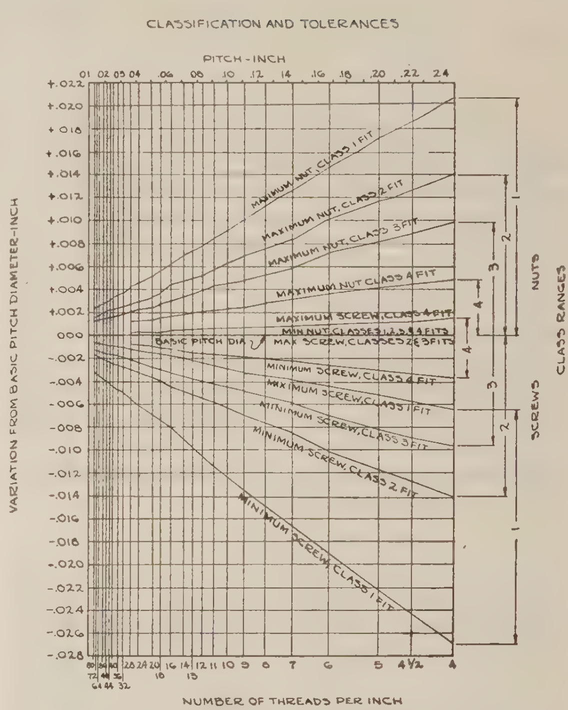

Fig. 30 Relation of Maximum and Minimum Pitch Diameters of Classes 1, 2, 3, and 4 Fits to Basic Pitch Diameters

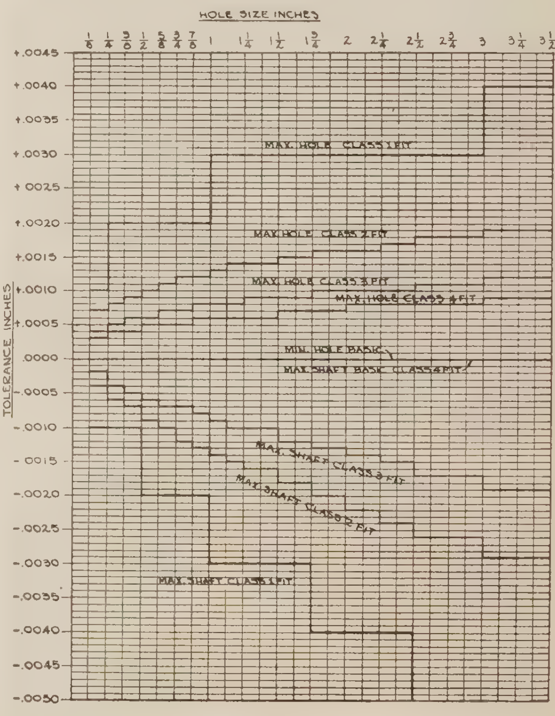

Fig. 31 Basic Minimum Hole for All Classes of Fits (American Standards Association.)

members are more easily measured with adjustable gages and also more easily produced with adjustable tools such as lathe tools or grinding wheels. Changes in allowances or tolerances for various fits can thus be more readily made on the shaft member than on the hole member.

Fig. 32 shows how a standard commercial reamer would produce holes of any class of fit desired, if the reamer were carefully used so as to cut close to its exact size. Also, if properly sharpened and cared for, a long life of the reamer may be obtained before its lands are worn so undersize that the reamer will produce holes below the minimum size.

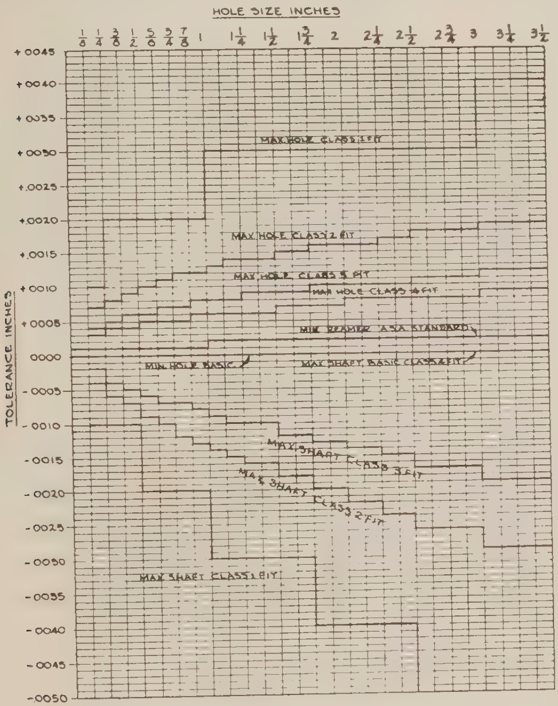

Fig. 32 Standard Reamer to Produce Standard Holes of All Classes of Fits

Another advantage of the basic-hole system of fits is that it allows one standard-size reamer with definite tolerances to be carried in stock throughout industry for the convenience of the customer and user. These reamers will produce universal interchangeable work for any of the classes of fit.

The fact that the reamer is accurately sharpened to size is no guarantee that the hole made will be as accurate as the reamer. The accuracy of the hole depends upon many factors, such as hand- or machine-reaming, machinery and fixture in good condition or badly worn, the kind of material reamed, the kind of lubricant, and the skill and carefulness of the mechanic.

It was to take care of this great variety of working conditions that different tolerances were put on the various classes of standard holes in the report of Allowances and Tolerances of the American Standards Association, ASA B4a-1925.

It was realized by the committee which wrote this report that the one standard-size reamer will produce holes of varying tolerances depending upon the conditions of its use. Extreme accuracy and precision are usually expensive, and it is uneconomical to make a hole perfect when one not so good will serve the purpose just as well. Therefore the various classes of holes and fits were provided as a measure of economy. It was intended all along, however, that but one standard size of reamer would be necessary; careful use of this reamer would produce a Class 3 or 4 hole; rough, rapid, or worn machine use would produce a Class 1 hole.

It has sometimes been thought that large tolerances were put on rough holes to allow for wear on the reamer, whereas the large tolerances are really there to indicate that costly tooling and hand-finishing are not necessary when an inexpensive rapid job is sufficient for the purpose.

Long wear life of the reamer will be obtained by keeping the point sharp and the lands in good condition rather than by using an oversize reamer which will produce holes that are too large.

It has sometimes been suggested that the American Standard for Tolerances, Allowances, and Gages for Metal Fits should contain a basic-shaft standard as well as the present basic-hole standard. It is believed that it would be better not to do this because it would imply the standardizing and carrying in stock of a different minimum or basic plug gage for each class of fit for each nominal size of hole, Fig. 33. Also it would imply the use of a different-size reamer for each class of fit. These solid tools are difficult to produce and expensive to maintain. It is more eco-

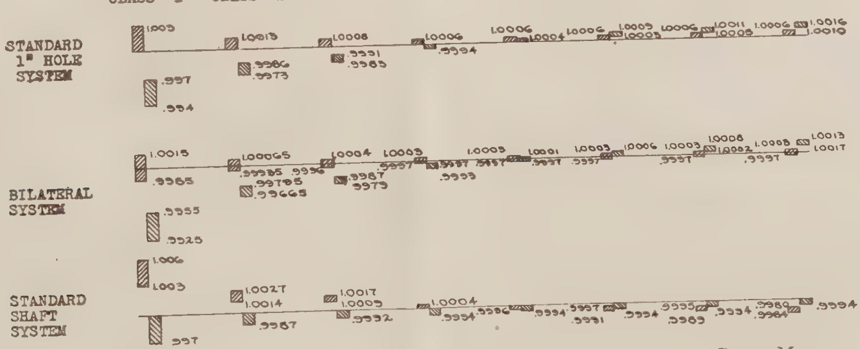

Fig. 33 Standard-Hole System Uses One Standard Basic-Size Plug Gage to Check Minimum Hole Size for All Classes of Fits

(Sketch made to show that standard-hole system uses the one standard basic-size plug gage to check minimum hole size for all classes of fits. This means that interchangeability is positively maintained for any allowance between hole and shaft and any tolerance on shaft. The "zero line," "interference line," or "metal-to-metal fit contact line" is always at the basic dimension, always at the standard minimum hole. Standard stock reamers, if carefully used and maintained sharp, will produce these standard holes for a long period of reamer-wear life.)

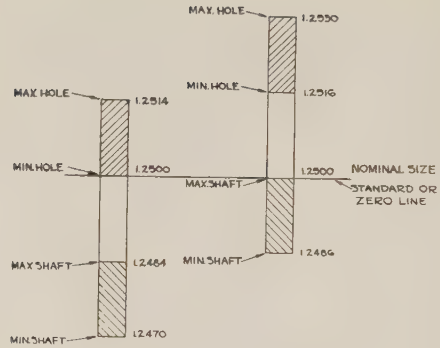

Fig. 34 Hole Sizes for Standard-Hole and Standard-Shaft Systems

nomical to obtain various classes of fit by maintaining standard holes and varying the shaft dimensions.

If for some particular reason it is found better to use the maximum size of the shaft as standard, as shown in Fig. 34, the size of hole becomes special and special drills and reamers must be obtained. Special tools are always more costly than the regular and also require longer waits for deliveries. As there is less use for a special tool, it lies in the toolroom for a longer time unused, and the overhead of manufacturing is increased.

For example, in Fig. 31 a standard shaft system of fit calls for 1.2500 as the maximum shaft size and 1.2516 as the minimum hole size. This can be considered the same as a basic-hole system of fits using a "special size" 1.2516 as the basic hole. This (as all standard shaft fits do) requires special-size reamers. From the tables of allowances and tolerances the maximum and minimum shafts and holes are easily found for special as well as standard sizes. In other words, the holes of the "standard-shaft" system, are nothing but "special" holes of the standard-hole system. The standard shaft system is really a special, nonstandard use of the standard-hole system.

In this connection E. C. Peck (1) states: "I see no need whatever for a basic shaft standard. The only example I have ever heard given is the one where two free fits are required on the ends of the shaft and a force or tight fit wanted in the center, and this has been done economically and satisfactorily for years by the 'basic-hole system.' All during World War I, I saw the drawings for millions of interchangeable parts, and I never saw any that were not made to the 'basic-hole' system."

Thus the publishing of special tables for the standard shaft system is unnecessary, as all the values for tolerances and allowances may be taken from the Standard Hole System of tables, merely by basing all the fits on the minimum-size hole.

Conclusions

A reamer is a cutting tool and like other cutting tools it must be provided with correct clearances, rakes, and other features of good tool design. Its cutting edges must be sharpened frequently to prolong the useful life of the tool and to produce accurately smoothly finished holes.

The standard reamer made with standard manufacturers' tolerance is one of the most important tools used by the interchangeable production manufacturers.

The standard reamer is an important feature of the standard-hole system of allowances and tolerances for metal fits and provides the most economical method of securing precision fits and universal interchangeability.

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Utilization of Producer Gas in Industrial Furnaces

By D. B. HENDRYX,[1] PITTSBURGH, PA.

Producer gas is the cheapest artificial fuel per Btu that can be made from solid fuel. This paper discusses the economics, the equipment, and methods of generating producer gas for industrial uses.

MOST gas-producer projects look good on paper. The basic idea of a low-cost gaseous fuel, made from coal at a central point and distributed in pipes to the furnaces, is excellent and attractive.

Producer gas for industrial uses only is discussed in this paper, as contrasted to gas manufactured for public-service distribution. It is made from coal or coke, and the solid fuel is completely transformed into gas in the process. Producer gas is the cheapest artificial fuel per Btu that can be made from solid fuel.

Comparative Cost of Fuels

The economics of the fuel situation must be considered first. The cost of fuels can be compared on a Btu basis, but this must be the total cost of heat delivered to the furnace, including fuel and ash handling, power, steam, depreciation, etc. In most industrial districts, coal is the cheapest fuel, but for various reasons it frequently cannot be used in furnaces in its crude form, either on grates or as pulverized fuel. Such considerations as furnace temperature and atmosphere, reversing and regeneration, control, delayed combustion, radiant flame, localized heating, freedom from ash, handling of coal and ashes, and many others may rule out the use of coal.

Under such conditions, if natural gas or fuel oil is available at a reasonable price, or if a by-product fuel such as coke-oven or blast-furnace gas can be used, one would hesitate before building an expensive gas-producer plant, with its attendant dirt, heat, and operating difficulties. If the fuel requirements do not amount to more than 1500 lb of coal per hr, a gas producer would seldom be justified; but if coal is cheap compared to other fuels, and if a gaseous fuel is needed, the use of producer gas will be worth investigation.

Table 1 may be used for a quick comparison between the costs of several fuels. It shows the price that can be paid for fuel oil or natural gas as compared to cost of coal delivered to the gas-producer plant, taking into account the cost of making the producer gas, including depreciation, and the cost of heating and atomizing the oil. The cost of gasifying 1 ton of coal or coke was estimated at $1.75; but this figure might be higher or lower, depending upon the size of the plant, wage rates, number of men required for operation, and similar factors (see Table 2). This table does not agree at all with the comparisons made in the advertising matter of many manufacturers, for the reason that they use a much lower charge for producer operation.

The choice of fuels cannot always be made strictly on a cost basis. The supply of natural gas or oil may be restricted or cut off suddenly in the coldest winter weather, while coal will probably always be available, in spite of numerous recent threats against the life of one of our greatest industries. The relative prices of fuels do not remain constant. Since 1939, the price of natural gas has stayed the same in most districts, while coal has advanced 30 to 45 per cent, and fuel oil 50 to 100 per cent. Any of these conditions may make the construction of a gas-producer plant imperative.

As can be seen from the cost of gasifying coal shown in Table 2, the item of depreciation is a very appreciable part of the cost. But once the money for a producer plant has been spent, due to emergency conditions or for other reasons, the matter of depreciation can be neglected, and natural gas or oil would have to be about 10 per cent cheaper than indicated in Table 1, in order to compete with producer gas.

TABLE 1 COMPARISON OF FUEL COSTS

Hot raw producer gas: Price of coal per ton delv'd. 27,000,000 Btu	Hot raw producer gas: Cost of producer gas per ton coal 19,200,000 Btu	Hot raw producer gas: Cost of producer gas per 1,000,000 Btu	Fuel oil: Equivalent price per gal, or 145,000 Btu	Natural gas: Equivalent price per 1000 cu ft or 1,000,000 Btu
$2.00	$3.75	$0.195	$0.0253	$0.195
2.50	4.25	0.221	0.0291	0.221
3.00	4.75	0.247	0.0328	0.247
3.50	5.25	0.273	0.0366	0.273
4.00	5.75	0.299	0.0404	0.299
4.50	6.25	0.325	0.0442	0.325
5.00	6.75	0.351	0.0480	0.351
5.50	7.25	0.377	0.0517	0.377
6.00	7.75	0.403	0.0555	0.403
6.50	8.25	0.430	0.0594	0.430
7.00	8.75	0.456	0.0632	0.456

Cost of gasifying coal per net ton is assumed to be $1.75 (see Table 2).

Cost of producer gas per million Btu is based on:	Per lb coal
60 cu ft of gas at 60 F per lb coal, and 140 Btu per cu ft	8400 Btu
Sensible heat with gas delivered to furnace at 700 F, 60 cu ft of gas at 11 Btu per cu ft	660 Btu
Tar, 60 cu ft of gas at 0.0006 lb tar per cu ft and 15,000 Btu per lb tar	540 Btu
Total	9600

Heat in producer gas at 700 F per ton coal, 2000 × 9600 or 19,200,000 Btu.

Price of fuel oil allows for charge of $0.003 per gal for steam, power, and depreciation but no charge for depreciation. If depreciation charge on gas-producer plant is neglected, deduct $0.0046 per gal from price of oil shown.

Price of natural gas covers fuel cost only. If depreciation charge on gas-producer plant is neglected, deduct $0.032 per 1000 cu ft from price of gas shown.

Selection of Producer-Gas Fuel

If the foregoing considerations seem to warrant the construction of a producer plant, the next step is to decide what fuel will be used to make the gas. There are two distinct types of producer gas—that made from bituminous coal, and that made from anthracite or coke. Since bituminous coal in most parts of the country is cheaper than anthracite or coke, it follows that gas manufactured from it will be the cheapest gas that can be made. Cost alone, however, may not be the deciding factor.

If the heat requirements of the furnaces are large, if the gas can be burned in the furnaces through a few ports instead of a

[1] Chief Engineer, Harbison-Walker Refractories Company.

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TABLE 2 GAS-PRODUCER OPERATING COSTS

	Small plant, 1 8-ft producer		Larger plant, 2 10-ft producers	
Fuel:	Per week	Per ton	Per week	Per ton
Tons of coal gasified	120	...	400	...
Tons of coal used to generate steam for blowing producers	8	...	25	...
Operating cost:				
Labor				
Gas producer and boiler operator 1 man 168 hr at $.80	$134.40	...	$134.40	...
Dust removal from dust legs				
Small plant 16 hr at $.65	10.40	...	...	...
Larger plant 32 hr at $.65	...	...	20.80	...
Coal and ash handling at $.10 per ton coal	12.00	...	40.00	...
Burning out flues, once every 6 weeks, averaging 1 man per week 8 hrs at $.65	5.20	...	5.20	...
Repair labor				
Small plant, 1 man 4 hr at $.90	3.60	...	...	...
Larger plant, 1 man 8 hr at $.90	...	...	7.20	...
Total labor	$165.60	$1.28	$207.60	$0.52
Power				
Small plant 1100 kwhr at $.012	13.20	0.11	...	...
Larger plant 2400 kwhr at $.012	...	...	28.80	.07
Repair materials, including coal machinery, producers, auxiliaries, boilers, painting, etc., estimated average during life of plant	30.00	0.25	60.00	0.15
Boiler feedwater and producer cooling water, based on recirculation of cooling water, 190 gal per ton coal at $.10 per M gal	2.28	0.02	7.60	0.02
Oil, grease, waste, supplies	3.00	0.03	6.00	0.02
Coal for steam, at $3.50 per ton	28.00	0.23	87.50	0.22
Total cost of gasifying coal, no depreciation charge		$1.92		$1.00
Depreciation, on basis of 10 per cent per year on investment of the following:				
Small plant, and gas mains $60,000	115.00	0.96	...	...
Larger plant, and gas mains 100,000	...	...	192.00	.48
Total, including depreciation		$2.88		$1.48

large number of individual burners, and if the producers can be located near the furnaces, the selection would favor hot raw gas made from bituminous coal. If less fuel is needed, or if gas must be distributed to a number of furnaces or individual burners located at some distance from the producer plant, the greater expense of using anthracite or coke might be warranted. Wherever practical, both kinds of gases are used in their hot raw state.

It seems to be a current belief that a "gas coal," with a volatile content above 35 per cent, is the best fuel for bituminous producers. It does make a rich gas, and it is usually a free-burning coal which does not cake badly in the producer. On the other hand, the gas may contain more vaporized tar which will be objectionable unless the gas can be kept hot until it reaches the furnaces. The ideal producer fuel would be a 2 or 3-in. lump coal, with fines below $^3/_4$ in. screened out, having an ash-fusion point above 2500 F, and free-burning characteristics which prevent the top of the fuel bed from becoming a sticky mass and caking. In actual practice, mine-run or slack coals are often used at some reduction in capacity, although more dust is carried over into the mains when using fine coal. A gummy condition at the top of the fuel bed can be overcome by keeping it at a higher temperature. The amount of ash in the coal does not make as much difference as its fusion point does, and even coal with low-fusion ash can be used by applying more steam in the blast.

Mine-run or slack bituminous coal can be used in producers because the coking action at the top of the fuel bed will provide passages for the gas. When anthracite or coke fuel is used, the only openings for the flow of gas through the upper part of the charge will be the spaces between the particles of fuel. Therefore most of the dust finer than $^1/_4$ or $^3/_8$ in. must be screened out. The term "coke breeze" can cover a multitude of sins and, before deciding upon this as a source of gas, it must be positively ascertained that the quoted price covers a fuel sufficiently free of fines to make it usable in a producer. Much of the finer coke is now used at the plants where it is made, in producers for underfiring the by-product ovens, and in steam boilers.

The Gas-Producer Plant

To take its rightful place among other fuels, producer gas must be made at the lowest possible cost, with the minimum amount of labor and operating difficulties. The gas producer is an important item, but it is only one part of the gas-making plant. The best producer cannot make gas without the proper operation and control, and it cannot deliver gas to the furnaces without a proper distribution system. There are few places where proper planning and anticipation of difficulties will have a greater effect on the success of the project than in the design of a producer-gas system. This is particularly true when using bituminous coal.

It is possible to lay out a plant so that one man per shift can take care of either one or two producers, a boiler, and the handling of coal and ashes. He might have to have occasional help in cleaning out dust legs, or in dumping a car of coal, but on the other hand, if the producer is located close enough to the furnace, he may also serve as the furnace operator.

Coal can be discharged from trucks or cars through a hopper, feeder, and bucket elevator to the coalbins. A double-roll crusher should be provided so that mine-run coal can be used even if it is planned to use sized coal. Bins above the producers can be large enough to carry a 2-day supply of coal, which is the preferable arrangement, or the feed bins can be smaller with an auxiliary storage silo. The equipment should be designed so that no hand labor is necessary except in occasional emergencies when coal must be reclaimed from an outdoor stock pile. Segregation of coarse and fine coal in the bins must be minimized. It should never be necessary for an operator to enter a coal-feed bin, as there is always a slight danger that gas may leak up into the bin and the man might be overcome.

The operating floor of a producer plant is really the second floor of the building, level with the top of the producers. All control instruments, motor starters, steam gages, water columns, valves, and air blowers for the producers should be located on this operating floor. The boiler stack can sometimes be combined with the burnout stack for the gas lines. Ashes from producers and boilers should be discharged to a skip hoist or conveyer which delivers them to an overhead ash bin so that no hand labor is necessary.

The lower floor of the building should be enclosed, as the producers do not supply enough heat to prevent freezing around the wet ashes, floors, and conveyers. Floors and pits should be drained so that a hose can be used for cleaning. The operating floor must be well ventilated and should have large doors which can be left open in summer and closed in cold weather. Louvers are usually used in the side walls just underneath the coalbins so that no gas can accumulate in the building. Oxygen apparatus should be available in case a man is overcome with gas, although this would never happen under normal operating conditions. Water-cooled poker tips have to be built up by welding every few weeks, and chain-hoist equipment must be provided for handling them.

Approximately 100 gal of water per ton of coal must be converted into steam for the producer blast, in addition to any heating or power load on the boilers, and 200 gal per hr is needed for cooling the poker, top, and coal feeder of each producer, regardless of capacity. If water is scarce or expensive, the cooling water can be recirculated to a tank on the upper floor of the building. Water-cooled valves in the gas mains have been experimented with but have not proved very dependable.

Bituminous-Type Gas Producer

Gas has been made for years with little improvement in the producing and control equipment, auxiliaries, and operating procedure. Stationary and revolving hand-poked producers gave way to mechanical types, but even today possibly 75 per cent of the producer systems in service have gas-making equipment which requires sledging of clinkers and much hard labor, delivering a gas which varies in heat value and temperature. The old distribution mains are even worse, requiring much labor and weekly burnouts. These things have given producer gas made from bituminous coal a poor reputation in industry, even though the facilities are now available for dependable operation.

The modern bituminous gas producer is a revolving vertical steel cylinder lined with refractory brick, with a stationary water-cooled top through which coal is fed, gas is removed, and pokers are operated. The bottom is sealed by a revolving water-sealed ash pan, which encloses the air-blast hood and plows for continuously removing the ashes. The air blast is usually furnished by a steam-driven turboblower, at a pressure of about 10 in. of water, and gas leaves the producer at a temperature of 1250 to 1500 F, and a pressure up to 1 in. of water.

The producer manufacturers have done a fine job in developing machines which will make good gas with almost no interruptions when operated in a proper manner. They claim, however, with a great deal of justification, that it is almost impossible to get the users to spend the money for the correct flue system and control equipment needed to make the operation clean, dependable, and free from hard labor. The manufacturers have developed a simple rugged machine requiring almost no repairs. They have designed a center blast hood to give proper distribution of air without the use of tuyères in the outer walls. They have removed the arms which formerly obstructed the flow of ashes, and have provided for continuous ash removal, both of which help to maintain an even fuel bed and to prevent the loss of carbon in the ash. They have equalized the flow of air through the bed of hot coke by providing a means for stopping the rotation of the ash pan automatically for a few seconds at a time, while the upper part of the producer continues to revolve. This causes a grinding action in the fuel bed which reduces blowholes and unequal flow of gas. Coal feeders now give excellent distribution and control of feeding rate without leakage of gas. The motion of the water-cooled pokers has been changed to give more frequent and thorough agitation of the upper part of the fuel bed without plastering the hot sticky ash against the walls of the producer. Both the poking and grinding actions are necessary to break up the caking of the coal in the fuel bed, and to prevent channels and blowholes which cause hot spots in the fuel bed and to permit the gas to burn to CO_2 in the producer.

Auxiliary and Control Equipment

The operation of a producer must be kept constant, in the face of continually changing demands for gas, and many variations in fuel and fire bed. This is almost impossible under strictly manual operation.

Gas is made by blowing a mixture of steam and air through a deep bed of incandescent carbon. If air alone were used, the fuel bed would become very hot, resulting in clinker trouble in the producer and a very lean gas.

The fire bed of the producer contains four zones as follows:

1 The ash bed at the bottom, which protects the blast hood from the heat and aids in distributing air from the hood to the fuel bed.

2 The oxidation zone, where the only reaction is the almost complete combustion of carbon to CO_2, with the liberation of 14,550 Btu per lb of carbon. This zone is only about 6 in. thick and is the hottest zone in the producer.

3 The reduction zone above the oxidation zone, where the CO_2 is reduced to CO by the hot carbon, with a corresponding endothermic reaction of 5850 Btu per pound of carbon, and the steam reacts with the carbon to form hydrogen, CO and CO_2, further reducing the temperature of the fuel bed.

4 The distillation zone at the top where the volatile matter in the coal is distilled off and moisture is evaporated.

The fuel requires 2 hr or more to pass through these reactions, but the air, steam, and gas are in contact with the fuel for little more than 1 sec in a bituminous producer, and for possibly 3 sec in the coke type. It is therefore exceedingly important that the various zones are held constant in thickness, "porosity," and temperature, and it is evident that this can only be done with equipment that has been carefully developed for the purpose, with controls made as automatic as possible, and with intelligent operation. When the load varies, these factors become increasingly important. Variations in fineness or caking qualities of the coal, and in the fusion point of the ash, affect the operation.

Air must be delivered to the blast hood in a manner which can be automatically controlled by the demand for gas, in spite of variations in the resistance of the fuel bed. Older producers usually were equipped with steam-jet blowers, but the need for fan-type blowers is now almost universally recognized, and many old producers have been equipped with modern blast equipment. When used on bituminous producers, the blowers are directly connected to small variable-speed steam turbines, and the exhaust steam is connected into the air blast. On coke and anthracite producers, the steam pressure from the water-jacketed shell is not high enough to operate a turbine, and the blower is therefore driven by electric power.

In most plants, particularly where the gas demand is not constant, the volume of the air blast is controlled by a pressure regulator connected to the gas main. The pressure in the main is usually between 0.2 and 1 in. of water. If the furnace operator turns on more gas, the pressure in the main immediately drops and the regulator feeds more steam to the turbine. The small pipe between the gas main and the regulator is kept free from tar and dust by blowing a very small amount of air through it into the main. Variations in resistance of the fuel bed will not affect the accuracy of the control, and the operator can adjust his regulator to allow for the pressure needed in the gas mains. When used with a motor-driven blower, the regulator adjusts a damper in the air line.

The volume of gas leaving a producer cannot be measured, due to the tar and dust, but the amount of air blown to the producer can be indicated and this is a measure of the gas delivered. All producers should be equipped with an orifice plate in the air-blast line, with an indicating or recording differential gage on the instrument panel to show the operator how much air is being fed to each producer. The operator will then know when the furnaces are using more gas, and he can regulate his coal feed and make other adjustments accordingly. He will also know whether each producer is carrying its share of the load. This indication of air volume is used on very few producers, but all operators who have tried it report that it is one of the most valuable instruments in the plant.

Where a producer plant is operating in connection with a single furnace, and both the producer and the furnace are under the control of one man, automatic regulation can be obtained by means of volume instead of pressure. The furnace operator will adjust the volume-control instrument to give enough gas for the furnace, and this instrument will then automatically control the speed of the turbine so that a constant volume of air is fed to the producer, regardless of the resistance against which this air must be delivered. This method has the advantage that the producer

will deliver the required amount of gas, regardless of obstructions in the line between the producer and the furnace.

The amount of steam in the air blast is one of the most important factors in producer operation. In the reduction zone in the producer, the CO_2 is reduced to CO by contact with hot carbon, with a consequent absorption of heat amounting to 5850 Btu per lb of carbon. The hotter the fuel bed, the more rapid and complete will be this reaction, resulting in less CO_2 and more CO in the gas produced. The high temperature also results in more active action between the carbon and the steam. However, the temperature which can be maintained is limited by the fusing temperature of the ash, so enough steam must be introduced to lower the temperature of the reduction zone to a safe point, by combining with the hot carbon to form H_2, CO, and some CO_2.

When the old steam-jet blowers were used, the proportion of steam to air could not be accurately controlled and too much steam was usually put into the producers. When a turbine- or motor-driven blower is used, the amount of steam is adjusted by observing the temperature of the mixture of air and steam. This is usually at some point between 130 and 150 F. When the correct figure has once been determined for a certain coal, it is not difficult for a producer operator to maintain this so-called "saturation temperature." The exhaust steam from the turbine is discharged into the air blast, and some live steam is usually added. When more air is needed the turbine is speeded up, automatically adding more steam to the blast. Automatic temperature control of the blast is often furnished but is not absolutely needed when turbines are used. It is necessary when blowers are motor-driven.

Fig. 1 shows the blast temperatures corresponding to various inlet-air temperatures and pounds of steam per pound of coal. Fig. 2 shows results of tests reported by W. P. Chandler of Carnegie-Illinois Steel Corporation indicating the increase in the quality of the gas when the proportion of steam in the blast was decreased. In practice, however, it is usually dollar efficiency rather than thermal efficiency which counts, and as coals with high ash-fusion temperatures are rarely available, we choose more steam and a poorer gas rather than a job of sledging clinkers out of the producers.

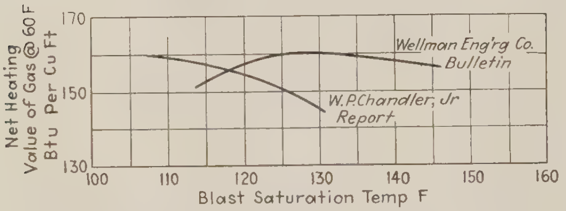


Fig. 1 Relation Between Steam Used per Pound of Coal and Blast Saturation Temperature

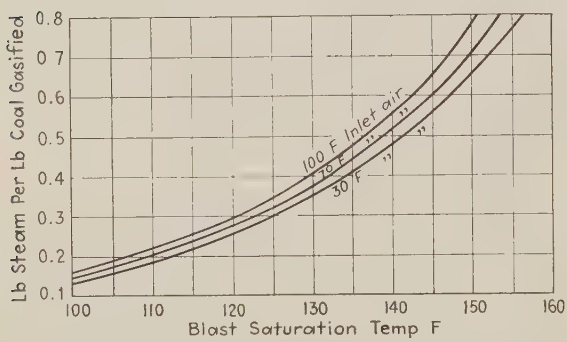


Fig. 2 Relation Between Heating Value of Gas and Blast Saturation Temperature

Producer gas is a very "lean" gas, due to the high percentage of inert nitrogen introduced with the oxygen in the blast. The use of steam as a reducing agent to lower the temperature of the fuel bed and prevent the formation of clinkers, actually enriches the gas because it adds hydrogen without increasing the amount of inert gases. However, fuel must be burned to generate the steam, and this fuel amounts to about 5 per cent of the coal gasified. Therefore, CO_2 is sometimes used in the blast, to lower the fuel-bed temperature, but this is only profitable where the CO_2 content of the gases is very high, as in the exhaust gases from lime kilns. Proper control of the percentage of CO_2 added to the blast is quite difficult.

High fuel-bed temperatures do not necessarily mean high gas temperatures leaving the producers. Gas temperatures usually run between 1250 and 1500 F, but may be as high as 1800 to 1900 F when used in kilns for burning limestone, where producers are usually located very close to the kilns.

All automatic producers have an adjustable mechanical coal feed. Practically all of these are adjusted by hand. There is a recording thermocouple in the gas stream leaving the producer, and the operator adjusts the coal feed to maintain a reasonably constant gas temperature. If more air is blown through the producer, the top of the fuel bed soon increases in temperature, and the operator feeds more coal to prevent the gas from getting too hot. If the operator is provided with an orifice meter on the air blast, he knows when more gas is being generated without waiting for the temperature to increase, and he can regulate his coal feed accordingly. Automatic regulation of coal feed is now being applied, using the temperature of the gas leaving the producer to regulate the rate of feed. This is very successful, as it re-

TABLE 3 PRODUCER-GAS DATA

Fuel used in producer	Bituminous coal	Anthracite coal	Coke breeze
Fuel gasified, lb per sq ft per hr	30–70	15–30	15–30
Volume of gas at 60 F, cu ft per lb coal	60	65	68
Temperature of gas leaving producer, deg F	1250–1500	550–650	550–650
Lower or net heating values, Btu:			
Coal or coke used, per lb	13500	12500	12000
Steam 0.4 lb per lb fuel	450	(450)	(450)
(Anthracite and coke producers make own steam)			
Heat input to producer, per lb fuel, Btu	13950	12500	12000
Raw producer gas, per cu ft at 60 F	140	130	120
Sensible heat of gas leaving producer per cu ft	23	10	10
Tar in hot gas leaving producer, 0.0006 lb tar per cu ft gas at 15,000 Btu	9	..	..
Heat output from producer, per cu ft, Btu	172	140	130
Heat output from producer, per lb fuel, Btu	10320	9100	8840
Efficiency of producer, not including losses in boiler or gas lines	73.9	72.7	73.6
Typical analysis of gas, per cent:			
CO	22.5	24.8	26.0
H_2	12.5	15.0	11.0
Hydrocarbons	3.1	0.9	0.7
CO_2	7.4	9.0	7.6
N_2	54.5	50.3	54.7
	100.0	100.0	100.0
Air for perfect combustion, cu ft per cu ft gas	1.17	1.04	0.95
Additional air for combustion of 0.0006 lb tar per cu ft of bituminous gas	0.11	...	...
	1.28		
Products of perfect combustion, cu ft per cu ft gas	2.14	1.83	1.76
Weight of producer gas, lb per cu ft	0.069	0.066	0.069
Weight of air for perfect combustion, lb per cu ft of gas	0.098	0.080	0.073
Weight of products of perfect combustion lb per cu ft of producer gas	0.167	0.146	0.142
Theoretical flame temperature, based upon gas at 60 F, deg F	3170	3150	3000

duces the work of the operator and improves the quality of the gas.

Table 3 gives general information on the different types of producer gas and is useful in the design of distribution systems, burners, and furnaces, and in the comparison of this gas with other fuel.

Distribution System for Bituminous Producer Gas

Raw gas made from bituminous coal is a hot dirty fuel to handle. It contains dust from the fine coal falling into the producer and soot from the cracking of the hydrocarbons. When using slack coal, the dust carried over into the mains may amount to 5 per cent of the weight of the coal. This will fill up a round pipe so rapidly that the flue would have to be cleaned or "burned out" once a week, shutting down the furnaces and producers during the operation.

This gas also contains tar vapors which will condense as the gas cools, gumming up valves and burners, seeping out of cleanout doors, and mixing with the dust to form a cokelike substance. It has a temperature of 1250 to 1500 F, when it leaves the producer, and therefore has nearly twice the volume of gas made from coke or anthracite and 18 times the volume of natural gas for the same Btu value. Consequently, mains and valves are large and costly. As a result, bituminous gas has been most frequently applied to large furnaces located near the producer plant and more rarely used where the gas must be distributed for greater distances and to a large number of burners.

Having outlined the causes of the difficulties encountered in the distribution of hot raw bituminous gas, the next step is to see what can be done to overcome them. In the case of old gas mains already in use, only a little improvement is possible without extensive alterations. Where a new distribution system is to be installed, it is entirely possible to increase the length of time between "burnouts" to 2 months, or even to a year, at the same time conserving the sensible heat of the gas and smoothing out the furnace operation.

Gas mains can be built without any horizontal bottom surfaces on which dust can accumulate. They can be constructed with a saw-tooth bottom, with provision for removing the dust at all of the low points. Dust will sometimes accumulate on the slopes but small openings can be provided for steam lances in the top of the pipe, so that the dust can be blown down into the pockets at the bottom. However, this is a hot and dirty job, and a better scheme is to install soot blowers at every point in the mains where dust can settle, with legs or pockets having double seals through which the dust can be removed while gas is flowing. These blowers can be standard boiler soot blowers, which will withstand the gas temperatures quite well, but are not entirely resistant to the higher burnout temperatures. On the other hand, if burnouts can be greatly reduced or eliminated by the soot blowers, the soot blowers will not be damaged by the burnouts.

The removal of the dust from the pockets is also a disagreeable job, particularly if it must be handled in wheelbarrows. This dust could be sluiced into a stream of water, but this is rarely available. A steam-ejector vacuum system is being used to remove the dust and deliver it in a dampened condition into the ash bin. This dust contains about 75 per cent carbon and 25 per cent ash and is "red hot" when removed from the mains.

To prevent trouble with the tar, the gas temperature must be kept high enough so that the vapors will not condense, or else the gas must be cooled and washed. Insulation of new gas lines is now relatively simple since insulating refractory brick can be used for the complete lining of the mains, and gas can be delivered 100 ft from the producer plant with a temperature loss of not more than 300 deg F, including dust catchers and mains. Washing of bituminous gas has been tried on numerous installations in the past, and there is at least one successful plant in operation today. However, it involves the use of expensive equipment and the loss of sensible heat amounting to 6 to 12 per cent of the total Btu content of the hot gas as well as the loss of the tar. Either the tar must be wasted, or costly recovery and handling equipment must be installed.

Producers and Distribution Systems for Anthracite and Coke Gas

Gas made from coke or anthracite, while higher in first cost, is cooler, cleaner, and easier to handle, and may be washed if necessary with less difficulty and loss of heat. It is comparatively free from dust as it leaves the producer because anthracite producers gasify only one half as much fuel as do the bituminous type, and because the fuel used contains less fines and almost no hydrocarbons, and flows directly from distributing pipes onto the top of the fuel bed instead of dropping through a stream of moving gases. It is entirely free from tar and has a temperature of about 600 F, making it possible to deliver it through steel pipe lines of smaller size, with insulation on the outside of the pipe. No time need be lost in burning out the mains. Higher pressures and velocities can be used and a booster blower can even be connected to increase the pressure of the gas after it leaves the producers. In the past this gas has usually been washed, but in recent installations the velocity has been increased by the use of bowers to the point where dust will not settle out, and gases are distributed in the hot raw condition. Standard valves, fittings, and burners can be used, and the burners require much less attention from the furnace operators.

The producers used for coke and anthracite are quite different from bituminous producers, mainly because these fuels do not contain tar and hydrocarbons. Therefore the fuel bed does not cake to any great extent and no mechanical poker is needed. The bed can be 5 ft deep instead of about 2.5 ft, resulting in cooler gas. The construction is very much simpler than in the bituminous machine, since the shell is stationary, while only the bottom revolves. The blast hood is large and is usually eccentric in shape, so that it grinds the ashes and agitates the fuel bed as it revolves. In some producers it is attached to the revolving ash pan in which case a deep water seal is used and the blast pressure is limited to approximately 20 in. of water. In others, the revolving blast hood supports the fuel bed, and ashes are discharged through the hood into a stationary hopper below. These producers can be operated with a blast pressure of several pounds, since no water seal is used. Coal feeders are either stationary or revolving. The producer shell is water-jacketed instead of being lined with brick and generates enough steam to mix with the air blast.

Refractories

The refractory-brick lining of a bituminous producer is subjected to continuous abrasion from the passage of fuel and ashes, and to occasional destruction from the formation of clinkers and the necessity of breaking them loose from the walls with heavy bars. The temperature of the lining usually is not excessive, but hot spots sometimes develop, due to channeling of the fires. A dense, strong, high-heat-duty brick is required to resist these conditions. The lining can be either $4^1/_2$ in. or 6 in. thick, backed up with $2^1/_2$ in. of strong insulation next to the shell. A high-temperature cold-setting refractory mortar should be used, and all joints should be kept as tight as possible. When using coal having a very low ash-fusion temperature a few courses of silicon-carbide brick are sometimes used at the clinker line.

Mains for carrying hot raw bituminous gas do not exceed 1400 F except during burnouts. Steam jets and lances must be used to stir up the dust. Formerly, mains were lined with $4^1/_2$ in. of

fireclay brick, sometimes surrounded with insulation. The development of lightweight refractory brick which will withstand these conditions has greatly simplified the construction of these mains. The entire 9-in. lining can be built of 2600-deg insulating refractories, or lower-temperature insulating brick can be used in the outer $4^1/_2$-in. portion. This brick is somewhat lower in cost and higher in insulating value. Flue linings usually involve a great deal of complicated brickwork and the use of insulating refractories which can be easily cut, sawed, ground, or rubbed into any desired shape greatly reduces the bricklaying labor cost. A cold-setting mortar should be used, with expansion joints about every 20 ft. No fireclay brick need be used except in the connection from the producer to the dust catcher, and for construction of the valve seats. Cast-iron seats are now seldom used.

Furnaces and Burners

Producer gas is not only a cheap fuel, but is admirably adapted to many heating operations, particularly where a soft luminous flame is desired. As can be seen from Table 3, it has a theoretical flame temperature of about 3100 F, as compared to 3700 F, for natural gas and 3800 F for oil. This makes it especially suited to steel reheating, where the entire furnace must be filled with a hazy slightly reducing flame and "hot spots" must be avoided. The same characteristics, at slightly higher temperatures, make it suitable for burning refractory brick. A glass tank can be completely filled with a luminous flame having the high radiation characteristics so desirable for heating this transparent molten bath. With producer gas, it is easy to get delayed combustion, even when using highly preheated air, an operation that is very difficult when using natural gas or oil.

Bituminous producer gas is used for firing vertical-shaft lime kilns and also rotary kilns for lime and other products where freedom from ash deposits is necessary. Freedom from ash makes it suitable for firing many ceramic products, where both bituminous and coke or anthracite gas are being used.

In many of these furnaces, the fuel requirements are large, and the gas and air are merely admitted through ports in the furnace walls. The air is usually preheated by regeneration or other means, and this preheating becomes very necessary when high temperatures must be reached. It is sometimes claimed that there is no object in keeping the gas hot between the producers and the furnaces, as this may reduce the furnace efficiency because the exhaust gases will not be cooled in the gas regenerators. Intermittent steam jets can be used if necessary in order to keep soot from building up on the reversing valves and burner ports.

Where a number of smaller burners have to be used, bituminous gas is usually fired through a cast-iron burner with a fireclay burner block. The gas flow is controlled by a slide or inverted-cone valve, which has to be opened and closed about once an hour to keep it clean. The air, preferably preheated, often acts as an injector to draw the gas into the burner block. A very short oxidizing flame can be obtained by using enough air and moving the air pipe away from the block. A reducing or delayed combustion flame is produced by using less air and moving the pipe closer to the throat of the burner. These burners must be thoroughly cleaned about once every 24 hr and large cleanout doors must be provided. Since the gas going to each burner cannot be measured, it is advisable to have an orifice and an indicating differential gage in each air line. The air can then be set for any desired quantity, and the gas adjusted to give the required temperature and flame quality.

Producer gas made from anthracite or coke, either washed or in the hot raw condition, can be burned in standard gas burners large enough to allow for the low Btu value of the gas, or it can be regenerated and burned in ports. It is particularly suited to plants where the burners are widely scattered, as in smaller heating furnaces, and in the underfiring of coke ovens.

Everything considered, producer gas made from bituminous coal, anthracite, or coke can be a very desirable fuel if proper consideration is given to the economic, engineering, and operating factors which so materially affect the success of the project.

Pulsation and Its Effect on Flowmeters[1]

By E. J. LINDAHL,[2] COLUMBUS, OHIO

Flowmeter readings are apt to be seriously in error if the fluid flowing through the meter is subjected to rapid recurring variations in velocity or pressure or both. Flow having these variations is thought of as being "pulsating flow." Much thought has been given to pulsating flow or pulsation and much has been written about it. It is the intention of this paper to bring together some of the results which have been published.

Causes and Effects of Pulsation

STEADY flow of a medium exists when the mass rate of flow through a device is constant, when the velocity, pressure, and temperature at any point remain constant, and when thermal equilibrium has been reached by the device through which the medium is flowing. In the case of unsteady flow, one or more of the conditions mentioned does not exist. Pulsating flow can exist when rapid velocity or pressure variations occur and is seen to be a special case of unsteady flow.

The causes of pulsation are numerous. One of the chief sources of pulsation in gas pipe lines is the reciprocating compressor. Another source is water surging back and forth in a low portion of a pipe line. Air enters the intake manifold of an internal-combustion engine having pulsation due to the pumping action of the engine pistons. Clattering check valves and undamped diaphragm regulators in a line are also offenders. Steam flowing to a reciprocating engine has pulsating flow. Any device that causes intermittent flows or pressures is a source of pulsation, since pulsations take place as a result of recurring rapid variations in either velocity or pressure.

One device which is greatly affected by pulsation and which concerns engineers is the inferential meter. While it is true that there are some pulsations which have no effect on the meter reading, there are other pulsations which render the meter reading invalid. Since the measurements which involve large quantities are made with flowmeters, it is extremely important from a dollars and cents standpoint that these meters read accurately. As a result, a great deal of time and money have been spent by concerns in the gas industry to determine means for reducing the errors in meter readings resulting from pulsation. This paper will deal chiefly with the effect of pulsation on meters. In passing, it might be said that at the present time very little study has been made to determine errors in the measurement of liquids having pulsating flow.

It is now generally accepted that any inferential-head meter will measure correctly if the secondary device will measure accurately the differential pressure across the primary element, and if the average of the square roots of the instantaneous readings can be determined. In the case of steady flow these operations are comparatively simple, but when the flow is at all unsteady this is not true.

If the differential is irregular as a result of rapid variations in velocity or pressure the meter is inaccurate because of the fact that most secondary devices are too sluggish to follow the actual changes in differential pressure caused by rapidly changing flow. Furthermore, the differential pressure, indicated by the secondary device, is an approximate average of the maximum and minimum differential pressures. When the square root of this average is taken, the resulting value is higher than the average of the square roots of the instantaneous differentials, and the flow as determined from the meter will be too high.

This can be illustrated by assuming that the flow through a meter varies in a manner such that the differential is, for example, 49 in. during one second and 25 in. the next second. The average rate of flow over this period is proportional to $\frac{(49)^{1/2} + (25)^{1/2}}{2}$ or 6; if the average meter differential is 37, the rate of flow as registered by the meter is proportional to $(37)^{1/2}$ or 6.083. The error due to pulsation is $(6.083 - 6.000)/6.000$ or 1.4 per cent.

According to most investigators the error is always positive, but cases of negative error have been found which have been attributed by some engineers to pulsation. The magnitude of the error may be up in the hundreds of per cent.

Mathematical Analysis of Pulsation Error

In order to show how pulsation acts to give erroneous meter readings a mathematical analysis of three types of waves will be made. It must be kept in mind that the method of calculating the error due to pulsation is only theoretical and no attempt should be made to apply it in a practical case. It will, however, indicate why pulsation error exists.

Three wave forms, i.e., the rectangular, the sine, and the triangular, will be analyzed. Two assumptions in regard to meter behavior are made in these analyses: (*a*) it is assumed that the pulsations occur with such rapidity that the portion of the meter upon which the differential is impressed cannot follow the rapidly fluctuating differential pressures; and (*b*) it is assumed that the wave is symmetrical and that the meter gives a reading which is the mean between the upper and lower limits of the pulsation wave.

The sine wave would result if a single-cylinder compressor had the connecting rod replaced by a Scotch yoke, as shown in Fig. 1(*a*), since the piston would have simple harmonic motion and the wave displacement assumed to follow the motion of the piston. The wave would then be as shown in Fig. 1(*b*), with p as the maximum displacement of the pulsation wave and with the angular displacement of the crankpin through one revolution plotted as the abscissa.

The rectangular wave form will be first discussed in detail, since its analysis presents less of a problem than the other two.

The differentials encountered are illustrated in Fig. 2(*a*) and apply to one revolution of the crank of a compressor. The phase angle of the crank is plotted as the abscissa and the instantaneous differential as the ordinate. The meter reading (indicated by h) is taken halfway between the maximum differential $(h + p)$ due to the pulsation and $(h - p)$ the minimum differential, p being the distance between the instantaneous differential and the meter reading. The instantaneous value of the true differential is indicated by H. If K is used to designate all of the constants, such as $(2g)^{1/2}$, velocity of approach factor, etc., then Q_m, the flow as indicated by the meter, is equal to $Kh^{1/2}$. If, instead of

[1] Thesis submitted to the Department of Mechanical Engineering and the Committee on Graduate Study and Research at the University of Wyoming in partial fulfillment of the requirements for the degree of Mechanical Engineer. Laramie, Wyoming, 1946.

[2] Assistant Professor of Mechanical Engineering, The Ohio State University. Mem. A.S.M.E

Contributed by the Research Committee on Fluid Meters and presented at the Semi-Annual Meeting, Detroit, Mich., June 17–20, 1946, of The American Society of Mechanical Engineers.

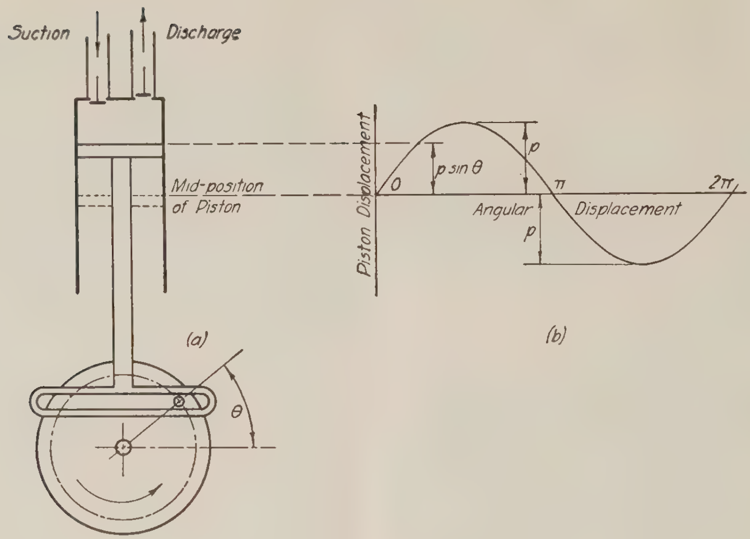


Fig. 1 Generation of Pulsation Wave

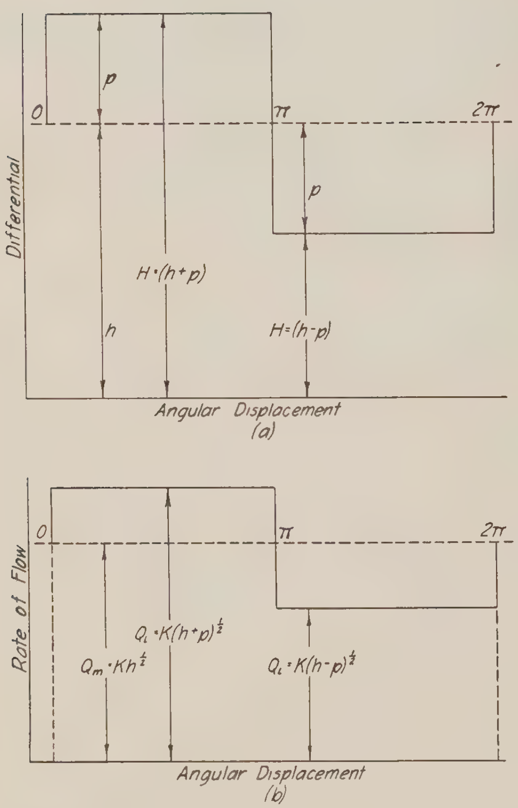


Fig. 2 Rectangular Wave

plotting the instantaneous differential, the instantaneous flow is plotted, the result is as shown in Fig. 2(*b*). If the area under the curve is taken for one cycle (angular displacement 0 to 2π), and the area is divided by the length of the cycle, there results the average ordinate which is the average rate of flow and

$$Q_a = \frac{K\,(h+p)^{1/2}\,\pi + K\,(h-p)^{1/2}\,\pi}{2\pi}$$

or

$$Q_a = \frac{K}{2}\,[(h+p)^{1/2} + (h-p)^{1/2}]$$

Then the ratio of the flow as determined by the meter to the actual rate of flow is

$$\frac{Q_m}{Q_a} = \frac{Kh^{1/2}}{\frac{K}{2}\,[(h+p)^{1/2} + (h-p)^{1/2}]}$$

or

$$\frac{Q_m}{Q_a} = \frac{2h^{1/2}}{(h+p)^{1/2} + (h-p)^{1/2}}$$

Sine Wave. As has been pointed out, a sine wave as shown in Fig. 3(*a*) will result theoretically if the connecting rod of a single-cylinder compressor is replaced by a Scotch yoke. Any distance H from the datum to the wave represents an instantaneous value of the true differential which for this particular wave form is $(h + p \sin \theta)$, while the distance h to the horizontal dashed line represents the differential as indicated by the meter. If K is again assumed to be the meter constant, the rate of flow as indicated by the meter is again $Q_m = Kh^{1/2}$, and the instantaneous true rate of flow is

$$Q_i = K(H)^{1/2} \text{ or } Q_i = K\,(h + p \sin \theta)^{1/2}$$

If the instantaneous true rates of flow are plotted the result is Fig. 3(*b*). In order to obtain the true average rate of flow it will again be necessary to determine the area under one cycle of the curve and divide this area by the length of one cycle. The flow which takes place while the crank turns through the angle $d\theta$ is

$$K\,(h + p \sin \theta)^{1/2}\,d\theta$$

and the total flow from one revolution of the crank is

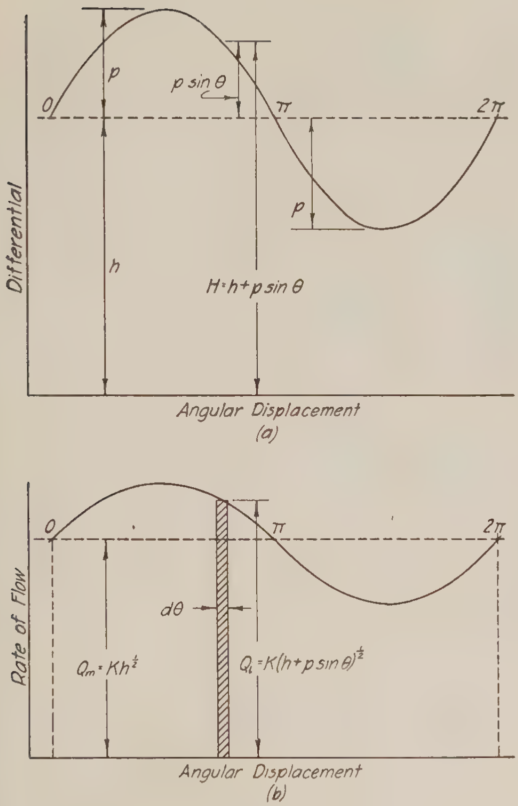


FIG. 3 SINE WAVE

$$K\int_0^{2\pi} (h + p \sin \theta)^{1/2}\, d\theta$$

The average flow over this cycle then is

$$Q_a = \frac{K}{2\pi}\int_0^{2\pi} (h + p \sin \theta)^{1/2}\, d\theta$$

If it is desired to find the actual discharge for pulsating flow from the foregoing equation, the equation must, of course, be integrated. This is done after the portion $(h + p \sin \theta)^{1/2}$ has been expanded by the binomial theorem. The binomial theorem in terms of symbols for five terms is

$$(a + b)^n = a^n + na^{n-1}b + \frac{n(n-1)}{2!}a^{n-2}b^2 + \frac{n(n-1)(n-2)}{3!}a^{n-3}b^3 + \frac{n(n-1)\,(n-2)\,(n-3)}{4!}a^{n-4}b^4$$

Then

$$(h + p \sin \theta)^{1/2} = h^{1/2} + \frac{p \sin \theta}{2h^{1/2}} - \frac{1}{8h^{3/2}}p^2 \sin^2 \theta + \frac{3}{48h^{5/2}}p^3 \sin^3 \theta - \frac{15}{384h^{7/2}}p^4 \sin^4 \theta = h^{1/2}\left[1 + \frac{p \sin \theta}{2h} - \frac{p^2 \sin^2 \theta}{8h^2} + \frac{p^3 \sin^3 \theta}{16h^3} - \frac{15\, p^4 \sin^4 \theta}{384h^4}\right]$$

and

$$Q_a = \frac{Kh^{1/2}}{2\pi}\int_0^{2\pi}\left[1 + \frac{p \sin \theta}{2h} - \frac{p^2 \sin^2 \theta}{8h^2} + \frac{p^3 \sin^3 \theta}{16h^3} - \frac{15\, p^4 \sin^4 \theta}{384h^4}\right] d\theta$$

or

$$Q_a = \frac{Kh^{1/2}}{2\pi}\left[\int_0^{2\pi} d\theta + \frac{p}{2h}\int_0^{2\pi} \sin \theta d\theta - \frac{p^2}{8h^2}\int_0^{2\pi} \sin^2 \theta d\theta + \frac{p^3}{16h^3}\int_0^{2\pi} \sin^3 \theta d\theta - \frac{15}{384}\frac{p^4}{h^4}\int_0^{2\pi} \sin^4 \theta d\theta\right]$$

Since

$$\int d\theta = \theta + c$$

$$\int \sin \theta\, d\theta = -\cos \theta + c$$

$$\int \sin^2 \theta\, d\theta = -\frac{1}{2}\cos \theta \sin \theta + \frac{1}{2}\theta + c$$

$$\int \sin^3 \theta\, d\theta = -\frac{1}{3}\cos \theta \sin^2 \theta - \frac{2}{3}\cos \theta + c$$

and

$$\int \sin^4 \theta d\theta = \frac{-\sin^3 \theta \cos \theta}{4} + \frac{3}{4}\left(-\frac{1}{2}\cos \theta \sin \theta + \frac{1}{2}\theta\right) + c$$

then the expression for discharge, when integrated is

$$Q_a = \frac{Kh^{1/2}}{2\pi}\left\{\theta - \frac{p}{2h}\cos \theta - \frac{p^2}{8h^2}\left[-\frac{1}{2}\cos \theta \sin \theta + \frac{1}{2}\theta\right] + \frac{p^3}{16h^3}\left[\frac{1}{3}\cos \theta \sin^2 \theta - \frac{2}{3}\cos \theta\right] - \frac{15}{384}\frac{p^4}{h^4}\left[-\frac{\sin^3 \theta \cos \theta}{4} + \frac{3}{4}\left(-\frac{1}{2}\cos \theta \sin \theta + \frac{1}{2}\right)\right]\right\}_0^{2\pi}$$

Since $\sin 2\pi = 0$, $\sin 0 = 0$, $\cos 2\pi = 1$, and $\cos 0 = 1$ when the limits have been substituted in the foregoing equation, the result is

$$Q_a = \frac{Kh^{1/2}}{2\pi}\left\{\left[2\pi\right] - \left[\frac{p}{2h}\right] - \left[\frac{p^2}{8h^2}\pi\right] + \left[\frac{p^3}{16h^3} \times \frac{-2}{3}\right] - \left[\frac{15}{384}\frac{p^4}{h^4} \times \frac{3}{4}\pi\right] - \left[\left(\frac{-p}{2h}\right) - \left(\frac{p^3}{16h^3} \times \frac{2}{3}\right)\right]\right\}$$

and

$$Q_a = \frac{Kh^{1/2}}{2\pi}\left[1 - \frac{p^2\pi}{8h^2} - \frac{15p^4\pi}{512h^4}\right]$$

or

$$Q_a = Kh^{1/2}\left[1 - \frac{p^2}{16h^2} - \frac{15}{1024}\frac{p^4}{h^4}\right]$$

Then the ratio of the rate of flow as indicated by the meter to the actual rate of flow when the pulsation is in the form of a sine wave, is

$$\frac{Q_m}{Q_a} = \frac{Kh^{1/2}}{Kh^{1/2}\left[1 - \dfrac{p^2}{16h^2} - \dfrac{15}{1024}\dfrac{p^4}{h^4}\right]}$$

or

$$\frac{Q_m}{Q_a} = \frac{1}{1 - \dfrac{p^2}{16h^2} - \dfrac{15}{1024}\dfrac{p^4}{h^4}}$$

It will be observed that if the amplitude of the pulsation p is zero, the flow as indicated by the meter is equal to the true rate of flow. If the meter differential and the amplitude of the pulsation are known the ratio of the two can be found and the percentage of error in the meter reading determined from the expression $\dfrac{Q_m - Q_a}{Q_a}$ which is equal to $\dfrac{Q_m}{Q_a} - 1$.

Triangular Wave Form. To take another hypothetical case, assume the wave form to be triangular as shown in Fig. 4(a). The reading of the meter is, as before, $K(h)^{1/2}$. The value of the instantaneous differential H in each case is as follows

$$\text{From } 0 \text{ to } \frac{\pi}{2}: \quad H = h + \frac{2p\theta}{\pi}$$

$$\frac{\pi}{2} \text{ to } \frac{3}{2}\pi: \quad H = h + 2p\left(1 - \frac{\theta}{\pi}\right)$$

$$\frac{3}{2}\pi \text{ to } 2\pi: \quad H = h - 2p\left(2 - \frac{\theta}{\pi}\right)$$

When the instantaneous rate of flow is plotted the curve takes the form as indicated in Fig. 4(b) and the rate of flow is as noted in the figure.

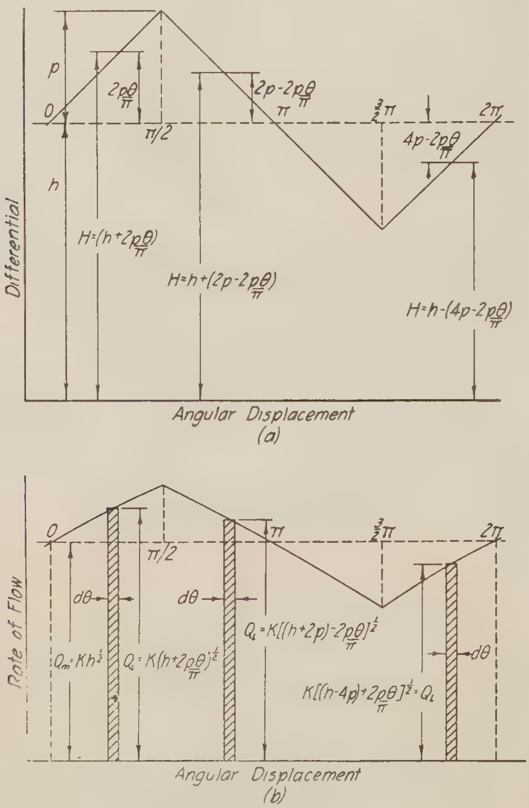


Fig. 4 Triangular Wave

The flow during the period that the crank turns through a complete revolution is

$$Q_a = \frac{K}{2\pi}\left\{\int_0^{\frac{\pi}{2}}\left[h + \frac{2p\theta}{\pi}\right]^{1/2} d\theta + \int_{\frac{\pi}{2}}^{\frac{3\pi}{2}}\left[(h + 2p) - \frac{2p\theta}{\pi}\right]^{1/2} d\theta + \int_{\frac{3\pi}{2}}^{2\pi}\left[(h - 4p) + \frac{2p\theta}{\pi}\right]^{1/2} d\theta\right\}$$

This equation is of the form $(a + bu)^n du$ which when integrated is $\dfrac{(a + bu)^{n+1}}{b(n + 1)}$. Since in this case $n = 1/2$ the form when integrated is $\dfrac{(a + bu)^{3/2}}{3/2\, b}$.

Then

$$Q_a = \frac{K}{2\pi}\left\{\left[\frac{\left(h + \frac{2p\theta}{\pi}\right)^{3/2}}{3/2 \times \frac{2p}{\pi}}\right]_0^{\frac{\pi}{2}} + \left[\frac{\left(h + 2p - \frac{2p\theta}{\pi}\right)^{3/2}}{-3/2 \times \frac{2p}{\pi}}\right]_{\frac{\pi}{2}}^{\frac{3}{2}\pi} + \left[\frac{\left(h - 4p + \frac{2p\theta}{\pi}\right)^{3/2}}{3/2 \times \frac{2p}{\pi}}\right]_{\frac{3}{2}\pi}^{2\pi}\right\}$$

and

$$Q_a = \frac{K}{2\pi}\left\{\left[\frac{(h + p)^{3/2} - h^{3/2}}{\frac{3p}{\pi}}\right] - \left[\frac{(h - p)^{3/2} - (h + p)^{3/2}}{\frac{3p}{\pi}}\right] + \left[\frac{h^{3/2} - (h - p)^{3/2}}{\frac{3p}{\pi}}\right]\right\}$$

and finally

$$Q_a = \frac{K}{3p}\left[(h + p)^{3/2} - (h - p)^{3/2}\right]$$

Then

$$\frac{Q_m}{Q_a} = \frac{Kh^{1/2}}{\dfrac{K}{3p}\left[(h + p)^{3/2} - (h - p)^{3/2}\right]}$$

or

$$\frac{Q_m}{Q_a} = \frac{3ph^{1/2}}{(h + p)^{3/2} - (h - p)^{3/2}}$$

The Theory of Pulsation

Numerous articles dealing with pulsation and the effects of pulsating flow have appeared in technical literature from time to time. Most of them deal with the inaccuracy of meters handling fluids subject to pulsation.

Among the first references is one by William Mayo Venable (1)[3] in 1905, which stated that piezometer tubes used in measuring the static head on centrifugal pumps gave values that were abnormally high. He attributed the error to pulsation caused by the limited number of vanes or blades in the pump runner.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

In the 1914–1915 Proceedings of The Institution of Civil Engineers Prof. A. H. Gibson (2) pointed out that "most devices for measuring fluid motion do not record the true mean of a pulsating flow and that the error increases," and that "fluctuations in velocity (are) accompanied by fluctuations in pressure."

In 1916 three interesting and significant articles appeared pointing out the inaccuracy of meters when measuring fluids having pulsating flow. One by Allen Hazen (3) indicated that wide and sudden variations in flow cause errors in the head produced by a Venturi meter and that elastic pipe and too-small air chambers when used with reciprocating pumps cause these errors. Another by E. G. Bailey (4) stated that steam flowing to a reciprocating engine is erroneously measured by a flowmeter because of velocity and pressure variations. F. P. Fisher (5) attributed to pulsation the discrepancy in meter readings when the meters were placed some distance apart in a pipe line from a compressor.

In 1922 the results of an investigation by Judd and Pheley (6) appeared in a bulletin published by The Ohio State University Experiment Station. This was the first real attempt to determine the nature of pulsation and to study methods of eliminating pulsation itself and the detrimental effects of pulsation. The conclusions drawn by Judd and Pheley are still considered sound; in fact, most of the later theorizing and investigating has been based upon the ideas originally brought out by them. Subsequent investigations have served to substantiate their conclusions.

Their studies were made with air on Venturi meters, orifices, nozzles, and Pitot tubes. Among the pertinent conclusions which these two men drew up are the following:

(*a*) Pulsations in pipe lines consist of sudden changes in velocity and pressure of the fluid.

(*b*) Pressure changes cause the largest pulsations.

(*c*) The pressure change is in the form of a wave front resembling a traveling sound wave of low velocity.

(*d*) The pressure wave travels with the velocity of sound.

(*e*) Velocity of pulsation is independent of the velocity of fluid flowing.

(*f*) The effect of pulsation on a flowmeter is to increase the reading. The magnitude of the error is a function of frequency and pulsation, static pressure of the fluid, type of meter, and adjacent fixture in the pipe line.

(*g*) Pulsation must be eliminated or reduced greatly to have the meter read without appreciable error.

It was also pointed out by them and now generally believed that no correction factors can be applied to a meter reading to correct for the error caused by pulsation. In most of the correction equations several things, such as frequency of pulsation, the shape of the wave, the amplitude of the wave, etc., must be taken into account. It is also felt that errors due to rapid pressure change are far more severe than changes in velocity. The fact is that there can be errors caused by rapid pressure changes without the presence of changes in the rate of flow of the mass in the pipe line. This has been illustrated by placing a positive-displacement meter on a dead-ended line. The meter valves will clatter and the meter will register a flow if pulsation is present. Likewise, a reading will be indicated on a manometer across an inferential meter, such as an orifice, when placed in a dead-ended line if pulsation exists.

Studies which have been made indicate that in addition to the primary pressure waves in the fluid between the disturbance and the primary element, there are also pressure waves of different amplitude existing in the primary element on the side opposite from the disturbance; and suggest that these waves might be slightly out of time with the waves on the other side, so that the effect of pressure variation is multiplied at times. Then, in addition to the primary waves, there are harmonics which are affected by the size and shape of the conduit on both sides of the primary element. The differential across the primary element where pulsation exists is thus a complicated varying pressure made up of many elements. This was shown by an electrical instrument developed by S. R. Beitler (7) in connection with the pulsation research sponsored by the American Gas Association and The American Society of Mechanical Engineers.

With the equipment as originally used it was possible to obtain a record of the pressure variations on either side of the primary element. However, a study of these records (along the line of reasoning mentioned) indicated that a knowledge of the differential pressure was desirable, and a special mixing switch was designed which made it possible to secure a record not only of the inlet or outlet pulsation waves but the differential pulsation waves as well. Fig. 5 shows some of the records taken with this instrument under various conditions of pulsation.

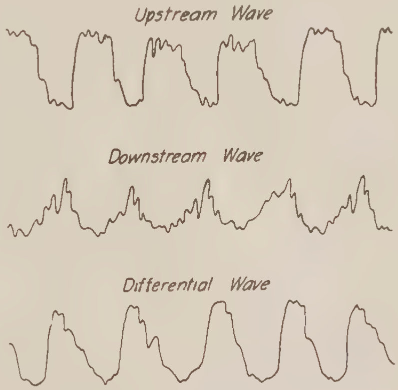


Fig. 5 Actual Pulsation Waves

A study of these records will show that there is no regularity of pulsation waves, and no apparent relation between the shape and amplitude of the waves on the two sides of the primary element, and that the differential wave bears no similarity to the other waves. These records were made at a gas-compressor station where the pulsation was produced by one compressor running at almost constant speed. The meter was placed directly on the outlet of the aftercooler so that the pulsations would be as nearly constant as possible. Because of the fact that the waves were so irregular it is practically impossible to include the effect of wave form in an equation for correcting the rate of flow.

If the pressure differential could be measured instantaneously, and if the square roots of these two pressures could be averaged, it is possible that a meter would indicate the rate of flow accurately. This would be true, because while the velocity through the primary element does not vary directly with changes in pressure waves, the energy present due to these waves will be used either in accelerating or decelerating the fluid passing through the primary element, so that the decelerations should balance the accelerations. The average flow will be indicated by the average of the square roots of the instantaneous differential pressures across the primary element.

While it is true that each meter installation presents its own problems and that each disturbance creates pulsations which cannot be analyzed, certain general conclusions in regard to pulsation can be drawn. In a paper by S. R. Beitler (8) which reports the results of research conducted at several stations, it is pointed out (*a*) that for a single diameter ratio the error becomes larger for decreasing values of the differential; (*b*) that

with a constant differential the error becomes larger with smaller diameter ratios; and (c) that at any one rate of flow the suction pulsation causes a smaller error than the discharge pulsation; but suction pulsation is just as serious as discharge pulsation if the amplitude of the pulsation is the same in both cases. These conclusions have been substantiated by other tests. Figs. 6 and 7 show the results of tests reported in this paper.

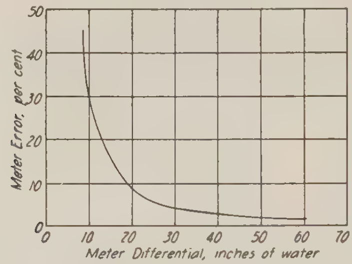


Fig. 6 Differential Vs. Error for One Diameter Ratio

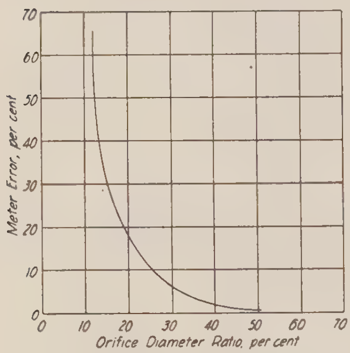


Fig. 7 Diameter Ratio Vs. Error for a Given Differential

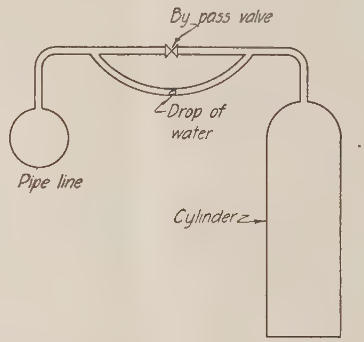


Fig. 8 Pulsation Detector (Bean)

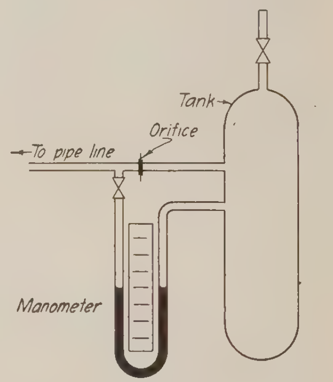


Fig. 9 Pulsation Detector (Beitler and Overbeck)

The magnitude of the error depends upon the quantity of fluid being measured, the resistance and capacity between the disturbance and the meter, and the viscosity and density of the fluid being measured.

According to H. F. Hagen (15) tests on centrifugal- and propeller-type fans are apt to be in error because of erroneous velocity and pressure readings caused by pulsating flow.

Tests (10) that have been conducted reveal that the meters made by several manufacturers gave the same amount of error when tested simultaneously.

Methods of Detecting Pulsation

If two meters are placed in series in a given line at some considerable distance apart and if the one nearer the potential source of pulsation gives a reading which differs from the reading given by the other by an amount greater than the usual discrepancy between meters, then it is certain pulsation has been created.

If a meter is subjected to constant quantity of flow and if the indicator on the meter produces a line which indicates a variable rate of flow by a wavy line, then the presence of pulsation can be suspected. This does not mean, however, that pulsation must exist if the meter charts have wavy lines nor does it mean that every case of pulsating flow will produce a wavy line.

One pulsation detector was devised by H. S. Bean (9) which had only the defect that it was too sensitive for most installations. In other words, it might indicate severe pulsations while the pulsations might be so slight that the meter error would be negligible. A sketch of Mr. Bean's detector is shown in Fig. 8. The by-pass valve is left open until the pressure in the tank reaches main pressure. The by-pass is then closed, and if the drop of water moves back and forth pulsation is present. Any slight change in static pressure or a pulsation of appreciable magnitude causes the water to be withdrawn into either the gas main or the pressure tank. If the drop of water does not move the flow is certain to be pulsationless.

A device designed by S. R. Beitler (7) and J. E. Overbeck is similar to the one just described and is shown in Fig. 9. In this device the gas-pressure tank is connected to the meter-run line through a $^1/_2$-in. line in which is placed an orifice plate, the diameter of the orifice being $^1/_8$ in. The manometer shown in the figure is then piped up to measure the pressure drop across the $^1/_8$-in. orifice. As has been pointed out previously, an orifice subject to pulsations will show a differential pressure even when there is no flow. This device takes advantage of this fact, and a differential reading on the manometer is an indication of the presence of pulsations. This device partially overcomes the defect of Mr. Bean's pulsation detector in that the liquid does not escape so readily but large rapid changes in static pressure will force the liquid out of the manometer. Slight changes of static pressure will cause differentials on the manometer which are not caused by pulsations. As in the case of Mr. Bean's detector, if no differential is shown on the manometer then it is certain that no pulsation exists.

Another instrument, which was electrical and also devised by Professor Beitler, consisted of two diaphragms of stainless steel about 1 in. diam and $^1/_{32}$ in. thick which were connected to each side of the primary metering element by very short connections and so constituted that a pressure variation would be indicated by a very slight movement of the diaphragms. Piezoelectric crystals were used to indicate the motion of the diaphragms, and the voltage produced was amplified and recorded on a recording voltmeter. The whole apparatus gave a record of the wave form at both the inlet and outlet pressure taps. The recorder was capable of responding to very rapid fluctuations of the pressure, and the complete record of the form and amplitude of the pressure variations was made on a high-speed chart. The wave forms shown in Fig. 5 were recorded by this instrument. If no pulsation is present a straight line is drawn. The entire apparatus was constructed of standard types of instruments manufactured by Brush Development Corporation, Cleveland, O.

Another instrument described by T. K. M. Smith and R. E. Morter (10), which was used for detecting pulsation, was made up of a sylphon bellows to which a pen was attached. When pulsation causes the bellows to move the pen makes a mark on a strip chart driven by a spring motor. The record of the variations of pressure is thus made on the chart.

Elimination of Pulsation

Although it is now generally felt that no equation can be written for the error caused by pulsation and no meter can be built that will measure correctly if pulsation exists, many such equations and meters have been devised.

It was pointed out by Judd and Pheley (6) and since found true by others that: (*a*) it is not feasible to apply a correction factor to a meter since the correction factor would probably change with every installation and every condition of flow; (*b*) the laws of pulsating flow are not well enough known to set up an equation for a correction factor; and (*c*) each installation presents different problems. Most of the writers who have developed equations for correction factors have failed to take into account the complexity of the pulsation wave and have treated it as a simple one. Some of those who have devised meters which correct for pulsation have failed to take into account one very important fundamental, namely, that pulsation is the result of pressure variations as well as velocity variations.

Literature which deals with many of these equations and meters will be referred to.

J. L. Hodgson (11) presents the description of a device for metering air with pulsating flow. A vane which oscillates with the pulsations carries a pointer through a hairspring. The pointer indicates the mean of pulsations. The rapid motion of the pointer is damped by vanes in an oil bath. He gives an equation for calculating a coefficient to be used with the reading obtained with the damped reading to obtain the true rate of flow. Mr. Hodgson also has another paper dealing with the subject (12).

Prof. N. P. Bailey (13) discusses a patented device which he indicates will give true-flow air under pulsating conditions. The flow of water through a calibrated orifice is claimed to be proportional to the square root of the air velocity head. Professor Bailey also has discussed similar meters in other papers (14). C. A. Dawley has patented two meters which are claimed to read correctly even if the flow is pulsating; one was patented in 1927 and the other in 1928.

H. F. Hagen (15) describes an instrument he has developed for determining the shape of the pulsation wave created by a high-speed fan.

Equations for determining the error due to pulsation or for the calculation of correction factors to be applied to the flow equation for pulsationless flow have been prepared by N. P. Bailey (14), J. L. Hodgson (12), D. Gilmour (16), and A. H. Gibson (2).

It was thought for a time that the effect of pulsation could be eliminated in a meter by placing obstructions or throttles in the gage lines. In 1922 it was pointed out by H. P. Westcott (17) that this would have no beneficial effect. Tests conducted by Eagle and Daberko (18) at The Ohio State University in 1937 showed that such devices increased the error rather than reduced it.

As far as is known now, the only way to remove every trace of error due to pulsation is to eliminate the pulsation itself. There can, however, be a slight amount of pulsation present which will not seriously affect the meter readings.

Judd and Pheley (6), as a result of their work, set down some conclusions in regard to the practical elimination of pulsation which have since been substantiated. Among them are the following:

(*a*) Because of high velocity of pulsation an excessive length of pipe line would be necessary to destroy pulsation.

(*b*) Throttling is effective, but a great deal of throttling is required to reduce pulsation error.

(*c*) Abrupt volume enlargements in the pipe line will eliminate the error if of sufficient capacity.

(*d*) For the same capacity a volume with a large diameter is better than one of equal volume and smaller diameter.

(*e*) Combination of throttling with a volume forming a muffler device is probably best for mechanically removing pulsations.

(*f*) Revolving baffles are partially successful in removing pulsation.

(*g*) The effectiveness of any quieting device depends upon its ability to dissipate or change the energy of pulsation.

One sure way to reduce pulsation to a negligible quantity is to have sufficient pressure drop between the disturbance and the meter. This can be accomplished by an orifice or a partially closed valve. In tests that have been conducted the pulsation error on the downstream side was insignificant when the pressure drop through the restriction equaled or approached the critical. This, however, in the case of a compressor, is a costly method since the purpose of the compressor will partially be defeated.

A meter placed at a considerable distance from a disturbance will not be subjected to the same degree of error as a meter placed near the disturbance. Edward Sackett (19) points out that in one instance a meter placed 6 miles from a compressor had no error. E. P. Fischer (5) noted that a meter 17 miles from a compressor had no error while one near the compressor gave considerable error. Just how far pulsation will be carried in a transmission line depends upon a number of factors and will probably be different for every installation.

Volumes placed in a line have proved quite effective in reducing pulsation to the point where meter error is no longer appreciable. The fluid having pulsation is discharged into a tank having considerable volume from whence it flows on into the line again. Several modifications of this principle have been used with success. Smith and Morter (10) discuss the use of volume tanks. They also suggest that a tank have a volume of 100 cu ft per million cu ft of gas per 24 hr reduced to actual volume under the existing pressure and that the length of the cylindrical portion of the tank be 1.75 times its diameter. One tank of their design is shown diagrammatically in Fig. 10. The pressure drop through these tanks is very low.

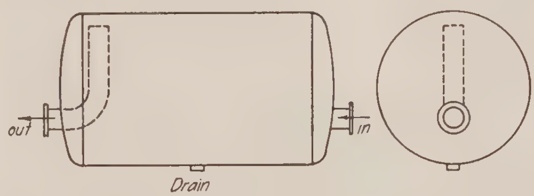


Fig. 10 Pulsation Eliminator Tank (From Smith and Morter)

Tests of volumes were conducted by Eagle and Daberko (18) in which they used the volumes in two different ways. The arrangement of the piping to the tank is shown diagrammatically in Fig. 11. The pulsation was produced by a two-stage compressor having a capacity of 350 cfm. In one series of tests they passed the air from the compressor directly through the tank by closing valve A and opening valve B (Fig. 11), valve C also being open; and in another series of tests they arranged the tank for "breathing" in the manner of a surge tank by opening valve A and closing valve B with the valve C being again open. It was found that the former method (passing the pulsating air directly through the tank) was far more effective in removing the pulsa-

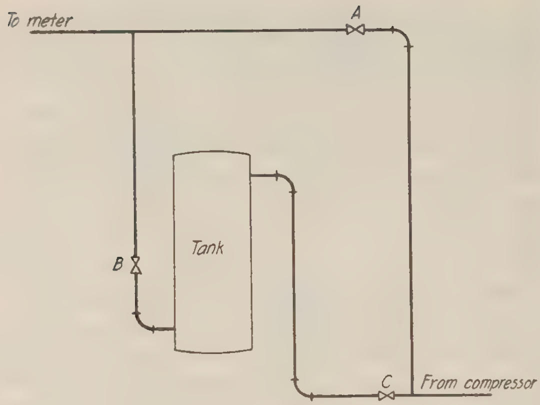


Fig. 11 Ohio State Test Tank Arrangement

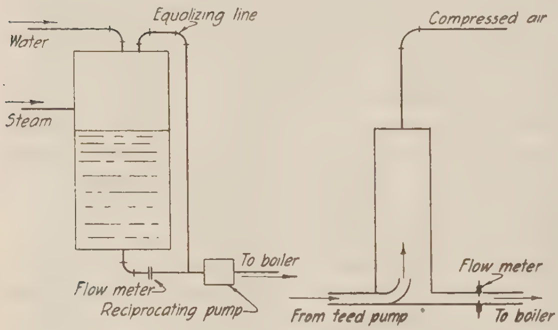


Fig. 12 Pulsation Reducers (Melas)

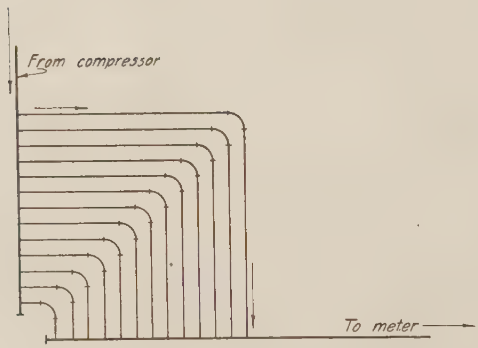


Fig. 13 Piping for Pulsation Reduction (Fischer)

tion than the latter. The tank used in these tests had a volume of 54 cu ft.

They also conducted tests when the air was passed directly through a tank having a volume of only 3 cu ft. They found that the smaller volume was about as effective in removing pulsation as the larger one. They then used the small volume lined with sound-absorbing material and placed baffles of the same material across the path of the air flow. They found that the acoustic material aided in removing pulsation to a small extent but that the acoustic material was destroyed under the action of the pulsation. They also found that the pressure drop through

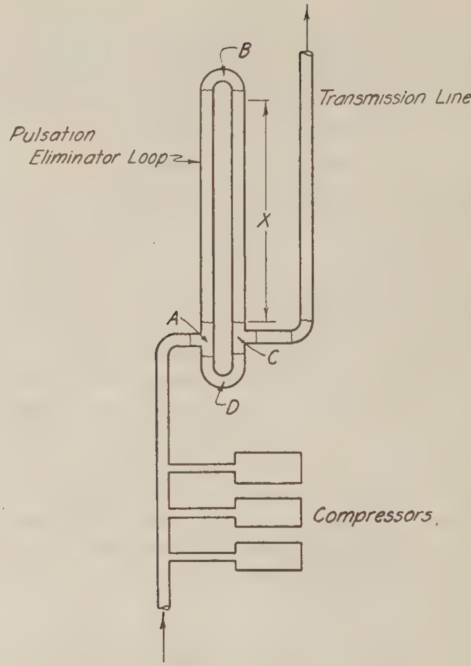


Fig. 14 Pulsation Eliminator Loop

the acoustical device was slight and concluded that the benefit was due to the sound-absorbing qualities of the device.

Two devices using the surge-tank principle are described by William Melas (20, 21, 22, 23) and shown in Fig. 12. It should be pointed out that these four articles are practically the same. Mr. Melas has used these devices in connection with a meter measuring the flow to a boiler feed pump.

F. P. Fischer (5) constructed a device (Fig. 13) which consisted of piping so arranged that various portions of the fluid under pulsating flow would have paths of different length so that the pulsation waves would interfere with one another as they joined the main pipe. Theory indicated that the waves would be broken up by this method. Elimination of the pulsation proved to be only partially accomplished.

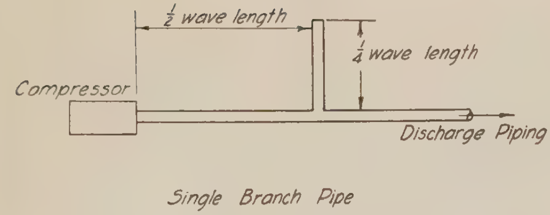
In a report written by William Mosteller to J. T. Cortelyou, both of the Southern California Gas Company, Mr. Mosteller includes a discussion of the pulsation eliminator shown in Fig. 14. He points out that for maximum effectiveness in pulsation elimination, the length of the path ABC should equal the path ADC plus one half the length of the pulsation wave. Mr. Mosteller also gives an equation for calculating the wave length and the length X in the loop.

Tests made on a pipe line from a compressor before and after the installation of this type of loop indicated that the pulsation downstream from the eliminator was less severe after the eliminator had been installed. The pulsation in the line between the eliminator and the compressor was more severe in some sections of the line and reduced in other sections.

Another device which might be used as an eliminator is mentioned by Mr. Mosteller and is shown in Fig. 15.

After having studied a good many reports of test work, including some of the reports just discussed, L. K. Spink (24) has concluded that meter measurement accuracy may be improved by the following:

(*a*) Operating at a higher differential, i.e., in a multiple-



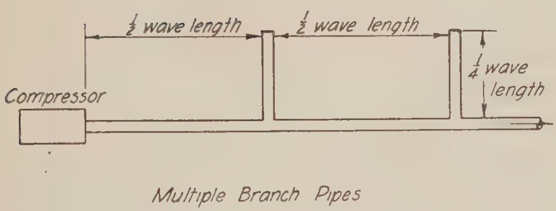


FIG. 15 ELIMINATORS CONSISTING OF BRANCH PIPES

meter run, shutting off one or more runs; or installing a smaller orifice in an existing meter run.

(b) Installing a higher range differential gage and changing operating conditions in order to use the increased range.

(c) Reducing the pipe-run diameter so as to use a higher orifice-to-pipe diameter ratio, still operating at differentials as high as practicable. Increasing the ratio of the orifice diameter to pipe-run diameter will reduce the pulsation error if the differential remains constant.

(d) Installing mufflers, headers, restrictions, or combinations of capacity and pressure drop between primary device and source of pulsation to reduce pulsation amplitude.

(e) Locating the primary device at a point where the pulsation amplitude is lower (as on the suction side of compressors).

MEASUREMENT OF PULSATION

Because of the fact that so many variables enter into an equation for pulsating error and because each installation has peculiarities of its own, computation of the error due to pulsation is out of the question. It is therefore necessary to have some instrument to determine the presence and extent of the pulsation error.

A second electrical instrument suggested by S. R. Beitler using piezoelectric crystals was developed in which the amplitude of the pulsation wave was measured by an indicating voltmeter mounted on top of a small box containing the diaphragm holders, the amplifier, and the mixing switch, so that the entire apparatus was self-contained. The power for amplification was supplied by dry cells so that the apparatus was portable. This instrument was also developed and manufactured by the Brush Development Corporation. Operating experience indicated that there was some question about the calibration of the electrical apparatus and, because of its sensitivity to pressure variations, it was extremely difficult to obtain a representative reading. A mechanical device which gives the amplitude of the differential-pulsation waves has been developed by S. R. Beitler (25) and J. E. Overbeck. This device, shown diagrammatically in Fig. 16, is a variation of the indicator and has proved very satisfactory in operation.

In its present form the mechanical pulsometer consists of two cylindrical volumes, A and B, Fig. 16, separated by a sylphon

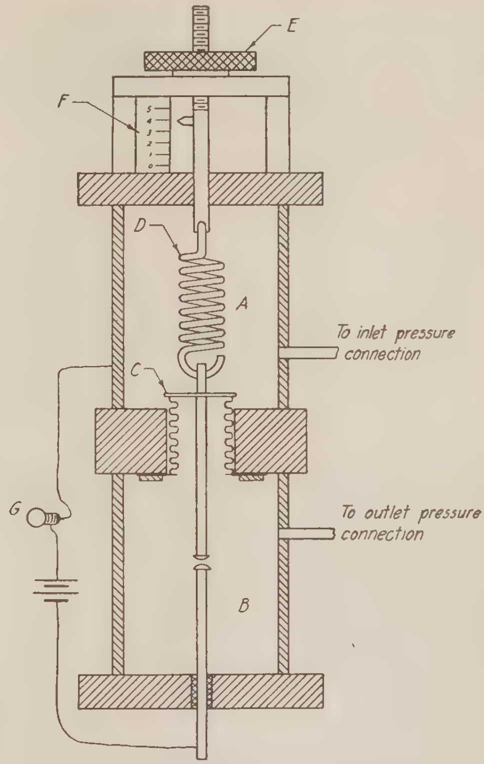


FIG. 16 MECHANICAL PULSOMETER

bellows C so that there will be no leakage between the two volumes which are connected to the inlet and outlet pressure connections of the meter, and as close to the primary element as possible. A coil spring D is mounted in volume A, which volume is connected to the high-pressure tap. The tension on this spring can be changed by turning the handwheel E which is outside the instrument. The scale F (each division represents $^1/_{10}$ in.) indicates the elongation of the spring. Electrical connections are arranged in volume B, which is connected to the low-pressure tap, so that any motion of the bellows caused by differential pressure will be indicated by a light G located on the outside of the instrument. During operation the tension of the spring is gradually increased until the light goes out. This indicates that the differential pressure has been equalized by the spring tension. When this occurs the tension of the spring should represent the differential pressure due to the flow plus the maximum amplitude of the differential pressures.

The instrument spring can be calibrated by comparing its scale readings with any differential gage when connected to a set-up where there are no pulsations present. When measuring pulsation amplitude the instrument is almost totally unaffected by inertia, since the moving parts are practically at rest when the determination is made.

To determine the correlation between pulsometer reading, meter differential, and meter error a research project was sponsored by the American Gas Association and The American Society of Mechanical Engineers. A special test station, shown diagrammatically in Fig. 17, was constructed near Sugar Grove, Ohio, at the Crawford No. 2 Station of the Ohio Fuel Gas Company. (It was the good fortune of the author to be a member of the research working committee during the period that many of

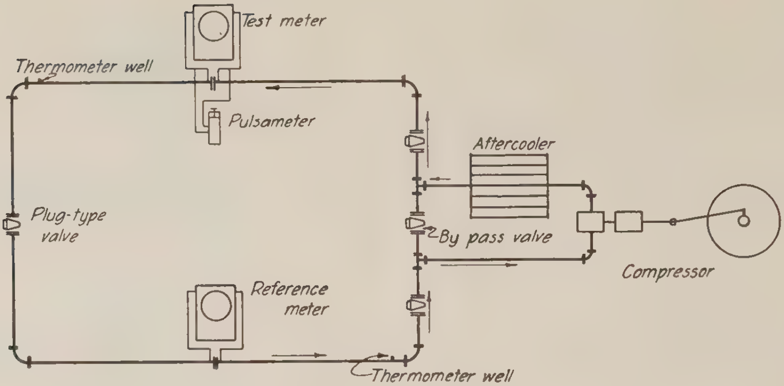


Fig. 17 Arrangement of Pulsation Test Piping
(Crawford No. 2, Sugar Grove, Ohio.)

the tests were conducted.) This test station was built with two orifice meters in series, one on the discharge and the other on the suction line of a large gas compressor. Gas was supplied to the first orifice directly from the outlet of the aftercooler of one compressor, and after passing through the second orifice meter, was discharged back into the suction of the same compressor.

With this arrangement pulsation could be maintained as nearly constant as possible and the other measured conditions could be easily changed to give the pressures and rates of flow desired. Throttle valves (plug-type valves) were placed in the line between the two meters and also between the compressor and the inlet and outlet of the test setting. By means of these valves it was possible to regulate the pressure on either one of the meters so that it would not be subjected either to suction or discharge pulsations from the compressor.

This meter was then used as a standard or reference meter and the measurement error due to pulsation was determined by comparing the quantity indicated by the meter subjected to pulsation (or test meter) against that indicated by the reference meter. The test meter was equipped with both "flange" and "pipe" taps, and readings were taken on both simultaneously in order to determine what effect the location of pressure taps might have on the test results.

In order to eliminate the effect of small variations in orifice plates and runway construction, so-called "unity tests" were run for each arrangement of orifice plates. In making these tests the flow into and out of the setting was throttled to the extent that there was no pulsation on either meter. By comparing the quantities as measured by the two meters under these conditions, an indication was obtained of how close the meters would check each other when no pulsation was present.

As shown in Fig. 17, the setup is as it was arranged for the determination of discharge pulsation. When the study was made on the effect of suction pulsation the function of the meters was interchanged and the pulsometer was moved to the other meter.

Since changing the speed of the compressor would change the type of pulsation, a by-pass was built into the setting between the inlet and outlet so that it was not necessary to change the speed of the compressor in order to change the rate of flow through the setting. The compressor used was a two-cylinder double-acting Ingersoll-Rand machine having $18^1/_2$-in. × 48-in. cylinders and operating at approximately 70 rpm.

The curve shown in Fig. 18 was determined from the tests at this station. In preparing this curve the results of about 500 tests on orifices with diameter ratios of from 20 per cent to 80 per cent in 2-in., 4-in., and 6-in. lines were studied. It was decided not to attempt to determine any correction factors but to concentrate on the determination of the conditions under which meters would measure accurately. Taking many facts into consideration, it was decided that a meter, subject to pulsation, would be normally within the limits of commercial accuracy when the error was less than 1 per cent. Taking into account spot tests made with commercial setups and recording gages, it was felt that the probable accuracy of any individual test not subject to pulsation was within the range of ±0.5 per cent.

All of the test points were plotted on a large sheet, and the percentage of error was marked on the points. The line was then drawn through the points where the error was approximately 1.5 per cent. Because of the inherent limitations on the accuracy of the tests there were three or four points with an error greater than 1.5 per cent below the curve and ten or twelve points with less than 1 per cent error above the curve. It will be

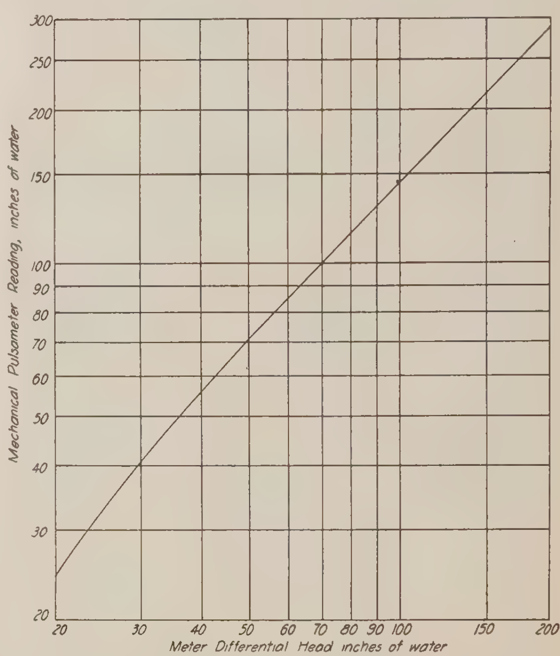


Fig. 18 Relationship Between Meter Differential and Pulsometer Reading for One Per Cent Error

noted that the same curve applies to both flange and pipe taps.

This curve can then be used to determine whether or not a meter measuring pulsation flow is accurate. It is necessary only to attach the pulsometer to the pressure taps and take a reading on it at the same time that the differential pressure is determined. The calibration curve is then consulted to find the pulsometer reading in inches of water. If the point, when plotted in Fig. 18, falls below the curve, then the pulsation error is less than 1 per cent and the meter is probably within the range of ordinary commercial accuracy. If the point falls above the curve, then the pulsation error is greater than 1 per cent, and the measurement with the meter will not be accurate.

For differential pressures less than 32 in. of water the experimental curve was intentionally curved downward at the lower readings, since operating experience has shown that measurements made at low differentials when the flow is pulsating, are very sensitive to changes in pulsation, and it was felt that the line should be lower than normal to allow the effect of these accidental variations. Because of erratic results the empirical curve was not drawn for differentials below 20 in. of water and if pulsations are present, it is believed better not to attempt to meter the flow where the differentials are as low as that. These curves were plotted on logarithmic paper so that points with the same percentage of error would be approximately the same distance from the curve for any value of the differential head.

In order to make use of the available test results all pulsometers must have the same general dimensions as the one used to determine the effect of pulsation, since the shape and dimensions will undoubtedly have some effect on the instrument reading. This pulsometer is patented and is now being built by the Refinery Supply Company, Tulsa, Okla.

As previously mentioned, the location of the curve drawn in Fig. 18 was determined by experiment, and it was thought that the results might be better understood if there was a rational explanation for drawing it in this manner. In order to see the relationship of the empirical curve to some theoretical curves, Fig. 19 was plotted.

Equations developed in the second section, "Mathematical Analysis," were used in obtaining these curves. Values of the error were found for several values of p/h. Then p/h was plotted against the meter error for each wave form, and the value of p/h for 1 per cent error for each of the three wave forms was determined from the curves. Then a relationship was set up between the pulsometer reading $(p + h)$ and the meter differential (h).

In comparing the theoretical curves with the empirical curve it has been assumed that the pulsometer reading is the sum of the differential plus the amplitude of the pulsation. These curves are all drawn for 1 per cent error. It will be seen that there is a reasonable degree of relationship between the empirical and theoretical curves.

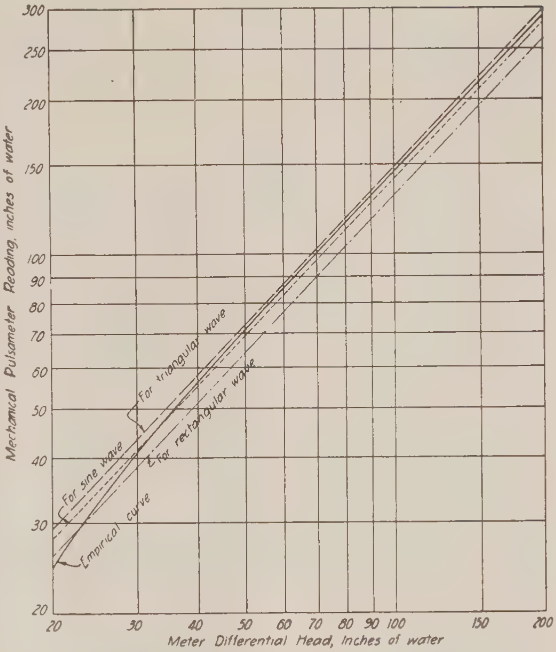


Fig. 19 Theoretical and Empirical Curves for One Per Cent Error

Research to Be Done

Although pulsation has received considerable thought and attention for a number of years and although devices have been built which indicate the presence and others which to a degree indicate the magnitude of the pulsation, considerable work remains to be done.

No use of the pulsometer shown in Fig. 14 has been made in connection with a vapor having pulsating flow. Just how the instrument would behave if the volumes on both sides of the bellows or if the lines leading to the instrument were filled with a liquid remains to be learned through an investigation. At the present time it also seems quite necessary that an instrument be developed for use with meters having low differentials, say, 30 in. and less. The curve now used, Fig. 18, cannot be used with such low differentials with any degree of confidence.

It is also felt that more of a study of the pulsation theory should be made. Heretofore, pulsation produced by a commercial machine has usually been studied. It would seem to be logical to make a laboratory study of pulsations which could be carefully controlled and varied. In this way the degree of error produced by different devices and with different wave forms could be determined.

The studies mentioned are only a few which need to be made. There are, in fact, many others of equal importance.

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Discussion

J. E. Overbeck.[4] The writer has had the good fortune of working on some of the research studies explained, as well as of using some of these research data in practical applications.

If the operating gas-measurement engineers will take full advantage of the information now available, which is set forth in this paper, there is certainly much to be gained. In putting these data to practical application, the first step should be to make a study of all measuring stations to determine which are liable to be affected by pulsation. This can be done by reviewing all stations and determining which ones are located adjacent to compressor stations. The term "adjacent" may mean that the meters are located on the compressor-station lot, or it may mean that they may be up to 20 miles away from the compressor station where there is very little pressure drop or resistance between the measuring station and the compressor station.

[4] Columbia Engineering Corporation, Columbus, Ohio.

Each station suspected should be checked with a mechanical-type pulsometer of the type shown in the author's Fig. 16, to determine whether or not the effect of pulsation is sufficient to cause an error in measurement. Such tests should be made under all of the operating conditions encountered, including rates of flow, number of compressors being operated, lines and connections being used, whether or not the gas is passing through scrubbers, coolers, dehydrators, etc. Generally, the pulsation effect will be the greatest with the least number of compressors being operated and with the least number of connections, etc. It is probable that these tests will eliminate a high percentage of the stations suspected, that is, the effect of pulsation will be within the limits as shown by the author's graph, Fig. 18.

Where the effect of pulsation exceeds these limits and is to be reduced, local conditions surrounding the meter stations are big factors, and which method to use, of course, will depend upon these local conditions. In many instances it may only be necessary to change operating conditions or make only slight changes in the equipment, such as changing orifice size, operating at higher differentials, etc.

From the operating standpoint, the charts shown in Fig. 6 and Fig. 7 of the paper, are helpful in determining whether or not to change the meter differential or change the orifice-to-pipe diameter ratio to obtain the most efficient results. For example, where it is not necessary or practical to change the pipe-run size, it will be found that it is more effective to operate at the highest differential, even though flow conditions require a slightly lower orifice-to-pipe diameter ratio.

For those stations where the pulsation effect is beyond the limits, as shown in Fig. 18, the suggested remedies pointed out by the author will certainly be helpful in reducing the error. Many other types of pulsation dampers have been used. Generally, any change in piping that will, through interference, break up the pulsation waves at the point of measurement will increase the frequency, lower the amplitude, and thus reduce the meter error.

It may be found necessary at some stations to make more than one change, that is, increase the orifice-to-pipe diameter, increase the operating differential range and install some pulsation dampers, or change the arrangement of piping between the source of pulsation and meter.

Author's Closure

The comments by Mr. Overbeck relative to checking and correcting meter installations for pulsation are a valuable addition to the paper.

The author feels certain that there are many others who have something to offer which will shed more light on pulsation. There should be several who have tried eliminators besides the ones shown. There should also be additional references. No doubt the theory of pulsation as given in the paper can be expanded.

The author will appreciate receiving any information on pulsation that any reader may have.

Boiler Design for Extreme Load Variations

By M. H. KUHNER,[1] WORCESTER, MASS.

A summation of required design provisions for boilers adapted to operate under quick demand changes. Reference is made to the quick-pickup boiler in the Harbor Steam Plant of the City of Los Angeles and the steel-mill boiler at the Homestead Works of the Carnegie-Illinois Steel Corporation. The importance of water storage, unrestricted path for water and steam in circulation, steam release with minimum disturbance of water body in drum, steam-temperature control during unbalanced firing, and the effect of each item on the others, is discussed. Recommendations are made for determining required capacity of fuel-burning equipment and fans in reference to the extent and rapidity of steam-load changes.

MOST boilers in public-utility and industrial plants are called upon to carry reasonably steady steam loads. Changes in steam demand are gradual and are often predicted in sufficient time to permit required preparations for load increases or decreases. Boilers of what can be termed standard design usually give a good account of themselves under this type of service.

There are, however, other steam power plants, both public-utility and industrial, in which standard boilers would be quite unsuited. These are the plants where steam demand is unpredictable and where load changes are instantaneous and extreme. Steam-electric plants which stand by hydroelectric stations come under this category.

An outstanding example in the utility field is the Harbor Steam Plant of the City of Los Angeles, where a single boiler and turbine-generator unit must be prepared to pick up loads at the rate of 50,000 to 60,000 kw within seconds and without previous warning, Fig. 1.[2]

In the industrial field the steam power station of the Homestead Works of the Carnegie-Illinois Steel should be mentioned, Fig. 2. A single boiler and turbine-generator unit feeds steam and power into the lines of the entire system of "Big Steel's" Pittsburgh district. Whenever a deficiency in steam or power develops in any one of the steel-mill units, it must be made up by the Homestead Works Power Station. Instantaneous load-rate changes as great as 250,000 lb of steam per hr and 20,000 kw are quite frequent.

In the design of boilers suitable to serve safely and reliably in a power plant with sudden and extreme load variations, the following major points must be given careful consideration:

1 That portion of heat stored in the boiler which will become available at once for steam formation with reduction in pressure.

2 Path of water and steam through circulatory system.

3 Release of steam formed by flashing in the water body.

4 Control of steam temperature while steam output and firing rate are out of balance.

5 Effect of rapid thermal changes in steam, feedwater, and combustion gas on boiler pressure parts.

6 Methods of firing and capacity of fuel-burning equipment.

7 Selection of mechanical-draft equipment and drives as to type, capacity, and control.

8 Construction of setting enclosure for momentary high internal setting pressures.

9 Automatic equipment for control of firing and flow of fuel, air, gas, and feedwater.

Heat Storage

When the steam-output rate of a boiler increases elevenfold in a matter of 4 or 5 sec, as is the case with the Harbor Steam Plant boiler, it cannot be expected that the firing rate can be made to increase proportionately and at the same time. Response of the automatic control naturally lags because the control receives its primary impulse from steam pressure or steam flow. It takes time for the fans to speed up, for dampers to arrive at the new setting and for increased fuel and air quantities to reach the furnace. During this time lag the boiler is required to perform somewhat like a steam accumulator. Steam delivered over and above that quantity generated by the existing furnace heat input must be produced by flashing, using heat stored in the boiler water. Part of this heat is made available with reduction in boiler pressure. Steam is formed in the body of water contained in drums, tubes, and headers. The greater the weight of water below the working level, the more heat is available for flashing of steam and the smaller will be the pressure loss for a given steam output. To have the highest possible pressure and heat head available at the moment of load pickup, it is advisable to control the drum pressure of the boiler rather than the pressure at the superheater outlet or at the turbine throttle. The difference between the drum pressure existing at the moment of the beginning of the load pickup and the lowest drum pressure permissible for the work to be done determines the steaming rate during accumulator operation, and the maximum permissible time lag between start of high steam output and correspondingly increased furnace heat release.

The example of the Harbor Steam Plant boiler is probably best suited to illustrate the importance of heat storage. A standard boiler unit of 650,000 lb capacity at 1000 psig, 900 F usually contains approximately 2500 cu ft of internal cubical space below the working water level in drums, tubes, and headers. The Harbor boiler has 3500 cu ft. Its hourly steaming rate is between 60,000 and 100,000 lb, while it is under stand-by load. The drum pressure is maintained at 1040 psig. It can be assumed that during the low steaming rate of stand-by operation the space below the water line is occupied by substantially solid water, that is, water almost free of steam bubbles. The turbine is constructed so that the peak generating capacity can be maintained with 840 psig throttle pressure. Maximum pressure drop through the superheater, steam-temperature control system, and steam line from boiler to turbine, is approximately 100 psi. The boiler-drum pressure may therefore be permitted to drop 100 psi during accumulator operation before capacity power output of the plant is impaired.

Saturated water totaling 3500 cu ft at 1048 psig, which is the mean pressure between water level and lowest point of the system, equals 160,000 lb. It contains 551.9 Btu per lb, or a total of

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[1] Chief Mechanical Engineer, Riley Stoker Corporation. Mem. A.S.M.E.

[2] "The Harbor Steam Plant Boiler," by Max H. Kuhner, *Power Plant Engineering*, vol. 49, June, 1945, pp. 86–89.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 17–20, 1946, of The American Society of Mechanical Engineers.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

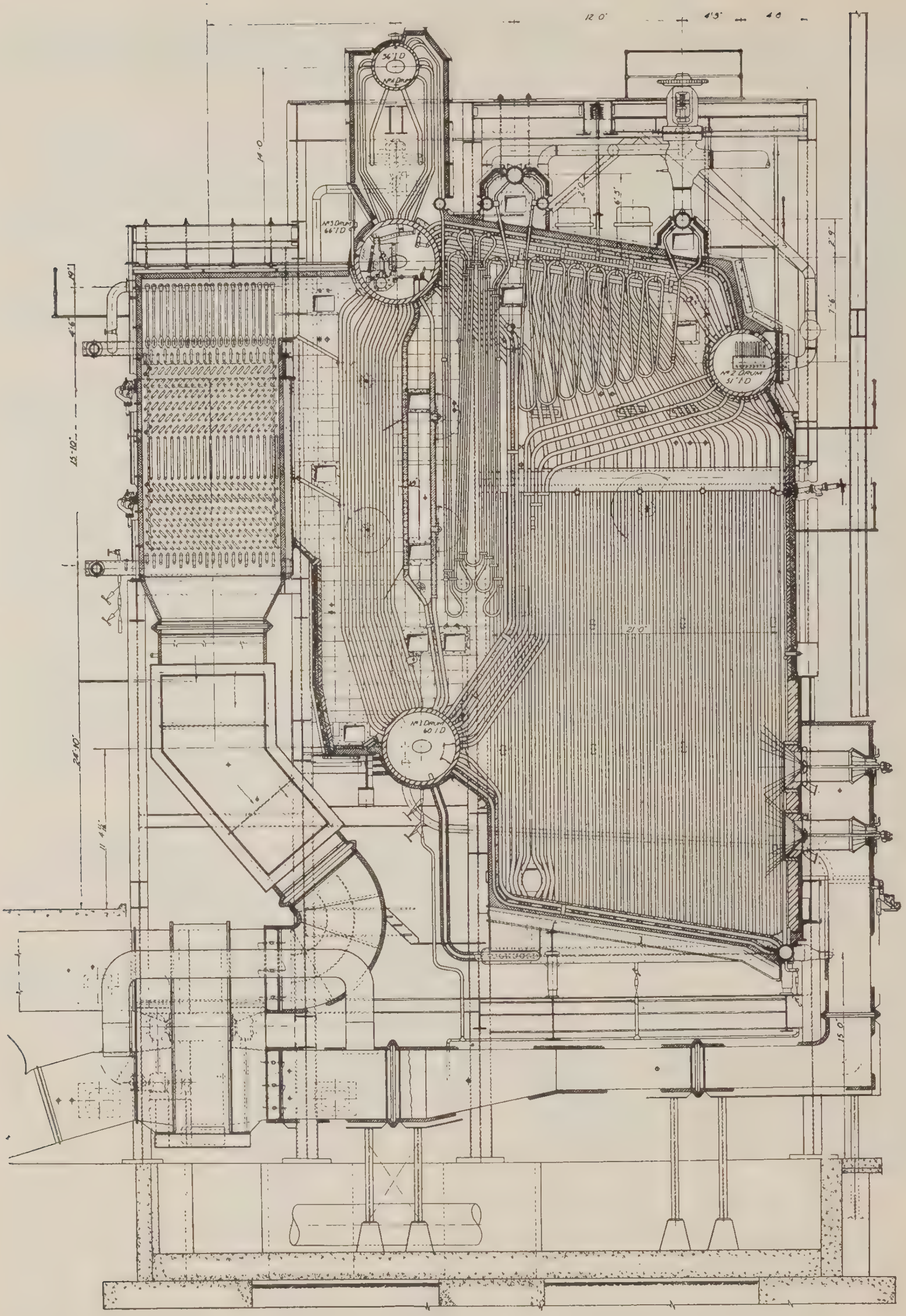


Fig. 1 The Harbor Steam Plant Boiler No. 1

(650,000 lb per hr, 1040 psig drum pressure, 915 F steam temperature. Boiler and 65,000 kw turbine generator operate as "spinning reserve" for hydropower. Boiler automatically picks up steam-load rates of 600,000 lb per hr within a few seconds.)

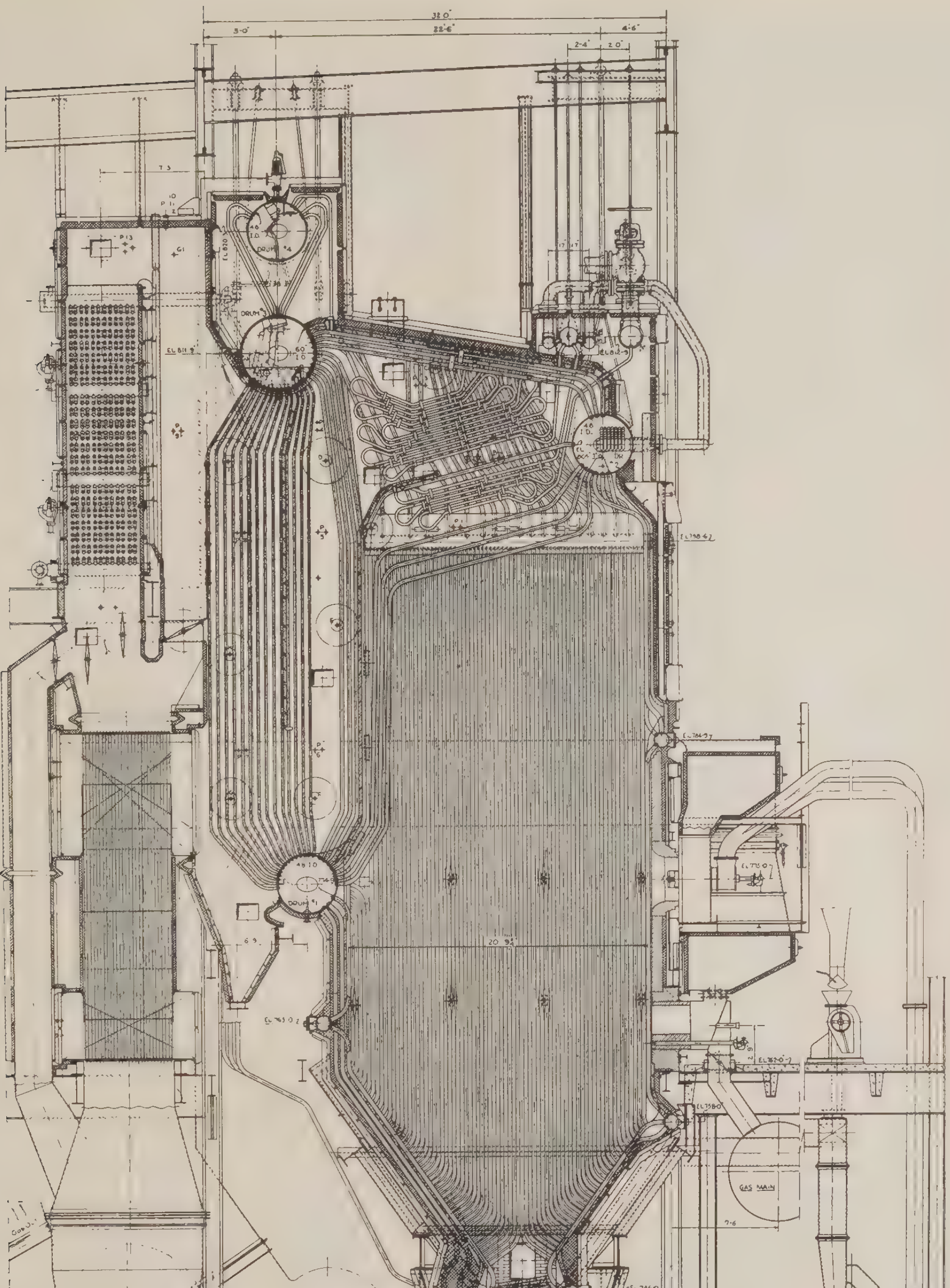
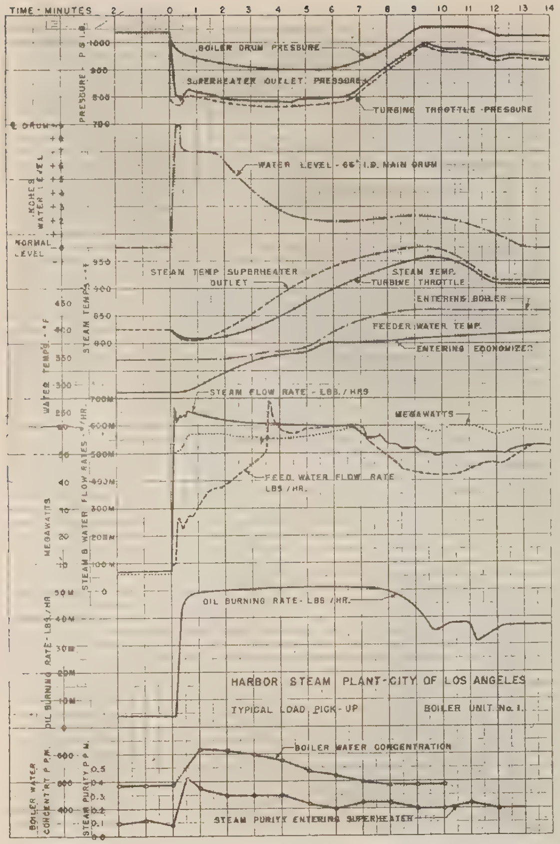


Fig. 2 450,000 Lb per Hr Boiler Unit of Carrie Furnaces Plant, Carnegie Illinois Steel Corporation

(Steam-output rates change rapidly between 200,000 and 450,000 lb. Fuel is blast-furnace gas and pulverized coal. Steam temperature automatically controlled by heat-exchanger-type control.)



RILEY STOKER CORP

Fig. 3 Performance Data of Harbor Boiler Recorded During Typical Steam-Load Pickup From 70,000 Lb Stand-By Rate, to Approximately 650,000 Lb Per Hr

88,200,000 Btu. If the mean pressure is 100 psi lower, or 948 psig, each pound of water contains 537 Btu and the total heat in 160,000 lb of water is 85,900,000 Btu. The difference, 2,300,000, is available as heat to be flashed into steam.

The heat of evaporation at the mean pressure between the initial drum pressure (1040 psig) and the pressure at the end of the accumulator cycle (940 psig) is 648.5 Btu per lb; 2,300,000 divided by 648.5 = 3550 lb of steam can thus be flashed from boiler water.

The Harbor boiler must deliver steam at the rate of approximately 650,000 lb per hr to develop 55,000 kw output under rapid pickup conditions. The firing rate is established to produce only approximately 60,000 lb per hr prior to the moment the load rise begins. The difference at the rate of 590,000 lb per hr must be made up by flashed steam. A steaming rate of 590,000 lb per hr is 164 lb per sec; 3550 divided by 164 is 22; therefore the heat stored in the boiler water permits the maintenance of 650,000 lb per hr steam-output rate for only 22 sec duration if the firing rate corresponding to 60,000 lb steaming rate is maintained. The boiler-drum pressure will drop from 1040 psig to 940 psig during this period. The 22-sec time lag is usually sufficient for the fuel-burning rate to increase and to meet the new steam-output rate, Fig. 3.

Circulatory System

While the design for unidirectional circulation of water and steam is a basic requirement for all water-tube boilers, it becomes much more important where boilers are subjected to rapid load variations. Not only must the design of the system insure that there can be no stoppage or reversal of flow in any of the steaming tubes during a quick load change, but the path of water and steam must be easy and direct. Abrupt changes in internal cross-sectional area within the circulatory system must be avoided.

A sudden change in steam-output rate results in a momentary disturbance of the flow of water and steam in circulation. If the load suddenly rises with a resulting drop in drum pressure, steam is formed in the entire body of water contained in steaming tubes as well as in downflow and feeder tubes, wherever the water temperature corresponds with steaming temperature. The buoyancy of the steam globules formed in the downflow column opposes the downflow of solid water. The rate of circulation is naturally low during low rates of evaporation. If there are restrictions and obstructions placed in the path of natural circulation, a quick load rise will certainly produce stoppage and reversal of flow in some steaming tubes. This may lead to tube failures. If, on the other hand, the circulatory system is designed so that water and steam can find no serious obstruction in their path, full use can be made of the flywheel effect of the mass of water in motion.

It is calculated that the average circulating velocity of water in the Harbor boiler is approximately 2 fps during 60,000 lb per hr evaporation. The momentum of 80 tons of water traveling at this velocity is an effective force opposing the buoyancy of steam flashed in the water during a load pickup. It assures that unidirectional flow is maintained in all tubes. This boiler has been subjected to a number of rapid load pickups during the past 3 years. The performance recorded by curves, Fig. 3, is typical for such surges. No circulation difficulties were detected.

Steam Release

Releasing the steam generated in the open drum space of the main boiler drum above the water level prevents geysering and turbulence of the water body in the drum and seesawing of the water level. In general, this method effects rapid separation of steam from water. A design incorporating this provision is an advantage for all boilers regardless of type and service.[3]

Rapid steam-load changes are accompanied by water-level variations because the steam formed in the body of water during a sudden load rise, not paralleled by corresponding increase in heat input, tends to lift the water level. The volume of the fluid below the water level increases in direct proportion to the volume of steam formed in the water body. Conversely, a sudden reduction in load, accompanied by a slight rise in boiler pressure, causes steam globules already formed, to collapse, resulting in a shrinkage of the volume below the water line and a drop in water level.

It is important therefore that the drum in which steam is released and in which the working water level is to be maintained, be unusually large. The maximum rise in water level must not reduce steam-release space to the point where there is danger of water carry-over, and the drop in water level must not be such that the entrance to the downflow system becomes exposed and steam, in place of solid water, is permitted to enter the downflow column.

When the drum is equipped with a steam-cleaning and drying device the efficiency of this apparatus is usually dependent upon the height of the water level in the drum. If the level is too high the steam-purifying system may become flooded; if too low, dirty steam and boiler water may by-pass the purifier with disastrous results for superheater and turbine. Because large changes in water-level elevation cannot be prevented during

[3] "How Steam Is Released in Water-Tube Boiler Drums," by Max H. Kuhner, *Power Plant Engineering*, vol. 48, September, 1944, pp. 92–94.

sudden large changes in steam-output rates, it becomes necessary to study carefully the placement of steam-purifying equipment and a larger than normal drum diameter, or a separate steam-purifier drum is indicated.

The Harbor Steam Plant boiler is equipped with a 66-in-ID steam-release drum plus a 36-in-ID steam drum placed above at 14 ft vertical center distance. The volume of water contained in the 66-in-ID drum with the water line at normal elevation is 350 cu ft while the steam space above the water line is 1110 cu ft, Fig. 4. The normal water level is established at 9 in. below the horizontal center line of the 66 in. drum. The water level may rise 11 in. from normal during a load pickup or may drop 6 in. below normal with a sudden steam-load reduction before steam release, steam purification, and water supply to downflow column are affected. The effectiveness of this arrangement is indicated by the steam purity recorded during load pickups. The solids carried away in the steam usually increase momentarily from approximately 0.1 ppm to 0.5 ppm, Fig. 3.

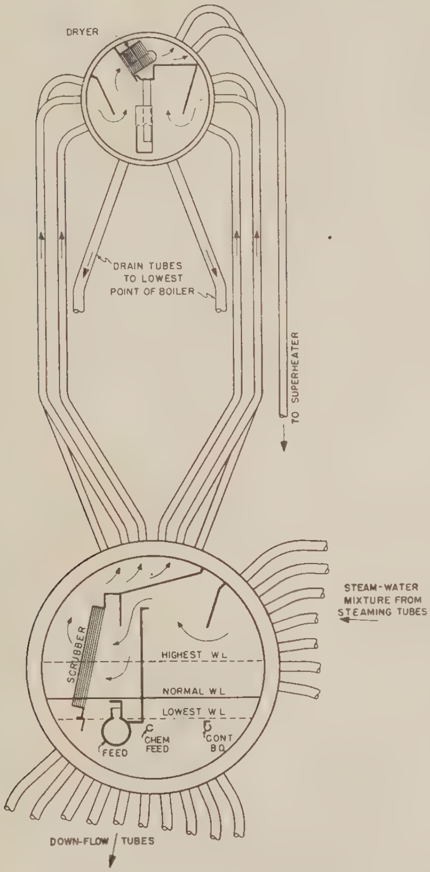


Fig. 4 Steam-Purifying System Harbor Steam Plant Boiler No. 1

Steam-Temperature Control

During normal operation of a central station or industrial boiler final temperature of steam delivered by the superheater is influenced by moisture content of the steam entering the superheater, steaming rate, length of flame, excess air used for combustion, type and kind of fuel, cleanliness of furnace wall and water heating surfaces placed ahead of the superheater, and cleanliness of the superheater surface. These are all factors which can change the steam temperature of an installation. A relatively simple system of steam-temperature control is usually sufficient to produce a workable steam-temperature characteristic in spite of the factors of normal operation mentioned. A system based on control of gas flow over superheater surface or on partial desuperheating is often sufficient to prevent a rise in steam temperature to the point where superheater and turbine are endangered, because each individual factor, or a combination of those mentioned, will not so seriously influence the final steam temperature that the range of a correctly designed control is exceeded.

With a quick-load-pickup boiler additional factors must be considered as affecting steam temperature. During a sudden load rise, and while the firing rate lags 20 to 30 sec behind the increased steam output, very little heat is applied to the superheater surface and a drop in steam temperature must be expected. On the other hand, steam temperature will rise sharply with a sudden drop in steam demand because the firing rate, corresponding with the higher load, continues several seconds after the load decrease has taken place. More heat is therefore applied to the superheater surface than can normally be carried away with the steam. Conditions are aggravated for a purely convection superheater because its normal characteristics produces a steeply rising temperature gradient with increasing steaming rate. A superheater designed so that part of the heat is received by radiation and the remainder by convection, produces a more uniform steam temperature over a wide range in load. Obviously therefore a superheater of this design insures a higher steam temperature at the beginning of the load-pickup cycle, and the effect on steam temperature by the unbalance of steam flow and firing rate at the moment of a sudden load drop is not as pronounced as with the purely convection superheater.

In addition to the points mentioned as affecting superheater performance there is one factor identified with quick-load-pickup boilers only which has a severe effect on steam temperature. Heat removed from boiler water by flashed steam, with the reduction in pressure, must be restored after the high steaming rate is established and while pressure is raised to normal. The fuel-burning rate is therefore not only that required to generate the steam delivered by the boiler, but in addition, that necessary to restore the heat to the boiler water and thus re-establish the normal operating pressure. The heat input to the superheater is considerably greater during the time pressure is being restored

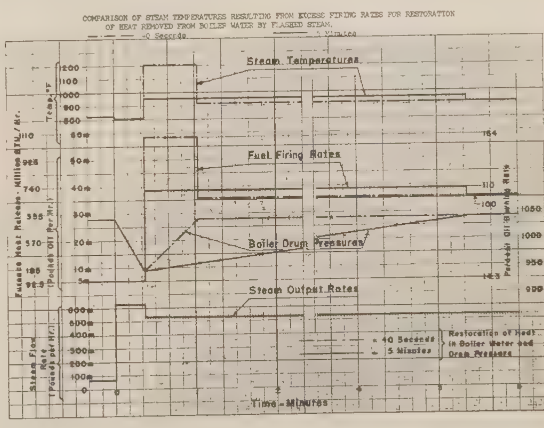


Fig. 5 Comparison of Steam Temperatures Resulting From Excess Firing Rates for Restoration of Heat Removed From Boiler Water by Flashed Steam

than required for normal steam temperature at the existing steam flow. During this period steam-output rate and firing rate are out of balance with the firing rate a considerable percentage greater than needed for the steam output. The result is high steam temperature. The excess rate of firing is, of course, in inverse proportion to the time permitted to restore the heat in the boiler water and to establish normal drum pressure, Fig. 5. The maximum permissible steam temperature usually limits the excess rate of firing and thus controls the time required for restoring normal drum pressure.

Gas by-pass dampers for controlling steam temperature are too sluggish to be effective at the moment of a rapid load increase. The heat-exchanger type of control, or one employing direct-contact water sprays, is more suitable because its action is almost instantaneous if it receives its impulse automatically from the steam temperature, and especially so if the control anticipates a steam-temperature rise by receiving its primary impulse from steam flow.

Thermal Changes

It is apparent that rapid load changes produce rapid temperature changes in feedwater, steam, and combustion gas. Fig. 3 shows that during an instantaneous load pickup by the Harbor Steam Plant unit from 70,000 to approximately 630,000 lb steaming rate, the feedwater temperature rose from 270 to 380 F in less than 6 min, while the steam temperature dropped from 825 F to approximately 800 F in 45 sec, then rose to 975 F in 8 min. The increase in firing rate, from approximately 4000 lb of oil per hr to over 50,000 lb in less than 1 min, naturally produced a rapid rise in the temperature of combustion gas passing over the heating surfaces.

Rolled tube joints and welded pressure connections must be insulated against direct impact of combustion gas to prevent damage from temperature shocks. Feedwater entrance connections must be designed to prevent water of rapidly varying temperature from contacting the drum steel, Fig. 6. The entire pressure system must be suspended in a manner which permits free thermal expansion and contraction without placing unpredictable strains and stresses on pressure parts, setting enclosure, and supporting members.

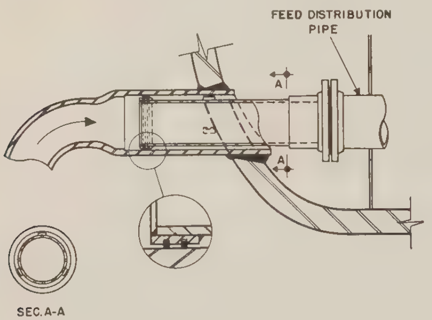


Fig. 6 Boiler-Feed Nozzle

Fuel-Burning Equipment, Type and Capacity

The rapid changes in heat-input rates required by widely varying steam demands call for extremely flexible fuel-burning equipment. Liquid and gaseous fuels are naturally well adapted for this service but burners must be of the wide-load-range type. There is no time for lighting additional burners or for changing burner tips or guns at the moment of load pickup. Neither will the available time permit taking burners out of service or making manual adjustments during a sudden drop in load.

In the case of the Harbor Steam Plant, where oil is the principal fuel, all ten burners are in service at all times. The combined burning rate is only approximately 4000 lb of oil per hr during stand-by operation, but in case of a load pickup the burning rate will have increased to more than 50,000 lb of oil per hr in less than 1 min, Fig. 3. A maximum burning rate of 60,000 lb per hr can be maintained.

Where coal must be burned stokers are usually not suitable because their operating characteristics prevent the required flexibility of heat liberation for quick steam-load changes. Pulverized-coal-firing is better adapted. Although response to load changes is rapid with correctly designed pulverized-coal equipment, the wide range of burner performance easily obtainable with oil and gas is usually not secured. A supplementary fuel, such as gas or oil, may have to assist the maintenance of stable ignition where the load swings to extremely low points.

An outstanding example of this method of operation is the steam-generating unit of the Rankin steam-electric station of the Carnegie-Illinois Steel Corporation, Fig. 2. This unit, of 450,000 lb per hr design capacity, is fired by blast-furnace gas as the main fuel, with pulverized coal to make up any deficiency. Sudden changes in steam demand at the rate of 250,000 lb per hr take place, and because the supply of the blast-furnace-gas fuel cannot be changed as rapidly as necessary to parallel steam demands (whatever quantity of blast-furnace gas remains from other operations in the steel mill must be burned under the boiler), the load swings must be handled by pulverized coal.

The typical steel-mill load of this plant differs from the spinning-reserve service of the Harbor Station. While Harbor receives its high load demands without warning as, for example, in case of a break in the transmission lines coming from the Boulder Dam hydroelectric station, the Rankin plant is usually informed of a major load change a few minutes in advance. If a load pickup is announced, pulverizers are started and remain spinning without coal. As soon as the steam demand exceeds the available supply of blast-furnace gas, raw-coal feeders start delivering coal to the mills, and within a few seconds a pulverized-coal flame is established in the furnace. The pulverized-coal burners contain oil pilots permanently maintaining an oil flame sufficient for instantaneous ignition of the pulverized coal and for support of the flame at extremely low burning rates.[4]

The capacity of fuel-burning equipment for boiler units subjected to extreme steam-load variation must be greater than for a standard boiler installation, because the heat to be released in the furnace immediately following a sudden load increase is not only that necessary to produce the steam actually generated in the boiler, but in addition, that necessary to restore boiler pressure and to replace the heat taken from boiler water while the boiler operated as a steam accumulator. This excess fuel-burning capacity must be determined by the time permissible, or available, starting with the moment of load pickup until full boiler pressure is restored. Fig. 5 shows that in the case of the Harbor Steam Plant unit an excess fuel-burning rate of 64 per cent would be required if it were necessary to restore full boiler pressure in 40 sec, whereas if 5 min were permissible, only approximately 10 per cent excess burning capacity would be necessary. This boiler is actually equipped with 28.5 per cent greater fuel-burning capacity than that required for the continuous steaming rate of 675,000 lb per hr

Mechanical-Draft Equipment

The selection of type, capacity, and driving method of fans is governed to a large extent by the required or desired rapidity and magnitude of steam-load changes for which the installation is designed. The construction of forced- and induced-draft fans must,

[4] "Modern Boilers for Steel Mill Plants," by M. H. Kuhner. *Power*, vol. 87, 1943, pp. 650–657.

in any event, be such that mechanical failures caused by the high stresses produced by the torque required for acceleration are prevented. The drive best suited for the requirements should be selected on the basis of the following considerations:

1 Direct-connected constant-speed alternating-current motors. Gas and air flow controlled by dampers or fan-inlet vanes. Quickest response to load change. High power consumption. High rate of wear and tear of fans and dampers. Operating units for dampers must be extra powerful.

2 Direct-connected variable-speed wound-rotor induction motors. Speed range only 50 per cent of maximum. Dampers required for accurate flow control. Power consumption relatively high, but lower than (1). Slower response to load changes.

3 Constant-speed alternating-current motors with hydraulic or electromagnetic couplings. Power consumption less than (1) and (2). Considerable time required for response to load changes. Dampers required for flow control in low load range.

(4) Direct-connected adjustable-speed direct-current motors, variable-voltage control (or electronic control when available for large motors). Economical power consumption. Quick response to load changes. Speed range corresponding to entire steam-load range. No dampers necessary for flow control. Motor generator set required. High investment cost. (Used with Harbor Steam Plant boiler).

The determination of capacity of mechanical-draft equipment must, of course, be governed by the capacity of the fuel-burning equipment. (Refer to section "Fuel-Burning Equipment, Type and Capacity.")

Boiler Settings

It cannot be expected that ten to twelvefold acceleration of burning rates, often taking place in less than 20 sec, will in every case occur without some disturbance of furnace conditions. In spite of the most carefully adjusted combustion control there is always the possibility that the increased fuel flow reaches the burners a fraction of a second in advance of the corresponding air supply. This may result in a furnace puff or, under severe conditions, in a light furnace explosion. One of the fans may speed up faster than the other, creating momentary high positive or negative pressures in furnace and gas passages.

It is evident that greater than ordinary care must be taken in the design and construction of the setting enclosure and gas and air passages. The setting must be strong enough to withstand minor gas and fuel explosions. It must remain tight under this severe service but it must also be sufficiently flexible to absorb internal shocks and damp them so that they are not transmitted to the supporting structure of the boiler and building. This calls for an entire steel shell enclosing the complete unit, reinforced and supported by a system of steel buckstays and girders which is independent of the main supporting structure. Permanent tightness of the joints of casing panels must be insured by flanges with closely spaced bolts, or by complete welding.[5]

The steel casing of the Harbor boiler is constructed to resist internal pressures up to 100 psf (approximately 19 in. water gage) without noticeable deflection. Air and gas passages are arranged for the most direct flow between furnace and fans.

Control Equipment

A reliable system for automatically controlling firing rate, air and gas flow, steam temperature, and feedwater flow must be in use with boilers operating under extreme load variations. It is impossible to attempt manual control of fuel, air, draft, steam, and water because all adjustments are required simultaneously and in step with the variation in steam demand.

It is not within the scope of this paper to recommend the type of control equipment to be used, but in selecting the equipment it must be kept in mind that the time lag of the control mechanism has a very important effect on the performance of the boiler equipment. The time consumed from the moment the control receives its first impulse of a load change, until the control has repositioned the equipment governing flow of fuel, air, gas, steam, and water, must be as short as mechanical limitations permit. If the response is fast the period during which the boiler must perform as a steam accumulator is short and the reduction in drum pressure is small. In this connection it is desirable to study the design of the equipment to be controlled because speedier response may be obtained by having the apparatus to be controlled designed for the least inertia. For example, multiple-leaf and louver dampers in air and gas passages have less inertia than large single-leaf dampers.

The combustion control receives its primary impulse either from steam pressure or steam flow. Because the time element of response to sudden steam-flow changes should be measured in fractions of a second, an arrangement whereby the primary impulse is received from power demand at the output side of the generator rather than from steam demand at the turbine throttle, might be worthy of consideration. Obviously the control mechanism must be self-compensating to avoid vicious hunting cycles.

It is demonstrated that the water level of a boiler rises sharply during a sudden load increase and drops as fast if the steam-output rate is suddenly reduced. The accumulator performance of a boiler and utilization of heat stored in the boiler water, are discussed in the section "Heat Storage." Keeping these facts in mind, it is readily apparent that it would be quite undesirable to have large quantities of relatively cold feedwater enter the boiler at the moment of a load surge. Cold feedwater would not only reduce the heat available in the boiler water for flashing steam, but would tend to raise the water level still further. On the other hand, the entrance of large quantities of colder feedwater is desirable when it immediately follows a sudden drop in steam demand. The water level is then below normal. Excessive heat in the boiler raises the pressure. Safety valves may start to pop. If there is heavy feedwater flow during this critical time the water level will be promptly restored to normal, and part of the excess heat in the boiler water will be dissipated in the fresh feedwater. It is indicated therefore that feedwater flow should be controlled solely by the working water level if the boiler is subjected to rapid load changes. With feedwater flow so controlled, the entrance of fresh water into the boiler is almost entirely interrupted during a sudden load rise and, conversely, large quantities of fresh feedwater will enter the boiler immediately after a sudden load drop. This action effectively stabilizes the water level and reduces the required distance between highest and lowest working levels.

Conclusion

A thorough analysis of the change taking place in a boiler during sudden and extreme steam-load variations must precede the design of the equipment. Expected magnitude of load swings and time permitted for restoring normal conditions in the boiler must be considered in determining the required volume below normal working level and the necessary capacity of fuel-burning equipment and fans. More time is available for response of control equipment when heat storage in boiler water is liberal, but large-diameter costly water drums and headers are required. A study will show whether large accumulator capacity of the boiler can be justified, or if the use of a more conventional boiler equipped with unusually large firing and fan equipment, and controls of extreme speed of response, is more economical.

[5] "Steel Encased Settings for Modern Steam Generating Units," by Max H. Kuhner, A.S.M.E., unpublished paper No. 716, 1940.

In conclusion it can be mentioned that a large number of the design points discussed as being important for boilers handling extreme load changes are equally desirable for boilers operating under more conventional central-station and industrial loads. Other points become of greater or lesser importance, proportionate to the range and rapidity of load changes. The problem thus narrows down to an intimate study of the plant load cycle, a clear understanding of what happens in a boiler during the expected changes in steam demand, and the solution of each problem thus developed by common-sense design.

Discussion

A. G. Ericson.[6] In this paper the author has capably analyzed and co-ordinated the major design factors to be considered in building a boiler for operation under extreme load variations. The importance of a large heat-storage capacity, made available by substantially increasing the size of the main boiler drum and by increasing the weight of water carried below the working level, is readily appreciated. This feature perhaps marks the essential difference from boilers of standard design and is a fundamental requirement for stable operating conditions. The quantity of heat available for flashing of steam within the allowable pressure drop sets the maximum permissible time lag between start of higher steam output and corresponding increased furnace heat release following rapid load rises. As discussed in the paper, the larger quantity of water circulating also assures that unidirectional flow will be maintained during rapid load pickups, and the larger drum size allows the necessary leeway to offset variations in water level accompanying load swings, preventing water carry-over during load rises and entrance of steam to the downflow columns during load reductions.

Installation of rapid and reliable control equipment is particularly important as the time required for adjusting the burners, damper settings, feedwater flow, and fan speeds results in a heat deficiency that must be supplied by flashing, and extended lags result in proportionately greater steam-boiler temperature and pressure losses.

The author refers to the 450,000-lb per hr boiler in the No. 3 powerhouse at Carrie Furnaces, Rankin, Pa., as an example of a quick-pickup boiler in the industrial field. Although the boiler was designed for this service, the extreme load variations originally contemplated have not developed in actual operation. This powerhouse is the most efficient in the "Monongahela Valley Power System" of the Carnegie-Illinois Steel Corporation and functions as the major generating unit with an average load rating of 85 per cent of rated capacity. Flexibility of operation is essential, for although the operators are informed in advance of general power requirements, there are inevitably some load swings that cannot be foreseen.

Operation of this boiler, described by the author, through load swings of 100,000 lb per hr has been quite satisfactory. Depending upon boiler-operating conditions at the time, the steam pressure will vary up to 50 psi and the steam temperature rarely varies more than 15 deg F. The steam quality remains good and excessive variation in the water-level, geysering, or turbulence of the water body does not occur. Following a decrease in steam load, the temperature of the superheated steam is effectively regulated by a heat-exchanger type of control. This type of control has a slight disadvantage in imposing additional resistance to the flow of steam through the superheater, therefore causing a greater differential between boiler drums and line pressure. However, it is rapid and accurate and the high-temperature element of the superheater is protected against overheating.

In this respect, it may be added that superheat is one of the most important items requiring control in the modern power plant. Excessive steam temperature is likely to cause damage through creep stresses or loss in tensile strength in both the piping and turbine parts. In addition, fluctuation in steam temperature causes opening up of the joints in the high-pressure unit and steam chest of the turbine, resulting ultimately in leaks.

The benefits to be realized from careful and efficient application of the design features discussed in this paper are dependent principally upon the degree of accuracy with which future load cycles are analyzed and power requirements foreseen.

Dan Gutleben.[7] It is refreshing for an operator of a small sugar-refinery steam plant to learn that a condition can arise which requires a public-service plant to give thought to sudden large load fluctuations. Fortunately, the problem submits to logic and arithmetic in the hands of an experienced engineer. The sudden swings were once considered an unavoidable evil visited only upon the sugar-house engineer whose plant was beset with sugar boilers endowed with untold stupidity. The public-service plants, which set the pattern for steam-plant design, accomplished an amazing job and made empirical design facts available to industrial plants long before the bondholders were repaid for the cost of their full-size research. In 1931 when the Pennsylvania Sugar Company in Philadelphia installed two boilers of 250,000-lb per hr capacity and 400 psi pressure, there was a limit to the size of forged drums, besides their cost; and welding had not become available. Accordingly, the boiler designer's talents for meeting the specification of large and sudden swings did not include accumulator operation. That existed only in the good old Scotch marine boiler of revered memory. The design staff (to which the author was attached) answered our clamor for a steam flywheel by providing a six-drum design, two of the drums being on the economizer. The steam-separating drum was forged 51 in. in diam to suit an available mandrel.

Under a premeditated test, with one boiler on base load at 400 psi pressure, the second one was increased in output by 105,000 lb per hr in 2 min with no pressure drop discernible on the gage, and with a rise of 6 in. in the water level. With the thoughtlessness of the sugar-boiler operators this was improved. Early on a Sunday morning, as the demand tapered off, one boiler was shut down and then the boys phoned that the week-end job was complete. Suddenly a large quantity of thin sweet water was found that required to be concentrated and the hour for adjournment was at hand. Without notice, the flowmeter skyrocketed. The steaming rate jumped from 75,000 lb per hr to 225,000 lb. in two minutes, accompanied by a 15-inch spear in the water-level chart. In a public-service plant such a jump in water level would wreck some turbine blades, but in a sugar refinery allowance is made for stupidity, which is a "gift from heaven." Our plant is equipped with a steam accumulator which can provide an instantaneous supply of 40,000 lb of steam for the pans, but this "dinosaur egg" happened to be empty.

H. J. Klotz.[8] The subject of this paper is most timely, as indicated by the recent increase in the number of proposed power-station installations required to operate under wide load variations. The author is to be commended for his logical analysis of the problem and his clear-cut treatment of the various factors which have important influence on the successful operation of equipment under the severe conditions described. In fact, the success achieved in developments such as he has described has en-

[6] Chief Engineer, Homestead Steel Works, Carnegie-Illinois Steel Corporation, Munhall, Pa.

[7] Parkmerced, San Francisco, Calif.

[8] Chief Power Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Mem. A.S.M.E.

couraged operators, never noted for their backwardness in asking for the impossible, to demand even greater load changes in smaller time intervals. Whereas the Harbor Steam Plant is required to handle a load increase at the rate of about 10 to 1 in a matter of seconds, operating conditions on some utility systems make it desirable to increase the ratio to 20 to 1. The resourcefulness of boiler and other equipment manufacturers, as typified by the work behind this excellent paper, leads the operators to expect that the increased severity of their requirements will be met.

Of the various factors involved, it would seem that all of the features described by the author, with the exception of the fuel-burning facilities, could be expected to be adapted without too great difficulty for the higher ratio of load increase. As pointed out in the paper, only gas and fuel oil are suitable for the minimum rates of firing required by installations of the type under consideration, and where pulverized coal is the basic fuel, supplemental use of the former fuels would be a necessity. A range in firing rates of 20 to 1 is a severe requirement which imposes serious problems in the achievement of a satisfactory design for burning even such flexible fuels as gas and fuel oil. Also, in this connection, a furnace arrangement suitable for stable combustion at the minimum firing rates is of great importance, especially in cases where pulverized coal is to be the basic fuel. Maintenance of stable combustion of course is the factor which prevents pulverized-coal firing from having the required maximum load range without use of supplemental fuel.

C. C. McKee.[9] The author has selected a subject that well deserves considerable attention; especially for those utilities which depend upon hydropower for the major portion or all of their electric supply. Usually in these systems, after installations are complete, the cost of hydropower generation, particularly secondary or dump power, is lower in the hydroplant. For this reason the power produced from steam plants is held at a minimum to reduce fuel cost. Under such circumstances steam plants are required to operate at minimum loads or about enough power to supply all auxiliary drives.

Then when a major disturbance occurs, such as loss of a transmission line, it is necessary for the steam plant to assume full load, without any previous warning, and with automatic controls to bring the steam generator into action, not only to meet all turbine requirements, but also to replace energy lost from the unit by the accumulator action while the automatic controls were bringing combustion and combustion equipment from low load to maximum capacity.

There are points of interest shown in Fig. 3 which are not indicated in the text of the paper. A study of the pressure-drop curves shows there was a slight indication for a return to normal between 30 and 40 sec. This indication was shown to much better advantage when these curves were plotted to a larger scale. The further drop following this point was caused by the feed-heating system going into action at that time. This early increase of feedwater flow has been corrected and a recent test for quick pickup indicates that increased flow of feedwater was delayed for about $1^1/_2$ min after application of load. There was little, if any, secondary pressure drop; however, the load picked up was less than that shown in Fig. 3 by about 5000 kw.

There is no rebuttal to the question that the boiler for such service must be designed to meet the severe demands; but automatic controls must also come in for their share of praise in their ability to be adjusted, timed, and synchronized to bring all steam-generating equipment to overload capacity in less than 25 sec.

Another interesting item which was checked in particular in our recent quick-pickup test is that the temperature of the feedwater entering the economizer was higher than that leaving, by some 10 deg. In the test data obtained for the curves in Fig. 3 the temperature readings for economizer feedwater in and out were read and indicated this condition, but they were considered unreliable. There was a reluctance to show this condition before checking it thoroughly. During the recent test this particular item was checked carefully, with the result that there is no hesitancy in saying that for a period of 2 to $2^1/_2$ min, the temperature of the incoming feedwater is actually higher than the outgoing feedwater from the economizer. Our estimate of the reason for this condition is that it takes about that long to move the cold water through the economizer.

W. C. Rowse.[10] The writer, who is in charge of the Steam Design Section of the Los Angeles Department of Water and Power, set forth the fundamental requirements for a steam plant which must operate in parallel with hydroelectric plants in an article which appeared[11] before the first unit of the Harbor Steam Plant went into operation. In that article it was stated that the plant should have the ability to operate over a wide range of load factors with reasonably good economy and in addition, the ability to pick up full load instantly when floating on the line at a light load, which latter requirement affected the design of the boiler.

In his paper the author clearly sets forth the boiler-design problems and how they were solved, together with the results of the first quick-pickup test with special fast-recording instruments made on October 12, 1944. The writer's discussion of the paper will consist of a brief statement of actual experience in operation.

Following the usual ups and downs incidental to placing a new unit in operation, particularly the first unit of a new plant, by March 1, 1944, the plant settled down to regular operation, except for the 4000-kw house turbine-generator unit which was appropriated by the Federal Government and shipped to Russia. Since that date the plant has been subjected to all kinds of loads from a minimum output of 64,000 lb of steam per hr continuously to a maximum output of 723,000 lb of steam per hr for 1 hr, all under full-automatic control with all burners on and including quite a few cases of quick pickup of load in emergencies. The most publicized of these was the occasion on March 18, 1946, when two airplanes collided, cutting conductors on two circuits from Boulder Dam, thus cutting off 300,000 kw of generating capacity furnishing energy for the total load of 533,000 kw being carried at that time. Although the frequency at the Harbor Steam Plant dropped momentarily to 40 cycles and continued for 10 min at 50 cycles, which shut down the fans three times, the plant picked up 55,000 kw immediately and 75,000 kw within a few minutes and carried this 75,000 kw for 1 hr. The value of the accumulator effect of the boiler was abundantly demonstrated by the ability of the boiler to supply steam even during the fan outages. These fan outages were due partly to the low frequency of energy for auxiliaries which will not occur again because the turbine-driven house generator is now installed, and partly to the setting of the protective thermorelays on the fan drives, which latter has been corrected. Since March 1, 1944, the only unscheduled outages of the steam-generating unit were for three leaks in economizer welds which were repaired over week ends.

An exhaustive test was made of this boiler during the first part of 1944, under the supervision of Mr. Sheppard T. Powell, primarily to check circulation and to recommend boiler-feedwater

[9] Mechanical Engineer, Department of Water and Power, City of Los Angeles, Calif.

[10] Engineer of Steam Design, Department of Water and Power, City of Los Angeles, Calif. Mem. A.S.M.E.

[11] "Some Features of the Harbor Steam Plant of the Los Angeles Bureau of Power and Light," by W. C. Rowse, *Mechanical Engineering*, vol. 64, 1942, pp. 773–776.

treatment and control procedures. At first a total of over 100 readings and analyses were made on each shift. As knowledge of the circulatory and other characteristics of the boiler increased, some of these readings and analyses were discontinued until, after about 6 months, a normal feedwater-control procedure was evolved. One important result of these tests was to demonstrate clearly that the boiler had been properly designed especially as to positive circulation and delivery to the superheater of steam of the required purity.

On April 24, 1946, the installation of the 4000-kw house turbine was completed, and on May 8, 1946, a quick-pickup test was made, with adjustments to the automatic controls which resulted in holding back boiler feedwater longer, and in faster opening of the fuel-oil valve. The method of increasing the load instantly is interesting. One unit at Boulder Dam was loaded with the exact load which the Harbor Steam Plant was to pick up. The 4000-kw turbine-generator unit at the Harbor Steam Plant was carrying the auxiliaries, the main turbine-generator unit was carrying 10,000 kw on the low governor block, the high governor block was set for the load to be picked up and the governor was set for a slightly higher speed than 3600 rpm. When the circuit-breakers at the Boulder unit were opened, the momentary drop in system frequency tripped the low block on the Harbor unit which picked up the load instantly with only a momentary change in system frequency. Although the load picked up (48,000 kw) was not as large as the engineers desired, the results showed a great improvement in every way over the previous test of October 12, 1944. The time required for return to normal conditions was shorter, and the maximum steam temperature lower (only 925 F at turbine throttle) than the previous test. It should be noted that both tests were made under full automatic control.

The over-all efficiency of the plant has been better than the expected efficiency. A Willan's line, in which barrels of oil per month are plotted against kilowatt-hours net output per month, results in a formula of 5250 bbl of oil (6,250,000 Btu per bbl) per month for zero net output plus incremental oil at the rate of 595 kwhr net output per barrel of oil. This results in an over-all efficiency of 11,250 Btu per kwhr net output at rated capacity (62,000 kw net output), as compared with an expected value of 11,400 Btu.

Author's Closure

It is very gratifying to have the correctness of the conclusions reached with this paper verified by the actual performance experiences with quick load pickup boilers reported by Mr. Rowse, Mr. McKee, and Mr. Ericson. These discussers' comments are of specific interest because they point the way to further improvements in steam plants required to operate under unusual load conditions.

Progress in any field is gradual and if the steps forward are not too large the result of the future step can be predicted with reasonable accuracy by the experience with the present. A boiler load increase of 1 to 10 taking place in a few seconds and without trouble was considered quite hazardous only a relatively short time ago. Mr. Klotz mentions the future requirement for load pickups in the ratio of 1 to 20. A safe boiler design for this condition is today entirely feasible if the Harbor Steam Plant experiences are made the basis for this next step.

Mr. Gutleben's reference to the human element in steam plants points to the importance of the correctly designed complete automatic-control system mentioned by Mr. McKee, but the steam-plant operator is well advised not to take the designation "automatic" too literally. Automatic control cannot replace human thought. It is especially important in plants subjected to extreme load variations that the men in charge know intimately the sequence of the various operations normally performed by the controls. The operators must be trained to take over and operate manually in case of failure of the automatic controls. Several of the pickup tests referred to by Mr. Rowse were performed to acquaint the Harbor Plant operators with all that can happen during a bona fide pickup and to instill confidence in the equipment.

The valuable supplementary data and information supplied by the discussers are gratefully acknowledged.

Acid-Cleaning of Boilers and Auxiliary Equipment

By S. T. Powell,[1] Baltimore, Md.

Acid-cleaning of equipment has been demonstrated over a long period as a useful and economical means for removing encrustations from boilers and many other types of power-plant equipment. This paper explains the use of various cleaning reagents and inhibitors, the procedure which should be followed, and the precautions to be taken to safeguard both personnel and equipment from the hazards inherent in the process.

The use of acid to remove deposits from metallic surfaces has been practiced for many years and in widely divergent industries. Such processes have been and still are extensively employed for the removal of mill scale from sheets, plates, tubes, and other products in mills making steel and copper alloys, and enormous quantities of acid are used for such services. The economy and speed gained in removing deposits in this manner have directed attention to many applications, and the treatment has been extended to the cleaning of oil wells, water mains, heat-exchange equipment, boilers, and other equipment. The extensive acidizing of oil wells to improve yields, by removing deposits and opening the formations, did much to attract interest and suggest other applications.

One of the most interesting acidizing projects,[2] which was performed more than 25 years ago, was the cleaning of the entire piping system in The Bankers Trust Building in New York City, a 35-story office building. After 12 years of service the pipe lines throughout the structure were seriously clogged with corrosion products. It was estimated that repiping would cost from $100,000 to $300,000 and could be done only at serious inconvenience to the tenants. Sections of the piping system were cleaned by means of inhibited muriatic acid in about 8 hours, after the offices were closed on Saturdays. It is surprising that this novel method of cleaning pipe lines did not receive more widespread interest at that time since the basic principle of the procedure has much merit. It is obvious that removal of deposits is merely a temporary expedient and must be followed by other forms of corrective treatment to avoid their recurrence.

The removal of scale from boiler surfaces was a logical development after the practical application of acidizing for other uses was demonstrated. The most rapid expansion of this method of boiler cleaning, however, has occurred within the past 4 or 5 years because of the need for reducing boiler outages to a minimum, and because of the shortage of labor during the war period. The effectiveness of acid cleaning is demonstrated in Fig. 1 which shows the heavily encrusted tubes of a chemical evaporator. Fig. 2[3] shows the same evaporator after being cleaned.

Thousands of boilers and miscellaneous equipment units have been cleaned by acid during the last few years and a majority of these treatments have been successful. It is a recognized fact, however, that even though inhibitors may be used, these compounds do not completely prevent corrosion, but merely reduce the rate and extent of the attack. Under ordinary conditions the corrosion of the surfaces is relatively minor if an effective inhibitor is used, if the temperature is satisfactorily controlled, the contact time limited, and subsequent washing out and neutralization of remaining acid properly performed.

Considering the large number of boilers which have been acid-cleaned, there have been only a relatively few cases where noticeable corrosion has taken place; however, there is no accurate record of the extent to which all acid-treated equipment has been corroded. From our personal experience we are inclined to believe that where the temperatures have been properly maintained and if the acid was thoroughly removed from the surfaces after cleaning, the extent of attack has been minor, and repeated acidizing may be undertaken without seriously damaging equipment. The primary object of this paper is to direct attention to the potential possibility of serious damage if the work is performed carelessly and without proper regard to precautionary measures required to insure satisfactory results.

Although acidizing of boilers is most widely practiced in power plants, still there are many other applications in this field. These may be broadly classified as follows:

1 Boilers, economizers, superheaters.
2 Deaerators, vent condensers, and stage heaters.
3 Surface condensers and heat exchangers.
4 Feedwater-treating equipment.
5 Valves and miscellaneous equipment.

Each of these applications requires special study and the treatment must be selected and applied to meet the conditions encountered.

Mechanism of Effect of Inhibitors on Acid Corrosion

There are two general concepts regarding the mechanism by which inhibitors protect steel or other metals when in contact with acid solutions. One group of investigators[4] suggests the following mechanism: Iron goes into solution at the anodic regions, forming ions, while hydrogen is discharged in equivalent amount at the cathodic areas. The cathodic areas occur principally in the narrow spaces of grain boundaries in steel or between the slag and metal in wrought iron. Most inhibitors are bases or positively charged colloids which, when present, travel to the cathodic areas and are deposited and adsorbed on the surface. The electrochemical process is stopped when the reaction at either pole is stifled, and the inhibitor thus retards corrosion. It has also been stated that the presence of the inhibitor raises the hydrogen overvoltage above any potential which can exist in the acid-metal system.

A second explanation is that the inhibiting compounds serve to blanket the entire metal surface by a protective layer of the

[1] Consulting Chemical Engineer. Mem. A.S.M.E.

[2] "Removal of Rust From Pipe Systems by an Acid Solvent," by F. N. Speller, E. L. Chappell, and R. P. Russell, Trans. A.I.Ch.E., vol. 19, 1927, pp. 165–171.

[3] Reproduced by permission of The Dow Chemical Company, Midland, Mich.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 17–20, 1946, of The American Society of Mechanical Engineers.

[4] "The Electrochemical Action of Inhibitors in the Acid Solution of Steel and Iron," by E. L. Chappell, R. E. Roetheli, and B. Y. McCarthy, *Industrial and Engineering Chemistry*, vol. 20, 1928, pp. 582–587.

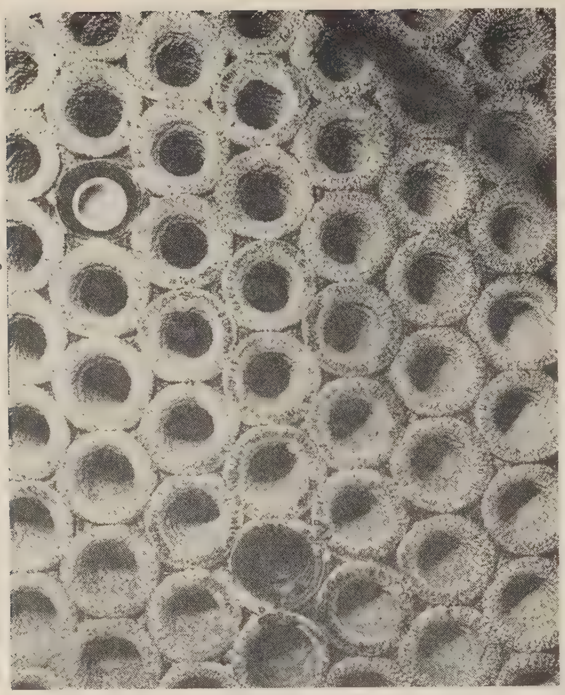


Fig. 1 Scale on Evaporator Tubes, Before Acid-Cleaning

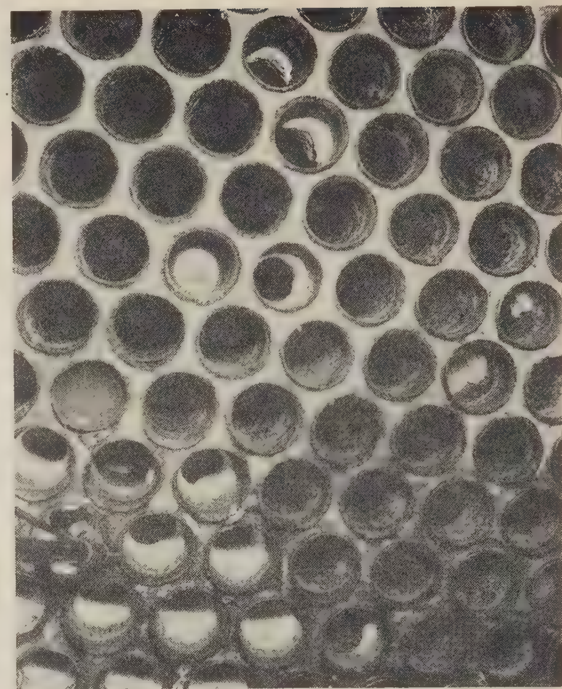


Fig. 2 Condition of Evaporator Tubes, Shown in Fig. 1, After Acidizing

organic compound used. This occurs as simple adsorption of the inhibitor ions on the metal surface, through the nitrogen atom which is a component in most organic inhibitors. The stereochemical structure of the inhibitor determines its effectiveness because the structure controls the degree of concentration that will occur.

Others have gone further and contend that inhibiting substances form a nonpermeable blanket through which the hydrogen cannot diffuse and which is limited to the metal surface, as contrasted to the oxide surface.

One investigator has submitted data to show that the action of inhibitors cannot be explained by the electrochemical or overvoltage theory nor by the formation of an impervious pore-free film on the oxide-free metal surface. Test results have been submitted to indicate that inhibitors are adsorbed simultaneously on both metal and oxide surfaces; however, the concentration is heavier on the clean metal surface. The film which forms increases the electrical resistance from liquid to metal, but the film is porous, being built up of molecule aggregates or micelles spaced closely together, similar to a pincushion filled with pins. The theory developed from these studies is that the long-chain molecules thus form long narrow channels similar to capillaries or gel structures. This condition is reflected in reduced diffusion velocity and migration velocity of the hydrogen and sulphate ions which are active in the corrosion of iron in sulphuric acid. The net result is a reduction in the corrosion of the metal, and inasmuch as a more concentrated film is present at the clean metal surface, the oxide is attacked relatively faster than the metal. This results practically in removal of the oxide and preservation of the metal surface.

It has been contended that the corrosion of some boilers during cleaning with inhibited hydrochloric acid is due largely to the formation of ferric chloride which, in itself, is a highly aggressive salt and against which the common inhibitors are not effective. According to this theory the inhibitors will retard the electrochemical reaction involving the displacement of the hydrogen of HCl by iron, but they will not prevent the oxidation-reduction reactions by which ferric chloride oxidizes metallic iron and copper to the ionic form. If metallic copper is in the sludge, ferric chloride attacks it as it does iron, producing cupric chloride which is also an oxidizing agent and a second source of trouble. It is held that the plating out of copper, or its solution to form a chloride, depends upon concentrations of the various ions present.

To limit the concentration of ferric chloride it has been suggested that the solution be constantly circulated and thereby avoid raising the ferric chloride to a level where active attack is possible. There is no assurance that circulation will bring about this desirable condition. If ferric chloride pockets or otherwise concentrates at surfaces and remains in a quiescent or semiquiescent state, it may react more slowly if only diffusion is available to carry away and remove the products of the reactions.

To support the foregoing theory, corrosion has been produced experimentally with dilute solutions of ferric chloride containing no hydrochloric acid except that resulting from hydrolysis.

It should be pointed out that if inhibitors retard corrosion by covering the entire surface and preventing contact with the acid, they should be equally effective against solutions of ferric or cupric chloride. If the impervious-film theory of inhibitor mechanism is correct, any specific vulnerability of the film to certain solutions would seem doubtful, but if the film leaves certain areas exposed, they could be attacked by oxidizing agents in the absence of an opposite electrode reaction. It is a known fact that inhibitors do not give complete protection, and the actual over-all attack which occurs may no doubt be aggravated by oxidizing agents. Whatever the role of ferric chloride and copper may be, it remains true that with proper care equipment can be acidized repeatedly without hazard.

Practical Effect of Acid-Cleaning on Boiler Steel

Studies of metal which has been repeatedly acid-cleaned show that some corrosion occurs each time, but the rate of attack is not sufficiently rapid, generally, even after many cleaning cycles, to cause serious concern. In some instances acidizing has been blamed for active attack on the metal when this treatment has not actually been responsible but has merely revealed the presence of corrosion which occurred previously and passed unnoticed because the pits and surface etching were filled and coated over with iron oxide or encrustations which the acid removed.

The effect of acidizing may be best evaluated by meticulously cleaning the surfaces before the acidizing, either by wire-brushing or local application of weak acid, and inspecting the metal for pitting, under low magnification and with tangential illumination. The same surface should be similarly cleaned and inspected after the entire unit has been acidized. The pitting shows up on stressed metal, hence for these inspections it is best to select the rolled ends of boiler tubes, the heads or nuts of highly strained bolts, the bends of angle irons, welded zones, etc. In most cases such scrupulous before-and-after inspections have not been made owing to the need for returning the equipment to service as quickly as possible after the cleaning has been completed, or because the operators were unaware of what to look for.

It is a well-known fact that the rate of attack on metals by acid or any other electrolyte is greatly affected by the internal stresses in the material. Examination of boilers which showed corrosion after acid-cleaning demonstrated that the most severe attack was on metal which had been greatly stressed. This condition was particularly pronounced on the flared end of tubes where stresses of large magnitude occurred as the result of cold-rolling. This is illustrated in Fig. 3 which shows the "honeycomb" type of attack that is characteristic in units where corrosion developed during acid-cleaning. A photomicrograph of a section of tube which has been attacked by acid is shown in Fig. 4. The honeycomb or lace form of deep pitting is more clearly demonstrated by the photomicrograph. Inspection of numerous boilers has further demonstrated that cold working of the metals used in the fabrication of internal appliances within the drum are also readily attacked by inhibited acid, even when the unstressed metal has shown no corrosion.

Fig. 3 Corrosion Resulting From Acid Attack on Stressed Metal at Flared End of Cold-Rolled Tube
(*A*, This shows honeycomb type of attack.)

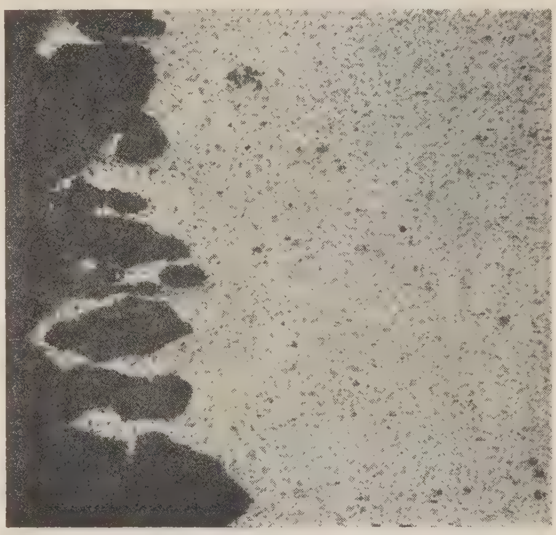


Fig. 4 Photomicrograph of Section of Boiler Tube Taken From Rolled End, Showing Honeycomb Attack During Acid-Cleaning

This type of attack has been observed by the author in a number of installations. Some of the boilers referred to had been acid-cleaned and the temperature of the acid solution was relatively high. In one case the attack was probably due to the effect of an acid contaminant in the feedwater and not to acid-cleaning. The pattern of the corrosion, however, was identical to the type and pattern of attack which was noted in acid-cleaned boilers. This occurrence tends to demonstrate that all corrosion of tube ends may not be the result of acid-cleaning, but may have resulted from some contaminant present in the feedwater. This observation suggests the need for an intensive investigation to determine all conditions which may be responsible for corrosive attack, and the avoidance of unwarranted criticism of acid-cleaning, which in the final analysis may not be responsible.

Some concern has been expressed as to the possible intercrystalline attack on steel due to the penetration of hydrogen into the steel during or subsequent to acid-cleaning. Although relatively large amounts of hydrogen may be generated during the cleaning period, investigations have shown no attack on the intercrystalline structure or any evidence of embrittlement attendant with such form of corrosion. Such investigations have not been extensive, still it seems reasonably safe to predict that embrittlement will not occur if temperature and strength of acid used is within the safe limits, and if the acidized units are thoroughly washed and acid adhering to the surfaces is completely neutralized.

Factors Influencing Effectiveness and Safety of Acidizing Equipment

There are numerous conditions which require control during

acidizing of any equipment in order to insure effective removal of encrustations from the surfaces and a minimum of corrosive attack on the metal. It is not possible to list specifically all of the many conditions which may be encountered, since there are a multitude of variables which may alter the effectiveness of the acidizing process. In general, however, the items which should be considered, and controlled if possible, are as follows:

1 Composition and concentration of acid.

2 Nature and composition of inhibitor.

3 Additional chemicals employed for their wetting, catalytic, or special solvent properties.

4 Temperature of the metal and of the solutions.

5 Circulation, mechanically induced or by diffusion.

6 Composition of the metal.

7 Physical and chemical composition of the scale to be removed.

8 Procedure followed, from taking equipment out of service through removal of acid and washing down and neutralization of acid remaining on the surfaces after treatment.

Time does not permit complete discussion of all the foregoing items, nor would it be appropriate to elaborate on all of them here, since many of them involve complex chemical reactions which are outside the scope of this paper. However, there are certain items in the list which may be profitably discussed as they are of interest to mechanical engineers and others responsible for acidizing of power-plant equipment.

Type and Concentration of Acidizing Solution. It is a common misconception that the cleaning of equipment requires merely the addition of a mineral acid, to which has been added a chemical inhibitor for the purpose of slowing down the rate of attack on metal surfaces. In some cases such treatment may effect the desired results provided temperature control and other conditions are adequately maintained. There are, however, certain types of encrustations occurring in boilers, condensers, and heat exchangers which completely resist solution by muriatic, sulphuric, or other common mineral acids. For instance, the silicate, sulphate and other scales, either alone or combined with other types of encrustations, are not appreciably soluble in muriatic acid. Where such compounds are encountered, hydrofluoric acid or other sources of the fluoride ion must be used to insure their removal.[5] Certain metallic deposits also resist solution, and where such are present, specific solvents must be employed. If oils or greases are encountered, special solvents are necessary for their removal, and effective acidizing results cannot be secured without first removing such materials.[6]

Experience has demonstrated, also, that in addition to the inhibiting chemicals, other compounds improve the effectiveness of the acidizing process. Most of these treatments involve trade secrets, and their use and application are the result and outgrowth of research and development, coupled with practical experience over a wide range of conditions encountered.

The strength of acid to be used should be governed by the time available for cleaning the equipment and the chemical composition of the acidizing solution. In the past the strength of the acid employed has been as high as 15 per cent. Recently, however, there has been a marked trend to the use of acid of lower strength, even though the cleaning takes more time. Concentrations now generally used are from 4 to 7 per cent.

Composition of Inhibitor. The technical literature on the subject of acid-cleaning and pickling of metal contains a long list of organic and inorganic substances which have been used as inhibiting agents to repress the attack on the metal, but which will permit the solution or removal of oxides or other deposits on the surface. A partial list of some of the materials which have been used is shown in Table 1. This list includes only a very small percentage of compounds which are available or have been employed. Actually, there are thousands of organic and inorganic substances which are known to have inhibiting properties. The relative effectiveness of these products varies widely, and they have specific adaptabilities for different services. Most inhibitors are harmless, but others are hazardous and should not be used. In this group are compounds of arsenic, cyanide, and some of the coal-tar by-products. At least one fatality has occurred from the use of an arsenic inhibitor. Operators should be warned against such materials and be informed of their potential danger.

TABLE 1 PARTIAL LIST OF MATERIALS USED OR INVESTIGATED FOR INHIBITING CORROSION OF METALS BY ACIDS

Niter cake	Aniline
Sumac leaves	Pyridine
Formaldehyde	Coal-tar extracts
Glue	Slaughterhouse wastes
Flour	Ethyl iodide
Starch	Paramethylaminophenyl sulphate
Gelatine	Carbozole
Glucose	Cyanide
Thiourea	Quinoline ethiodide
Arsenious oxide	Barium salts

NOTE: The inhibiting effect of the compounds listed, and many other substances, varies greatly as does their cost and availability. Some of these materials possess slight inhibiting value and some are dangerous to use. No inhibitor should be employed without specific information as to its effectiveness and the potential hazard involved in its use.

Practically, the selection of the inhibitor should be left to experts in this field, who should be advised in complete detail about the design of the equipment and the physical and chemical nature of the deposit to be removed. Organic inhibitors are marketed for cleaning operations (such as pickling) which can safely be handled by plant operators. For complex jobs involving major equipment units of the magnitude of boilers and heat exchangers, it is best to rely on the experienced contractor performing the work to select and prepare the acid solution, if the operating personnel in charge of the plant are not informed on the subject.

Temperature. Possibly the most controversial factor with regard to acidizing is the temperature at which the process is carried out, and there is fairly definite evidence that corrosion has occurred in a number of instances where temperatures have been high during cleaning. Many boilers have been cleaned at temperatures ranging from 170 to 180 F, and in one case the temperature was well above the boiling point of the solution and much damage was done. It is a known fact that the inhibiting effect of all chemicals used for retarding corrosion in acid solutions is rapidly reduced as the temperature is elevated. As the result of corrosion of equipment by acidizing at too high temperatures, there is now a trend toward maintaining the temperature of solution during the cleaning period between 140 and 160 F. If sufficient time is allowed for the cleaning, proper removal of the encrustations can ordinarily be effected at the lower temperatures.

In Fig. 5 is shown the extent of corrosion which occurred on the face and edge of caps in waterwall headers. It is probable that the relatively high temperature was responsible for the condition.

Circulation. Various schemes have been used requiring special equipment for applying the cleaning solution to the boiler. In some instances the operators merely inject the chemical into the top drum, while others prepare the solution in a tank and pump it into the boiler and recirculate from the tank back into the boiler. In the final analysis there is not now available sufficient evidence to prove that either circulation of acid by means of a pump, or circulation induced merely by diffusion current is a preferable scheme to use. Both procedures have been followed, in

[5] "Chemical Removal of Scale From Heat-Exchange Equipment," by F. N. Alquist, C. H. Groom, and G. F. Williams, Trans. A.S.M.E., vol. 65, 1943, p. 719.

[6] "Scale Removal by Chemical Means," by M. E. Brines, *Power Plant Engineering*, vol. 44, May, 1940, p. 47.

Fig. 5 Corrosion on Face and Side of Cap Showing Honeycomb Type of Attack

large groups of boilers cleaned, without active corrosion occurring in either case.

Composition of Metal. The composition and the cleanliness of the metal are important factors in resisting or accelerating corrosion during acidizing of equipment. It has been observed repeatedly that low-grade steel containing slag inclusions or other impurities has been readily attacked during acid-cleaning, while the drum and tubes were not affected. In one installation the tops of practically all the acorn nuts used in the fabrication of the internal baffles were destroyed, and a similar condition was noted in a number of other units. To what extent the acid was responsible has not been fully determined, but it undoubtedly accelerated the rate at which the metal was attacked. It is a well-known fact that the material from which such nuts are constructed is low-grade metal, as might be expected, especially during the war years.

Ordinarily, black iron pipe, especially if cold-worked, is readily

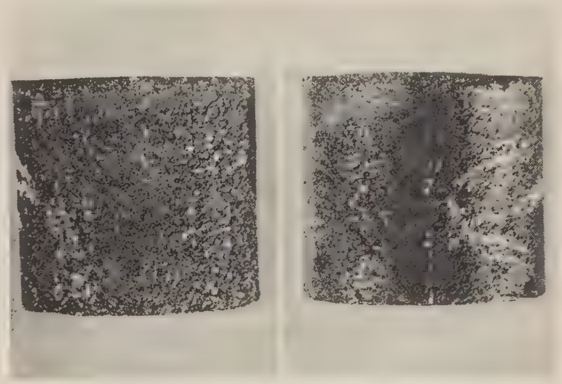


Fig. 6 Nipple Corroded During Acidizing, Illustrating Effect of Acid on Low-Grade Steel Containing Slag and Other Impurities

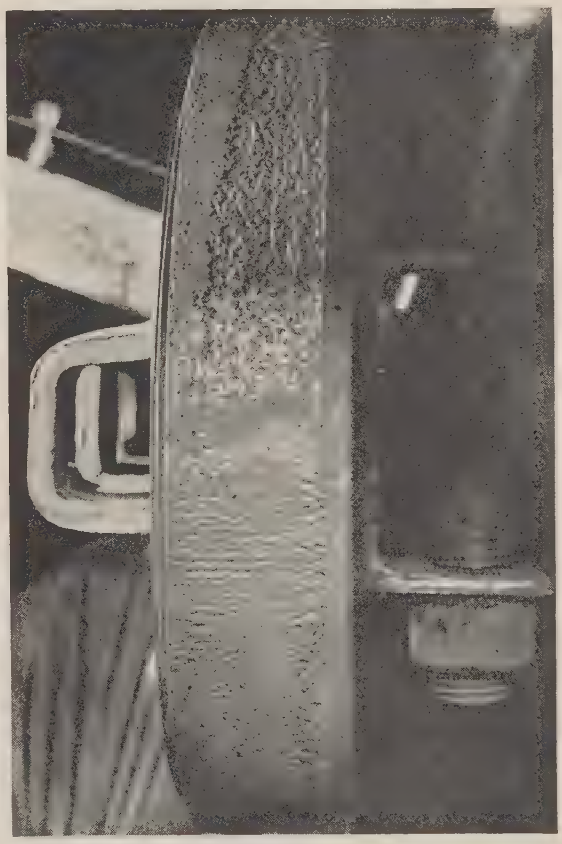


Fig. 7 Corrosion of Manhole Cover Showing Deep Surface Corrosion by Acid Attack

(Exact cause for selective corrosion not accurately determined but probably presence of slag inclusions or change in metal structure due to flame-cutting. Note wasting and honeycomb type of attack.)

TABLE 2 TYPES OF DEPOSITS WHICH HAVE BEEN FOUND IN BOILERS, AUXILIARIES, AND MISCELLANEOUS POWER-PLANT EQUIPMENT

Principal ion	Common name	Mineralogical name	Formula determined by x-ray analysis
Group A Boiler and Superheater			
Barium	Barium carbonate	Witherite	$BaCO_3$
	Barium sulphate	Barite	$BaSO_4$
Calcium	Calcium carbonate	Calcite	$\beta CaCO_3$
	Calcium sulphate	Anhydrite	$CaSO_4$
	Calcium sulphate	Hemihydrate	$CaSO_4.0.5H_2O$
	Calcium hydroxide		$Ca(OH)_2$
	Calcium phosphate	Hydroxyapatite	$Ca_{10}(OH)_2(PO_4)_6$
	Calcium phosphate	Whitlockite	$\beta Ca_3P_2O_8$
Magnesium	Magnesium hydroxide	Brucite	$Mg(OH)_2$
Sodium	Sodium sulphate	Thenardite	$Na_2SO_4(V)$
Silicates	Silicon oxide	Quartz	SiO_2
	Sodium-iron-silicate	Acmite	$Na_2O.Fe_2O_3.4SiO_2$
	Sodium-alumina-silicate	Analcite	$Na_2O.Al_2O_3.4SiO_2.2H_2O$
	Sodium-alumina-silicate	Natrolite	$Na_2Al_2Si_3O_{10}.2H_2O$
	Sodium-alumina-silicate	Noselite	$Na_8Al_6Si_6O_{24}.SO_4$
	Sodium-calcium-silicate	Pectolite	$Na_2O.4CaO.6SiO_2.H_2O$
	Calcium silicate	Wollastonite	$\beta CaSiO_3$
	Calcium silicate	Xonotlite	$5CaO_6SiO_2.H_2O$
	Calcium-alumina-silicate	Cancrinite	$4Na_2O.CaO.4Al_2O_3.2CO_29SiO_2.3H_2O$
	Magnesium silicate	Serpentine	$3MgO.2SiO_2.2H_2O$
Iron	Iron oxide	Ferrous oxide	FeO
	Iron oxide	Hematite	Fe_2O_3
	Iron sulphide	Triolite	FeS
Group B Condenser, Evaporator, and Heat Exchanger			
Calcium	Calcium sulphate	Anhydrite	$CaSO_4$
	Calcium sulphate	Gypsum	$CaSO_4.2H_2O$
	Calcium carbonate	Aragonite	$CaCO_3$
	Calcium-sodium-silicate	Pectolite	$4CaO.Na_2O.6SiO_2.H_2O$
	Calcium-magnesium-carbonate	Dolomite	$CaCO_3.MgCO_3$
	Calcium fluoride	Fluorite	CaF_2
Copper	Copper sulphide		CuS
	Copper carbonate	Malachite	$CuCO_3.Cu(OH)_2$
	Copper iron sulphide		$CuFeS$
	Copper oxide	Cuprite	Cu_2O
Magnesium	Magnesium silicate	Forsterite	Mg_2SiO_4
	Magnesium hydroxide		$Mg(OH)_2$
	Magnesium silicate	Serpentine	$3MgO.2SiO_2.2H_2O$
Iron	Iron oxide	Magnetite	Fe_3O_4
	Iron oxide	Goethite	$Fe_2O_3.H_2O$
Sodium	Sodium-aluminum-silicate	Noselite	$Na_8Al_6Si_6O_{24}.SO_4$
Group C Steam and Condensate Line			
Ammonia	Ammonium bicarbonate	Teschemacherite	NH_4HCO_3
Iron	Iron carbonate	Siderite	$FeCO_3$
Sodium	Sodium sulphate		$Na_2SO_4(III)$
Group D Miscellaneous Power Plant Equipment			
Calcium	Calcium carbonate	Calcite	$\beta CaCO_3$
	Calcium carbonate	Aragonite	$\lambda CaCO_3$
	Calcium aluminate		$3CaO.Al_2O_3.6H_2O$
	Calcium phosphate	Phosphate	$Ca_3P_2O_8.H_2O$
Magnesium	Magnesium hydroxide	Brucite	$Mg(OH)_2$
Group E Turbine			
Copper	Copper sulphite	Chalcocite	Cu_2S
Nickel	Nickel oxide	Bunsenite	NiO
Silicates	Silicon oxide	Quartz	SiO_2
	Silicon oxide	Cristobalite	SiO_2
	Silicon oxide	Opal	SiO_2
Sodium	Sodium sulphate	Thenardite	$Na_2SO_4(V)$
	Sodium-carbonate-sulphate	Burkeite	$Na_2CO_3.2Na_2SO_4$
	Sodium chloride	Halite	$NaCl$

attacked by acid even though better grades of metal show no visible attack. The type and extent of the attack are illustrated in Fig. 6. This shows a nipple used as a spacer in the construction of internal baffles in a boiler which had been acid-cleaned. An identical type of corrosion was observed in a number of other units employing the same grade of material.

Corrosion of such parts in boilers or in other equipment is not of major importance and is presented here merely to illustrate the susceptibility to corrosion by acid when the metal in contact with the solution is not metallurgically clean.

Active corrosion occurred on manhole covers in two boilers operating under different conditions. The pattern of the attack is shown in Fig. 7. In one instance it was indicated that the metal may have been fairly hot when the boiler was acidized. In the second case the unit was relatively cold, but the feedwater had been contaminated with an acid compound prior to acidizing. It is believed that the condition of the surface of the metal may have made it more susceptible than other portions of the boiler. These conditions were noted on boilers supplied by different manufacturers.

Composition of Deposits. Table 2[7] shows a number of scales which have deposited in boilers and appurtenant power-plant equipment. Frequently such scale will be composed of several formations, each having specific solution characteristics. Furthermore, the composition and extent of the deposit will vary widely in different positions of the same equipment. This condition is demonstrated by the analysis of the encrustations removed from tubes in two sections of the same unit.

Procedure for Acid-Cleaning Boilers. There is no standardized procedure for acid-cleaning of equipment and no standard solution may be employed. It is obvious that the treatment must be guided and adjusted to meet the many variables encountered. Where the work is performed by a commercial company, the job is taken by contract and the contractor furnishes all the necessary appliances and the material for the complete acidizing, including washing down and neutralizing the residual acid when the cleaning of the equipment has been completed. Generally, it is preferable to have the work done by a trained and experienced contractor rather than to attempt the work by local operators, unless

[7] Acknowledgment is made to Dr. F. N. Alquist, of The Dow Chemical Company, who supplied much of the data included in Table 2.

the necessary trained personnel have been developed who can employ the proper technique for effective cleaning and prevention of the industrial hazard involved in such work.

The program of operation which is usually followed when cleaning boilers consists of the following steps, in the order given:

1 Drain the unit, cool, and inspect.

2 Adjust the temperature of the metal to the desired level by the addition of feedwater.

3 Drain.

4 Fill with inhibited acid.

5 Soak for a predetermined period.

6 Continue tests to determine reduction in strength of acid If concentrations drop rapidly and continue to fall, draining and refilling with the solution may be required.

7 When soaking period is completed, drain the unit.

8 Flush two or three times by pumping feedwater into unit and draining as the pH of the test samples indicates the necessity.

9 When flushing is completed, drain boiler and refill with 1 to 2 per cent solution of alkali (soda ash or caustic soda).

10 Drain, then fill with feedwater above the level in the top drum which was reached by the acid, and then drain to the normal water level.

11 Heat with a slow fire and steam at 10 to 25 psi for 2 hr or more.

12 Empty boiler completely if unit is to be inspected, otherwise lower concentration to desired point by blowing down, furnishing feedwater as required.

Whenever possible, boilers should be drained and inspected after acidizing, and a record kept of the condition of the surfaces for future reference and comparison with conditions observed at subsequent inspections.

There may be many deviations from the foregoing operation, but the general plan is indicative of the steps which should be followed to lessen the danger of corrosion.

Hazards

Since all mineral acids are highly corrosive, especially when concentrated, it is obvious that much care should be exercised to protect both the personnel performing the work and the equipment which is being cleaned. If the treatment is carried out by experienced operators who are cognizant of the dangers involved and take the necessary precautionary measures, the resulting hazards should be negligible Aside from the danger of the corrosive characteristics of the acid, the treatment involves a very definite hazard of fire and explosion if the hydrogen generated is not effectively removed and steps taken to prevent its discharge into the atmosphere. Extreme care should be used, also, to prevent hydrogen from pocketing in any equipment which is being acidized.

How many accidents have occurred from explosions from hydrogen during or subsequent to acidizing is not known but during the past 2 years three accidents of this kind have been reported. In one instance two workmen were killed and extensive damaged resulted from an explosion of a boiler. The accident was caused by failure to vent the gas from the boiler drum, the pocketed hydrogen being ignited by a spark from a flashlight used after the boiler was opened for inspection.

This accident demonstrated the extreme danger of acidizing equipment by persons not thoroughly informed of the reactions which occur when the acid is in contact with steel. The two other accidents reported were also explosions resulting from ignition of accumulated hydrogen gas. These accidents were not fatal to personnel, but resulted in appreciable damage to the equipment.

A number of injuries to personnel, resulting from burning or gassing, have come to the attention of the author. There is one case on record during the past year where an operator cleaning a boiler by inhibited acid was killed by arsenious oxide released by the inhibitor which contained an arsenic compound. This type should never be used, although such inhibitors are quite effective for retarding corrosion.

Conclusions

Experience clearly demonstrates that acid-cleaning of equipment is a useful and economical means for removing encrustations from boilers and many other types of power-plant equipment. However, it should never be undertaken or practiced by inexperienced persons who are not fully aware of the dangers involved. Furthermore, acidizing should be practiced only if and when such treatment is necessary. It is a very questionable procedure for any plant repeatedly to acid-clean equipment as a routine maintenance schedule. Such a program is unwarranted both from the standpoint of cost and over-all depreciation of equipment.

No equipment should be treated with acid unless there is assurance that the acid used can be completely removed after the cleaning process has been completed. Most powerhouse equipment is susceptible to cleaning by flushing and flooding with neutralizing alkaline solutions. However, there are certain appliances that are so constructed that adequate removal of acid cannot be effected, or can be done only with great inconvenience and at appreciable cost. In this group the nondraining superheater is an example of apparatus which does not lend itself to acid-cleaning. In some cases where the superheaters may become badly fouled with deposits which are not water-soluble, acid-cleaning may be practiced. If such work is undertaken it should be attempted only when there is assurance that adequate provision has been made for flushing and neutralization when the work is completed and before the unit is returned to service.

Discussion

F. N. Alquist.[8] The acid-cleaning of boilers and auxiliary equipment has been called to the notice of the Society several times in the past few years. Briefly, attention is directed to a discussion by C. H. Groom, G. Williams, and the writer,[9] on scale removal from heat-exchange equipment. Before this publication and since, we have been working in close contact with the power industry and many others on cleaning problems. The present paper is a valuable contribution to the literature on the subject. Data are also given in a paper by J. L. Wasco and the writer.[10]

Acid-cleaning is an economical method of cleaning power-plant equipment. However, the hazards of the acid-cleaning process should be fully recognized so that only experienced persons will be entrusted with cleaning. All engineers recognize that acidizing should be done only when necessary.

The author's statement, "It is a very questionable procedure for any plant to acid-clean equipment as a routine maintenance schedule," requires further discussion. For example, when should a high-pressure boiler be cleaned? The proper answer is to turbine selected tubes to find the extent of fouling. If the fouling rate thus determined is such that annual cleaning is re-

[8] Organic Research Laboratory, The Dow Chemical Company, Midland, Mich.

[9] "Chemical Removal of Scale From Heat-Exchange Equipment," discussion by C. H. Groom, G. Williams, and F. N. Alquist, Trans. A.S.M.E., vol. 65, 1943, pp. 719–722.

[10] "Chemical Removal of Calcium Sulfate Scale," by J. L. Wasco and F. N. Alquist, *Industrial and Engineering Chemistry*, vol. 38, April, 1946, pp. 394–397.

quired, then the plant is acid-cleaned annually on a routine maintenance schedule. These decisions are made by the plant engineers on the basis of cost and over-all depreciation values. The practice of the writer's company on its boilers shows variable times for extent of fouling depending upon water treatments and loads.

It is well that the power industry be made aware of the distinction between corrosion due to operation and corrosion due to improper cleaning procedures. A proper conclusion to be drawn is to maintain treating temperatures as low as possible during cleaning and as stated, preferably between 140 and 160 F.

The author has ably discussed the theories of inhibition. Fifteen years of research by the writer's company on acid inhibitors indicates that both the inhibitor and the acid concentration must be selected so as to give a low corrosion rate under cleaning conditions.

Experience gained on thousands of installations where forced circulation has been compared to circulation by convection, shows that forced circulation is only occasionally necessary.

That the effect of inhibited acid varies as the metals differ in composition has been known in the art for a long time. Power-industry engineers have found that operation corrosion as well as corrosion in cleaning depends upon the composition of the metals involved. Considering Figs. 3, 4, 5, 6, and 7 in the paper, no statement can be made as to how much corrosion is due to operation and how much is due to cleaning.

The steps outlined under procedure for acid-cleaning boilers are good but, under the war-emergency conditions, these steps were not always used because of lack of time; however the engineers of the power industry now follow them more closely.

Finally, the hazards of handling acids and venting the hydrogen that may be produced during the cleaning operation are well known both to the chemical and power industries. With proper safety precautions and helpful papers such as this one, the power industry can take economic advantage of acid-cleaning methods.

R. E. Hall[11] and C. E. Kaufman.[12] We are in agreement with the general facts on acid-cleaning of boilers as developed by the author. More incisive analysis of some of the factors involved brings to light some significant considerations that require emphasis.

Acid used for the process is supplied with an inhibitor, that is, a substance whose purpose is to minimize or prevent attack of the acid on the boiler steel, while not interfering with its dissolving effect on other substances such as the calcium carbonate and phosphate, the iron oxide, etc., comprising the boiler sludge and scale. Concentration of the inhibitor in the acid-cleaning solution is a function of the type of inhibitor used, concentration of acid employed, and temperature at which the job of acid-cleaning is done. Thus one can depend upon the inhibitor to protect against direct attack of the metal by the acid within rather generous temperature limitations, if the concentration of inhibitor required in relation to the acid is consistently maintained, and if the products of reaction of the acid with the boiler sludge or scale are not of themselves potent in their attack on the metal, regardless of the inhibitor.

As a matter of fact, these products of reaction divide into two classes, one of which is innocuous as regards the boiler metal; the other of which is highly aggressive in attack on the metal, even with any usual inhibitor present in the acid solution. Materials which give rise to the first class comprise the calcium and magnesium salts of the boiler sludges and scales; those giving rise to the second class are the iron oxides, either magnetite or hematite. Whether formed in the feed line by the action of dissolved oxygen and thence entering the boiler, or whether formed in the boiler itself by reaction of boiler metal with the bonded oxygen of the boiler water, these oxides represent high oxidizing capacity. Concentrated bit by bit as the boiler is operated and held harmless by the insolubility of the oxides in the boiler water, they await only the trigger action of the acid solution which dissolves them to bring this concentrated oxidizing capacity into play against the boiler metal. Metallic copper, so frequently associated with the iron oxides, may enter into but cannot initiate the oxidation reactions, nor can it contribute thereto. In fact, by plating out on the metal it may even be protective in some measure.

Let us explore conditions in a thin layer of the acid-cleaning solution adjacent to the metal when the sludge or scale consists primarily of insoluble calcium and magnesium salts, and the normally quite low percentage of iron oxide. The acid dissolves the deposit, forming calcium and magnesium chloride, both of which are stable salts devoid of oxidizing capacity. Because of this, they originate no attack on the metal, and the protection provided by the inhibitor in the acid remains intact, unless undue temperature elevation has adversely affected protective conditions. Barring undue temperature, whether flow of the acid-cleaning material over the surfaces is provided for by means of circulating the solution or whether there is or is not considerable agitation of the solution by dissolving such material as carbonate, a protective concentration of the inhibitor in relation to the acid content of the solution prevails at the surfaces.

Let us now explore conditions in the thin film of acid-cleaning solution adjacent to the boiler metal, when the sludge in the boiler is mainly magnetic oxide Fe_3O_4, which dissolves forming ferric and ferrous ions. It is readily demonstrated that the ferric ion is the most important factor in promoting attack.

Fundamentally the aggressiveness of solutions containing ferric ion can be explained on the basis of the following reaction

$$2Fe^{+++} + Fe \longrightarrow 3Fe^{++} \quad [1]$$

It will be noted that this process does not intrinsically involve the disappearance or formation of an acid or a base. Therefore it might be presumed that the inhibitors customarily employed to stifle acid attack on steel would not be well adapted to minimize the reaction of ferric ion with metal. Laboratory tests have confirmed this point.

In the laboratory we have run a number of agitated-solution tests at 150 F, for 6 hr, in which the reality of ferric-ion attack is clearly apparent. For example, when 1 per cent Fe_3O_4 was added to an inhibited acid solution, attack on an inserted steel strip was increased nearly threefold as measured by weight loss. The loss, representing about one third of the potential attack as just defined in Equation [1] was undoubtedly limited by the incomplete solubility of the added iron oxide, behavior which of course may be found in full-scale cleaning operations. When the ferric material was present in considerably greater concentrations and as completely soluble chloride, without acid but with a standard inhibitor present, attack actually considerably exceeded that predicted from reaction [1].

In the removal of deposits from boilers by acid-cleaning, when magnetic iron oxide, for example, goes into the circulating acid, what quantitative potential has it for producing damage? If it is assumed that 1000 gal of cleaning acid builds up to contain homogeneously distributed in solution 1 per cent or 83.3 lb of material originally Fe_3O_4, 20 lb of iron can be dissolved if Reaction [1] goes to completion. If, for example, the dissolution of iron is assumed to take place evenly within 3-in. tubing, the general loss of metal will amount to less than 0.001 in. If, however, the ferric ion is not homogeneously distributed in the acid solution, but rather is selectively concentrated in localized areas

[11] Hall Laboratories, Inc., Pittsburgh, Pa. Mem. A.S.M.E.

[12] Hall Laboratories, Inc.

rich in magnetic oxide, then serious pitting or wastage in such areas may be expected.

Avoidance of such corrosion we believe is best effected by steadily maintained circulation of the acid-cleaning fluid so that localized high concentrations of ferric chloride cannot occur, and by limiting concentration of ferric chloride in the acid-washing solution through adequate blowdown and replenishment of acid solution, thus protecting against too much loss of metal in general corrosion of the surfaces by even the homogeneously distributed ferric chloride which develops mainly at the surfaces or in particular spots in the boiler. Satisfactory control of ferric-ion concentration can be established by analysis.

For the boilers that develop magnetic oxide by the bonded-oxygen process in regions where high heat release renders intolerable any decrease in circulation, acid-cleaning does a specially significant job, in that by cleansing the surfaces of iron oxide, and thus bringing back the original designed circulation in the tubes, the conditions for reaction of the bonded oxygen of the boiler water with the tube metal are minimized. Thus periodic acid-cleaning as required can keep circulation at these points maintained at a level where any formation of iron oxide will be slight, and the boiler will be dependable.

Naturally, acid-cleaning should not be looked upon as something to supplant normal careful procedure in taking care of the boiler. Nevertheless, especially in cases such as those comprised in the circumstances just discussed, it can play a very effective part. Likewise, alkaline-cleaning processes in some instances, dependent upon the composition of material to be removed, are more feasible than acid-cleaning.

No mention has been made of acid-cleaning operations involving nonferrous metals. It would be interesting to know if such cleaning has been conducted and the general types of inhibitors, if any, which proved useful.

L. E. Hankison.[13] This paper is especially valuable because it tells of the dangers that may be encountered, not only to the equipment being cleaned, but also to the personnel engaged in doing the cleaning. The definite warning given by the author that acid-cleaning should never be undertaken or practiced by inexperienced persons who are not fully aware of the dangers involved, should not be ignored. It is evident that this work should be done either by someone commercially in the acid-cleaning field, or by someone who is professionally able to prescribe the proper acid to be used, its strength, the inhibitor to be used, the proper temperature and the length of time for its contact, and then the best way to dispose of the partially spent acid, as well as the proper material and method of neutralizing or removing the residual acid from the equipment. This service presumably can be had only at a cost which must be charged against the benefits that may be obtained by acid-cleaning.

There is another, and probably a much greater cost that must be borne by those who indulge in this means of keeping their equipment in service. It is the loss in efficiency and capacity of the equipment from the time the surface begins to foul until the deposit is of sufficient magnitude to justify removing the equipment from service for acid-cleaning. By using the average between the clean condition and the fully fouled condition as the approximate continuous loss, it can readily be appreciated that this cost is quite considerable and should be avoided if at all possible.

It is our belief that it is much cheaper and very much more satisfactory to secure the services of competent water-conditioning consultants whose aim is to avoid the necessity of any cleaning rather than to acid-clean and bear the costs that have just been outlined. We believe that with this consulting service and properly trained operating personnel to carry out their recommendations, acid-cleaning, or any other type of cleaning that requires equipment outages is unnecessary. Of course, this implies properly designed equipment operated at ratings so that steam pockets, hideout, or reverse circulation is not experienced.

Acid-cleaning has a very important place in power-station operation when the equipment is run at ratings beyond the normal ability of the heat-transfer surface to function. In this case the surface may oxidize or change to form an insulation to a point where overheating and failure will follow. Acid-cleaning may remove the deposit that has been formed and prevent the equipment from failing.

There is another situation where acid-cleaning is beneficial and, in fact, is the only practical means of restoring equipment to serviceable operating condition. During the war, when materials such as condenser tubes and the personnel for their installation were very hard or impossible to obtain, condenser leakage allowed silica to build up in some of our boilers to a point where it was deposited as an incrustation that caused tube failures. Acid-washing restored the boilers to good operating condition, where they have continued to function with the aid of the proper water-conditioning previously mentioned.

One other acid-cleaning experience should be mentioned. Due to water leakage through the packing of the boiler feed pumps into the bearing cavities, it was found difficult to keep oil in the bearing reservoirs. An oil-company representative recommended that the straight mineral oil which had always been used be replaced with a compounded oil that would not float out so easily. This was done and the bearing situation was remedied, but this oil was emulsifiable and such oil as did leave the bearing reservoirs would not separate out in the drip collectors but was carried through into the boilers. There it mixed with the boiler sludge forming a tarry mass containing a small percentage of oil that adhered to the surfaces and caused burnouts. It was thought that increased alkalinity would dissolve this mass but it was not cleared up by this means, so we decided to acid-clean to dissolve the embedded sludge. Acid-cleaning partially dissolved this mass. It also removed much of the protective coating of iron oxide and probably other iron compounds that had been formed on the water surface of the tubes by some reaction of the treating chemical, and which are considered beneficial and desirable.

The presence of this iron compound can be well illustrated by our boiler-gasket experience. Due to scarred or cut hand-hole-seat surfaces, it was considered necessary to use solid copper gaskets to make these parts tight. It was recognized that copper gaskets between the iron or steel surfaces and in the presence of the alkaline boiler water as the electrolyte were not conducive to long gasket life, yet we were never able to find even the slightest attack on these gaskets and they were considered entirely satisfactory. After the boilers were acid-cleaned new gaskets would not last 3 months and we have records of heavy new gaskets practically entirely disappearing in 2 months of operation. This caused us temporarily to abandon copper gaskets. We now believe that the protective coating has been restored by the continued use of treatment, and are trying a few copper gaskets which we hope will again give satisfactory service.

As a further indication that acid-cleaning is detrimental in removing the beneficial iron-compound coating, our experience with corrosive barnacles should be related. After the boilers were acid-cleaned, large quantities of iron-oxide scale were found lodged in the tube headers and also in the tubes that had sluggish circulation. Frequent boiler outages were necessary until all this scale was removed from the system. These outages allowed the internal boiler surfaces to be exposed to atmospheric air, and

[13] Superintendent, Efficiency Department, West Penn Power Company, Pittsburgh, Pa. Mem. A.S.M.E.

small rust spots were formed. These rust spots may have provided the insulation that promoted barnacle growth and boiler-tube failures resulted. This is being given further study. It is now believed that the protective surfaces previously described have been fully restored, as failures of this type have not been experienced during the last $1^1/_2$ years.

While it is agreed that cleaning with acid is beneficial under certain extraordinary conditions, we wish to emphasize the part of the author's conclusion which says, "It is a very questionable procedure for any plant repeatedly to acid-clean equipment as a routine maintenance schedule." We believe that good housekeeping, coupled with intelligent water-conditioning to eliminate the need for acid-cleaning, is the preferred method of operating power-station equipment.

W. R. LaMotte.[14] The writer's company has been particularly interested in the cleaning of boilers by this method. It was first tried in 1939 to clean one of the 350-psi boilers at Kearny generating station. The results were so gratifying, both from the standpoint of low cost and degree of cleanliness obtained, that acid-cleaning has been standard practice at that station ever since. In 1940 the practice was extended to the 1250-psi boilers in the Public Service system. These high-pressure boilers were first acid-cleaned at half yearly intervals, but more recently they have been cleaned annually. As a result, some of them have been cleaned ten and eleven times.

This experience tends to confirm what the author has written concerning the precautionary measures which are advisable in the use of acid for boiler cleaning. The best inhibitor procurable should be used and solution and metal temperature should be kept well below the temperature at which the manufacturer states the inhibitor will break down, the time of contact should be limited, and the various hazards should be recognized and adequate protection provided.

In fact, we differ only with the following statement of the author: "It is a very questionable procedure for any plant repeatedly to acid-clean equipment as a routine maintenance schedule. Such a program is unwarranted both from the standpoint of cost and over-all depreciation of equipment." We look upon acid-cleaning as the normal and best way to clean periodically the internal surfaces of high-pressure boilers. Nothing so far discovered has indicated that measurable harm has been done during the past 5 years.

Perhaps one reason for the foregoing has been the practice of using low acid concentrations, in the range $^1/_2$ to 1 per cent HCl, as contrasted with the range of 4 to 15 per cent discussed by the author. These satisfactory results have been achieved using presently available inhibitors, and certainly improvement in inhibitors can be expected in the future. High-pressure boilers now on order are so designed that the tubes cannot be cleaned by conventional methods; only by the use of liquids can the internal surfaces be reached. It has been our experience that the art of treating the boiler water has not progressed sufficiently to dispense entirely with internal cleaning.

E. L. McDonald.[15] Our company was one of the early users of inhibited acid for cleaning power-plant equipment. For years we have acid-cleaned various heaters, feed lines, pumps, economizers, etc., all under the guidance of the station chemist.

Much was learned from our early experience. For example, do not try to clean old equipment too fast or too well with any scale-removing compound. Scale removed from the tube sheet of a steam-crane boiler by trisodium phosphate exposed loosely rolled tubes and caused a great number of leaks that did not exist when scale covered them. Scale removed from the cast-iron junction boxes of an economizer by acid-cleaning exposed porous spots in the box that did not leak before the removal of scale.

When the shortage of labor during the war first prompted us to employ a commercial acid-cleaning company, we had the idea we could pump the acid to the boiler through the blowoff line which emptied into a sewer at a convenient location for tying-in the acid truck, thus saving the cost of an acid feed line. After a short period of time the acid pumped through the line removed the scale and a great number of leaks developed, which again were not evident when the scale protected them.

All of this is in keeping with the author's statement to the effect that acid is frequently blamed for corrosion which actually existed before the acid was applied and was not evident until the acid removed the scale. The writer has observed corroded surfaces, similar to the type shown in the paper, in boilers and turbine casings that had not been acid-cleaned.

While we have made substantial savings by acid-cleaning with our own crew and equipment, we have not found acid-cleaning of large steam-generating units, as performed by a commercial cleaning company, to be very cheap. The average on five boilers with about 20,000 sq ft of surface each was \$84 per 1000 sq ft of surface. This was about 40 per cent higher than the previous mechanical cleaning cost. On these five boilers a 5 per cent acid solution was used. On three boilers cleaned at a later date, using $3^1/_2$ per cent solution, the cost dropped to \$52 per 1000 sq ft of surface. This was still about 15 per cent higher than the previous mechanical cleaning cost. However, two items definitely favored acid-cleaning, i.e., a shorter boiler outage, and a better job of cleaning. The fact is that it is practically impossible to clean the modern steam-generating unit mechanically.

Mention has been made that the control of acid temperature is important. A thermometer in the acid line can tell a false story. High "spot" temperatures can exist in a boiler if heat is transmitted from brickwork not previously cooled before acid treatment is applied.

The author comments on the possible metal loss due to acid-cleaning; however, it should be remembered the old rotary tube cleaner also removed metal and, before water treatment reached its present stage, more than one boiler tube had to be replaced due to this metal loss. Also, in our own case, we believe the loss of metal, due to corrosion and erosion on the outside of the tube, would exceed any loss of metal on the inside. During acid-cleaning, metal test strips have been hung in boilers, carefully weighed before and after the acid-cleaning, and have given us an idea of the possible metal loss.

Mention is also made of the hazard of generating hydrogen gas when acid-cleaning, and the author comments on a particular boiler explosion. A very serious explosion also occurred last year when acid-cleaning a condenser at the Wichita Station of the Kansas Gas & Electric Company. Details of this are covered in an article by E. D. St. John,[16] which should certainly be read by anybody contemplating acid-cleaning of such equipment.

While acid-cleaning of any equipment has merit, it should not be undertaken except under the guidance of a competent chemist or other competent help.

F. U. Neat.[17] In connection with acid-cleaning, the writer believes that considerably more importance should be attached

[14] General Superintendent of Generation, Public Service Electric and Gas Company, Newark, N. J. Mem. A.S.M.E.

[15] Kansas City Power & Light Company, Kansas City, Mo. Mem. A.S.M.E.

[16] "Repairs to Steam Turbine Unit Following Hydrogen Explosion," by E. D. St. John, *Power Plant Engineering*, vol. 50, no. 3, March, 1946, pp. 64–68.

[17] Assistant to Superintendent of Power Production Stations, Consolidated Gas Electric Light & Power Company, Baltimore, Md.

to the possibility of hydrogen embrittlement. Pickling embrittlement is too well known in the steel world to lead us to do anything but go well on the safe side with regard to removing hydrogen. Particularly in the case of boilers, it is quite possible that we may open up crevices in seams and around tubes and these crevices must be stopped. However, if we should permit the rerolling of tubes or calking of joints without making certain the removal of occluded hydrogen, then we are guilty of negligence. The writer's practice is to rinse the boiler thoroughly, after which a solution of 2 to 5 per cent soda ash is boiled in the boiler for 8 to 12 hr at from atmospheric pressure to 50 psi under open vents before the boiler is opened for inspection and certainly before any cold-working of the boiler metal is permitted. This boiling procedure accomplishes reasonable removal of hydrogen gas from the boiler and occluded hydrogen from the metal, largely prevents after-rust, and partially re-establishes the protective coating of magnetic oxide. It may also assist in the loosening of remaining scales which are incompletely soluble in acid.

Attention is also called to a remark by the author which may be misconstrued. In his discussion of the hazards involved he states that the discharge of hydrogen to the atmosphere must be prevented. The writer feels certain that the author really meant that the vents must be piped to the outside in such a manner as to make reasonably sure that discharged hydrogen may not re-enter the building and that these vents be taken from as near the top of the equipment as is possible. If vents are left open during the fill and drain-rinsing operations and then during the boiling-out procedure, we are assured nearly complete removal of hydrogen in the equipment.

H. B. Oatley.[18] Deposits on the heat-transfer surfaces of several types of vessels are very generally recognized by designers as well as users. This is particularly true for vessels or pipe lines conveying liquids at increasingly higher temperatures and pressures.

Attention has been given to the use of purification systems for the removal of these undesirable elements which cause failure of the apparatus. Acid-cleaning has been used with beneficial results by those who, like the author, realize the dangers of such use. The reverse is also true.

The paper has called attention to the outstanding features of this acid treatment and the dangers of such cleaning are clearly and emphatically stated in the author's conclusions.

The increasing severity of operation has brought the designer to the doorstep of the metallurgists for better material, which is to be more highly resistant to deteriorating effects resulting from impeded heat transfer. It may also be said that the chemical industry is looked to by those who require more effective and less dangerous scale-removing methods, entailing the use of chemical inhibitors.

W. F. Ryan.[19] The development of inhibited acids for the cleaning of metal surfaces is just another example of the fact that many of us were born 30 years too soon. When the writer recalls his filthy and laborious efforts to clean boilers and oil coolers in the old 74th Street Station, he wishes that the Dow development engineers had not been so dilatory in the development of acid-cleaning. His successors in the operating field can have these jobs performed with less labor and worry than he experiences when he sends his shirt to the laundry.

In saying less worry, he is not relying entirely on the fact that it may be easier to replace a boiler than to get a new shirt. The author's precautionary remarks are timely and well taken but, on the other hand, the hazards are very slight if one is willing to pay somebody else to worry about them. Most of the adverse experience with which the writer is familiar, has arisen from misguided efforts to save a few dollars by doing the job without experienced supervision or adequate equipment.

In addition to the routine cleaning of boilers, there are special conditions under which the use of acid-cleaning may pay tremendous dividends; for example, in a paper mill it was discovered that the two high-pressure boilers had accumulated an unusual amount of sludge due to changes in the raw-water supply. One tube was blistered and concern was felt for the rest of the heating surfaces. One would not ordinarily think of using acid to remove sludge, but based upon previous experience it was anticipated that a production loss equivalent to one half of the mill's output for at least 7 days would be involved in shutting down the boilers, allowing them to cool sufficiently, cleaning them thoroughly, and getting them back on the line. This was at a time when the mill output was heavily oversold and such a loss of production would be a major catastrophe. Dowell Incorporated was consulted regarding the practicability of using acid in this situation and found that both boilers could be cleaned in a single day. The two boilers were satisfactorily cleaned by circulation of inhibited acid and the entire operation consumed less than 19 hr. The cost of the acid-cleaning was of the order of $500, while the production hours that were gained were worth many thousands.

Another instance of high return on a small investment was experienced in the cleaning of cooling coils in large process tanks. For years these coils which are used with raw hard water, were installed with 3 times the calculated surface in order to compensate for the gradual accumulation of scale. At the end of 2 or 3 years the coils were removed and replaced. The coil itself was not extremely expensive, but the replacement required draining and cleaning the tank with attendant loss of valuable materials. By the use of acid the coil can be cleaned in place with no loss of production or of materials. Life of the coil will be at least equal to that of the tank, and when new tanks are installed the cooling coils can be designed in correct proportion to the demand for heat transfer.

The use of acid-cleaning in power plants and process industries is still in its infancy, with new time-saving and money-saving applications arising every day. Every operating or maintenance engineer who is confronted with the periodic cleaning of metal surfaces should investigate thoroughly what can be accomplished by this method.

H. G. Thielscher.[20] We have acid-cleaned eleven boilers in our plants with satisfactory results. The work has been performed by a commercial chemical company. The strength of the acid solutions used was 5 per cent, the length of time contained in the boilers varied from 5 to 14 hr, and the temperature of the boiler metal and acid solution ranged from 150 F to 190 F. Forced circulation of the solution was not used. We do not know the nature or composition of the inhibitors used as this is considered secret information by the commercial companies. Subsequent examination of these boilers has not revealed any definite evidence of acid corrosion.

Included in the boilers, which were acid-cleaned, was a new 525,000-lb per hr 675-psig 900 F total steam temperature 3-drum boiler, which was acid-cleaned immediately after the conventional boiling and drying-out period. The results obtained were excellent, and an appreciable saving was effected in the time

[18] Vice-President and Consulting Engineer, The Superheater Company, New York, N. Y. Mem. A.S.M.E.

[19] Stone & Webster Engineering Corporation, Boston, Mass. Fellow A.S.M.E.

[20] Mechanical Engineer, Potomac Electric Power Company, Washington, D. C. Mem. A.S.M.E.

and labor ordinarily required to prepare the drums for Apexiorizing. In addition, we believe a more permanent Apexior job was obtained because of the cleanliness of the metal surfaces after acid-cleaning. We expect to continue this practice of acid-cleaning new boilers before they are placed in service. The superheater in this boiler was the vertical nondraining type and was filled with clean warm water before admitting the acid solution to the boiler. In some of our older boilers we had deposits of analcite. The acid-cleaning did not remove these deposits but did soften them up so they were readily removed by turbining.

We agree with the author that there are corrosion and explosion hazards attached to the acid-cleaning of equipment, but, like him, we believe these hazards are negligible when proper procedures and precautions are observed.

G. F. Williams.[21] This paper gives an exceptionally clear picture of chemically removing scales and sludges with acid-base solvents. The author is to be congratulated on this excellent paper which so ably relates the history of acid-cleaning and points out that its effective application involves a careful consideration of all pertinent factors. When one considers the complexity of the questions involved, which deal not only with chemical engineering but also with metallurgy, electrochemistry and other sciences, it is even more apparent that all factors must be carefully weighed by one thoroughly trained and experienced in this line of work in order that the accumulated scale may be removed safely.

It is interesting to note that over 25 years ago the entire piping system of a large building was successfully cleaned of scale by the use of acid-base solvent but, as the author points out, "It is surprising that this novel method of cleaning pipe lines did not receive more widespread interest at that time since the basic principle of the procedure has much merit." Possibly the difficulties experienced on this job were sufficiently serious to deter this type of work in the immediate future and the present commercial use of acid-cleaning may be due in no small degree to the experience gained in using millions of gallons of inhibited acid-base solvents during the last few years for increasing the flow of oil wells.

In only a few cases has any corrosion been noted in boilers following acid-cleaning, and in most of these there was considerable evidence that the corrosion could have taken place during operation of the equipment rather than during the cleaning job. Our company, during the last 10 years, has made over 50,000 separate treatments using inhibited acid-base solvents on oil wells and industrial equipment, and in less than 0.1 per cent of these jobs has any question been raised concerning the possibility of corrosion due to the use of inhibited solvent. In fact, it is the opinion of many operators of large steam-generating equipment that the metal loss from cleaning with acid-base solvents is less than that which results from mechanical cleaning methods.

The manhole cover, shown in Fig. 7 of the paper, showed definite corrosion upon inspection following an acid-cleaning job, but as this unit was not inspected immediately prior to cleaning, there is room for considerable uncertainty as to the cause of the corrosion. Representatives of the writer's company were present with the author and others at the time this inspection was made. Other places on the edges of the manhole covers in this boiler showed occasional large corrosion "pits" much deeper and much larger than anything shown in Fig. 7. It seems practically impossible that such a large amount of metal could have been removed from such a small area in the few hours' time in which acid was in contact with the metal. Probably factors other than cleaning with inhibited acid-base solvent contributed to this corrosion.

[21] Development Engineer, Dowell Incorporated, Tulsa, Okla. Mem. A.S.M.E.

One of the conclusions to this paper, viz., "It is a very questionable procedure for any plant repeatedly to acid-clean equipment as a routine maintenance schedule," might be clarified. Certainly any preconceived schedule for periodical cleaning without any knowledge of the rate at which scale is formed would be questionable, but whenever there is sufficient scale accumulation to warrant its removal, acid-cleaning is the best and sometimes the only feasible way of removing it. After the deposition rate of scale in a boiler is known and therefore the frequency with which it must be removed, it seems entirely reasonable that acid-cleaning could be done on a routine maintenance basis.

Author's Closure

The comments presented on the author's paper by the several discussers are constructive and justify much consideration by those interested in the subject.

The theory on the mechanism of corrosion, as brought out by Messrs. Hall and Kaufman, requires consideration and indicates the need for further research along these lines. Although there is much disagreement among engineers as to the relative merits of circulation to avoid concentration cells, as suggested by Dr. Hall, this is a desirable procedure under some circumstances and, where it can be carried out in a practical manner, circulating the solution may conceivably overcome the tendency to cause spotty corrosion of the metal. Regardless of the advantages offered by circulation during cleaning of apparatus there are many cases, especially those involving cleaning of large boilers designed with complex circuits, where circulating the solution can not be readily affected. It is obvious that uniform circulation through a large unit is difficult and it is doubtful if pumping the solution through units will insure much better movement of the solution over the metal than may be obtained by normal thermal circulation which occurs during the cleaning process.

Regardless of these apparent operating difficulties to carry out the procedures for avoiding corrosion as suggested by Dr. Hall, the problem warrants further study. It is hoped that the Hall Laboratories, Inc., and other interested groups will continue to study and develop means for overcoming many of the troublesome problems encountered in the acid-cleaning technique.

Dr. Alquist's statement that "The effect of inhibited acid varies as the metals differ in composition" has been known in the art for a long time. Power-industry engineers have found that corrosion during operation as well as corrosion in cleaning depends upon the composition of the metals involved. This has not been fully appreciated by many operators. Why some metal corrodes during acid cleaning to a greater extent than others, when the cleaning procedures are practically identical, is not clearly understood but no doubt is related to the composition of the metal and other metallurgical characteristics.

The instructive and informative research now being carried out by The Dow Chemical Company under the guidance of Dr. Alquist and his associates should reveal much valuable information and place acidizing of equipment on a much more exact basis than is now possible with our limited knowledge. It is hoped that this and other organizations carrying on research in this field will release their findings from time to time. Such dissemination of data should serve as an invaluable guide to the profession.

The author is in agreement with many of the comments made by Mr. Hankison, since our experiences have paralleled his regarding a number of difficulties described. The acid attack on copper gaskets during cleaning of boilers has been encountered

many times with varying degrees of severity. These difficulties and other precautionary remarks mentioned by Messrs. Hankison, LaMotte, and MacDonald and the other discussers, demonstrate the need for intelligent planning and control for safe and effective treatment of any equipment by inhibited acid solutions.

With regard to Mr. Neat's comments, namely, "Attention is also called to a remark by the author which may be misconstrued," the author's precautionary remark, of course, related to adequate venting to the outside atmosphere. We regret if our statement in the text has been misconstrued by others. It is presumed to be obvious that the avoidance of a fire or explosion hazard would require complete and adequate venting outside the building by suitable facilities.

The aim and purpose motivating the preparation and presentation of the author's paper was to direct attention to the economic and practical value of acidizing boilers and other auxiliary equipment. The very favorable comments by so many engineers and chemists indicate that designers and operators agree with our viewpoint, differing only in details of application of the process. There are a number of unsolved factors which are not well understood and therefore may not be adequately evaluated. It is hoped that our effort has focused attention on these matters and will stimulate further study of the problem.